Design and Fabrication of a 400kW Pre-Prototype Engine Budget Period 4 Final Report

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ABSTRACT

The research and development effort of a new kind of compressor and hybrid engine is presented. The excellent performance of these two products arises from superior performance of rotating supersonic shock wave compression.

Several tasks were performed in compliance with the DOE award objectives. A 2.5:1 pressure ratio Rampressor rig was designed, built and tested. A higher pressure ratio Rampressor rig was designed and constructed. Conceptual design of a Rampressor Turbine was completed. An ultra-low NO_x combustor was designed, built and successfully tested. Implications to greenhouse gas generation of the Ramgen technologies were investigated.

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CHAPTER 1: EXECUTIVE SUMMARY

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1 Executive Summary

1.1 Budget Period 4 Executive Summary

This report summarizes the results of Ramgen technology development efforts over the Budget Period 4 contract from December 1, 2002 through March 31, 2006.

The Ramgen technology represents the unique application of ramjet principles to compression, combustion and power generation. The Ramgen Rampressor™ will provide superior compressor performance through rotating super sonic shock compression for use in compressing air, CO₂, or in combination with combustion and expansion to provide superior engine performance for power generation. The Advanced Vortex Combustor (AVC) can be used in many different applications including conventional turbo machinery, with the Rampressor in an engine application, or as a unique technology capable of burning low BTU fuels with high flame speeds in IGCC applications. These efforts were funded by Department of Energy cooperative agreement number DE-FC26-00NT40915.

The primary objective of advancing the research and development effort on the Ramgen engine technology was accomplished in the 40 month long Budget Period 4, by reaching significant technological milestones.

- Rampressor rig tests showed the rotor produced compressed air near the target design point and efficiency estimates are on track to achieving new industry standards (November 2003).
- A strategic decision was made in February 2004 to focus on development of the Rampressor compressor product.
- Ramgen calculations indicate that a 10:1 pressure ratio Rampressor compressor, conventional burner and standard non-air cooled turbine could achieve near 40% cycle efficiency if the system were recuperated.
- Completed the Rampressor Turbine Design Report topical report (June 2004)
- Completed Preliminary Design of the 2nd Rampressor for higher performance: pressure ratio, efficiency and speed.
- Designed, built and tested an Advanced Vortex Combustor in partnership with NETL Morgantown and achieved less than 3 ppm NO_x emissions with stable flame characteristics.
- Completed final design, build and initial testing of the 2nd Rampressor for higher performance: pressure ratio, efficiency and speed.

A design review task (Task 20.0) was planned to occur during BP4. Since testing was not completed during BP4, the design review was postponed for the next budget period.

The successful accomplishment of these tasks will culminate in the test rig compression data in the next budget period (DE-FC26-06NT42651 BP1). The successful demonstration of compression performance that matches our analysis tools is a critical step to demonstrating superior performance in a CO₂ compressor for IGCC application. The completion of this step and the Rampressor Design Review will allow Ramgen to move ahead on the CO₂ Compressor design.

1.2 DOE Budget Period 4 Tasks: Design and Fabrication of a 400kW Pre-Prototype Engine

1.2.1 Task 12.0 - Manufacture and Assembly of Compression Inlet Test Rig

Under this task, the Recipient shall manage the manufacture and assembly of the Compression Inlet test rig. Detailed manufacturing drawings shall be created. Procurement and manufacture of the hardware shall be close coupled to individual drawing completion. The test rig assembly will include "as built" drawing modifications, critical spares, and multiple rotors for discrete performance investigations and diagnostics. Engine assembly will include fit check at the manufacture, simple checkout of bearings, instrumentation, clearances, etc.. This task is complete when the engine and the rig are ready to spin in the test cell

1.2.2 Task 13.0 - Testing of the Compression Inlet Test Rig

In this task, the Recipient shall manage the testing of the Compression Inlet Test Rig, and lead the technical investigation into the performance of the Compression Inlet. A suitable facility shall be contracted to perform the testing. Test objectives include, but are not limited to, overall compression performance, tip seal leakage effects on performance, inlet starting, inlet bleed requirements and optimization, and inlet acoustics.

1.2.3 Task 14.0 – Complete the Conceptual Design of the Rampressor-Turbine Pre-prototype Unit

The goal of the Pre-prototype Unit is to demonstrate positive power production. In Budget Period 4, the Recipient shall complete the Rampressor-Turbine Pre-prototype Unit conceptual design task. This unit will be designed to utilize the Compression Inlet Test Rig demonstrated in Task 13.0. A static conventional burner will be incorporated into the unit along with a simple power turbine to extract energy from the simple Brayton cycle machine. The test rig will be flexible and adaptable. Features necessary to accommodate the testing, evaluation, and diagnosis of various configurations will be incorporated. Fixtures and instrumentation provisions to measure performance, gather data, and general condition monitoring shall be designed into the test unit.

1.2.4 Task 15.0 – Greenhouse Gas and Alternative Fuels Study

The Recipient shall assess the benefits from the reduction in Greenhouse Gases as a result of the Ramgen technology. This assessment shall include a review of the sources of information on emerging markets for greenhouse gas reductions. The assessment will also identify the greenhouse gas reduction potential as a consequence of the Ramgen engine introduction and Ramgen compressor technology.

1.2.5 Task 16.0 – Vortex Combustor Design, Fabrication & Testing

Under this task, the Recipient shall provide on- and off-site technical consulting support to the NETL Office of Science and Technology (OST) in order to develop a conceptual design of a vortex combustor including design of the pre-mixer, acoustic analysis, test plan development, and CFD analyses. In accordance with Exhibit 1 to this agreement (U.S. DEPARTMENT OF ENERGY FUNDS-IN AGREEMENT FOR RESEARCH AND DEVELOPMENT/STANDARD AGREEMENT FOR THE CALIFORNIA ENERGY COMMISSION, ENTITLED “DEMONSTRATION OF A LOW EMISSION ADVANCED VORTEX COMBUSTOR”), OST will oversee detailed design activities that include necessary facility modifications for testing, final combustor design, design review, and reports preparation, and prepare documentation for a “Hazard Analysis and Safety Review” necessary at NETL to permit actual hardware testing. In addition, OST will finalize process and fabrication drawings, and fabricate test and necessary facility hardware. OST will oversee the testing of the vortex combustor, and will lead data evaluation and all post-test analyses.

The Recipient’s consulting support will emphasize the disciplines of combustion, mechanical and thermal design. The Recipient shall lead the design of the pre-mixer that is a sub-component of the combustor, and translate OST’s conceptual design of the combustor into a detailed final thermal and mechanical design. The Recipient will also be present and participating in the test investigation of the testing of the Advanced Vortex Combustor and providing on-site technical support and immediate initial interpretation of the data.

1.2.6 Task 17.0 – Complete Tests on the 1st Rampressor Rig

In this task, the Recipient shall manage the testing of progressively more ambitious rotor configurations, namely a segmented rotor spinning at higher RPM than tested in Task 13.0, for the Compression Inlet Test Rig. A suitable facility shall be contracted to perform the testing. Test objectives include, but are not limited to: Overall compression performance; tip seal leakage effects on performance; inlet starting; inlet bleed requirements and optimization..

1.2.7 Task 18.0 – Complete Design and Build of the 2nd Rampressor Rig

Under this task, the Recipient shall manage the design of a higher compression pressure ratio (~9:1) segmented rotor. The design will include: A feasibility study to identify the critical design issues and ensure several configuration candidates are identified; a conceptual design review to identify the leading design configuration and define the high-level aero and mechanical design requirements; identification of the long lead manufacturing and procurement items that must be purchased in advance of completing manufacturing drawings; a

preliminary design review to confirm the design configuration is ready for the detail design phase; and a final design review to confirm the design configuration is ready to issue manufacturing drawings and begin procurement and manufacturing of the parts. By the end of Budget Period 4, the Recipient will host a Final Design review and begin procurement of test rig hardware.

1.2.8 Task 19.0 – Greenhouse Gas Reduction Marketing Opportunities

The information base established in the Task 15.0 study provides a solid starting point for an in-depth reduction plan in Task 19.0 that guides the marketing plans for economically viable opportunities for greenhouse gas reductions to be identified for the Rampressor, the Rampressor Turbine, and the Ramgen Engine. The study shall identify and prioritize the specific applications for the Rampressor which are the most economically viable opportunities to reduce greenhouse gases.

CHAPTER 2: 1ST RAMPRESSOR TEST RIG (TASKS 12, 13, & 17)

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2 1st Rampressor Test Rig

2.1 Overview

2.1.1 Program Goals

The high level Rampressor-1 (1st Rampressor) technical program goals consisted of:

- Demonstrate Basic Rampressor Rotor Operation
- Establish/Demonstrate Supersonic Compression Based On Flight Inlet Technology
- Characterize Rotor Efficiency and Flows

Achieving these goals would then lay the ground work for the Rampressor-2 (2nd Rampressor) program which is to be traceable to a 10:1 overall pressure ratio application.

2.1.2 Program Objectives

The specific technical objectives to achieve the program goals consisted of:

- Design rotor test rig intended to explore key Rampressor issues
- Supersonic stage performance (develop compressor map)
- Evaluate tip leakage effects
- Validate design tools and methodologies
- Characterize system spin-up and starting
- Identify surge limits and characteristics

Test results used to support follow-on supersonic stage design and integration at increased pressure ratios and efficiencies

2.2 Design

The design of the Rampressor-1 test rig consisted of Aerodynamic (static and rotating), Mechanical (static and rotating), and Peripherals design efforts. The aerodynamic design consisted of:

- Static Flow Path -
  - Stationary Inlet
  - Pre-Swirl Nozzles/Inlet Guide Vanes
  - Stationary Exhaust
  - Exhaust Collection
- Rotor Flow Path -
  - Inlet Geometry
  - Rotor Diffuser Geometry

The mechanical design consisted of:

- Static Structure –
  - Inlet
  - Pre-Swirl Nozzles/Inlet Guide Vanes
  - Tip Clearance Control

- Forward and Aft Bearing Housings
- Rotor Structure –
  - Monolithic Disk
  - Bearings/Rotordynamics

The peripherals design consisted of:

- Instrumentation
- Controls
- Data Acquisition
- Data Reduction
- Secondary Flows (into and out of the test rig)
- Test Cell Integration
- Test Rig Support
- Compressor Test Rig Drive System & Control

2.2.1 Overall Aero Design

The overall aero design of the 1st Rampressor (i.e., Rampressor-1) was primarily driven by the desire to demonstrate the Rampressor concept in the lowest technical risk, least costly, and quickest manner. Use of a monolithic conventional titanium alloy rotor was thought to be the least costly and quickest manufacturing time approach, with the alternative being a segmented design similar to that used on the Ramgen Engine prototype. As such, mechanical limitations of a monolithic rotor required relatively low rotor tip speeds to have acceptable stress levels. A pure monolithic rotor without certain aerodynamic features was capable of relatively high tip speeds (~2,000 ft./sec), but once all of the aerodynamic features were implemented the resulting stress concentrations reduce the allowable tip speed significantly. The key parameters of the aero design of a Rampressor are:

- Rotor & Overall Pressure Ratio (PR)
- Rotor-Relative Mach Number (M_{rel})
- Amount of Pre-Whirl (Pre-Swirl Mach - M_{ps})

The mass flow scales as the square of the rotor flow path mean diameter and the required RPM (for a given PR/ M_{rel} design) scales as the inverse of the diameter – i.e., smaller mass flow rates at a given PR required higher rotation rates. The rotor flow path features/geometry are primarily determined by the design M_{rel} with the rotor pressure ratio and physical tip speed determined by the combined values of M_{ps} and M_{rel} . – the lower the M_{ps} , the higher the rotor pressure ratio at a constant M_{rel} .

2.2.1.1 Why 1.6 Mach Design

The Rampressor-1 rotor was designed for Mach 1.6 M_{rel} operation with a M_{ps} value of approximately 0.75. This combination reduced the aero risk in two areas – inlet starting and tip leakage effects. “Starting” the inlet becomes more difficult as the design M_{rel} increases (the term “starting” refers to getting the flow through the converging area portion of the rotor to be completely supersonic). Tip leakage effects potentially become more problematic as the design rotor pressure ratio (and hence M_{rel}) increases – the higher the pressure ratio the larger the pressure differential becomes across the strakes, thus producing the potential for increased leakage rates.

In addition to the reduction in aero risk levels, the reduced M_{rel} and increased M_{ps} produces a required rotor mean line/tip speed of only around 1,000 ft/s which produce much lower disc stresses and reduce the bearing operating requirements (thrust loading and DN values).

2.2.2 Rampressor-1 Mean Line Design

The first step in designing the Rampressor-1 rotor test rig was to use a mean line design code, developed by

Ramgen, to determine the basic operating parameters such as pre-swirl Mach number and outflow angle, rotor geometry (strake angle, annulus hub and shroud radii, contraction ratio/throat radius, exhaust radius, and rotor axial length), rotor relative Mach number, and RPM. For a given desired pressure ratio, pre-swirl Mach number, and mass flow rate, etc. the rotor mean line design can be determined.

2.2.2.1 Station Numbering

A consistent station numbering scheme was developed in order to appropriately identify and track the design and performance of the various stationary and rotating components. The station numbering convention is listed in Figure 2.2.2-1.

Station Number	Station Name
0	Ambient
1	Inlet Duct Conditions
2	PreSwirler Inlet
2.1	PreSwirler Exit
2.2	DeSwirl Exit
2.3	PSN Inlet
2.4	PSN Exit
2.5.0	PSN Exit / Rotor Inlet
2.5.1	Rotor Inlet /Pre Oblique Shock 1
2.5.2	Post Oblique Shock 1 (Inlet panel 1 Exit)
2.5.3	Pre Normal Shock
2.5.4	Post Normal Shock
2.6.0	Rotor Exit
2.7	Rotor Exit
2.8	Exit Guide Vane, Exit Conditions

Figure 2.2.2-1: Overall station numbering scheme used by the mean line performance code.

The station numbering for the mean line model most relevant to the rotor design, illustrated in Figure 2.2.2-3 and Figure 2.2.2-2 with the test rig cross section, is as follows:

- 2.3 – Inflow conditions that would be achieved by an upstream compressor, experimentally this will be the conditions delivered by the Boeing test facility.
- 2.4 – Downstream of the pre-swirl nozzles that will be used to achieve the desired Rampressor inflow conditions. Stationary inflow conditions for the Rampressor rotor.
- 2.5 – Rotating frame of reference inflow conditions for the Rampressor rotor. Note: Static flow properties are the same at station 2.4 and 2.5.
- 2.6 – Rotating frame of reference rotor outflow.
- 2.7 – Stationary frame of reference rotor outflow.

Figure 2.2.2-2: Cross section of rotor test rig with station numbering call outs corresponding to mean line conditions.

Figure 2.2.2-3: Station numbering in the rotor reference frame. Note that the flow is from bottom towards the top of the figure.

The mean line design code assigns absolute and relative total pressure losses to various component sections as well as assigning certain geometry and relative speeds (Mach number). Typically, from a desired pressure ratio (rotor-only or stage), mass flow rate, and type of rotor geometry (strake angle, inlet aspect ratio, etc.) is calculated a rotor RPM, pre-swirl Mach #, efficiency estimates, and rotor outflow conditions (swirl angle and Mach). There are many loss parameters that are assigned which their values are based on CFD, 2-D boundary layer theory, rules of thumb (e.g., supersonic diffuser/inlet total pressure recovery functions like the MIL-E-5007D). Development of the mean line code is an ongoing task and will be anchored via CFD and experiment results.

2.2.2.2 Rampressor-1 Mean Line Design

A parametric analysis of various design variables available in the mean line code was performed. The specified variables included:

- Compression Ramp Entrance Rotor relative Mach number, $M_{rel}(M_{2.5}) = 1.6$
- Pre-swirl Mach, $M_{psn}(M_{2.4}) = 0.77$
- Rotor Inlet aspect ratio (AR) = 2.2
- Rotor diffuser exhaust Mach, $M_{2.5} = 0.5$
- Number of inlets = 3

The resulting design of the rotor was:

- Inlet Hub/Shroud Radius: 5.036"/5.714"
- Exhaust Hub Radius: 4.957"
- Inlet Contraction Ratio (CR) = 1.18
- Physical Rotation Rate @ 510° R of 20,482 RPM
- Corrected Speed of 20,656 RPM (100% corrected speed)
- Strake Helix Angle (on inlet mean line) of 7.62°
- Design Tip Speed/Mean Line Speed at 100% Speed = 1,021ft/sec/961 ft/sec
- Design Rotor-Only Pressure Ratio = 2.59:1
- Design Rotor (Flow Path) Adiabatic Efficiency = 0.90

This was the mean line design chose at the start of the design process. The mean line design code continued to evolve with the final design being updated even while experiments were ongoing. The final mean line predictions of the as-built hardware will be discussed in later sections.

2.2.3 Static Flow Path Aero

The static flow path consists of the three components the inlet, PSN and Exhaust duct. The inlet is defined as the flow path from the Boeing flow conditioner to the Pre-swirl Nozzle (PSN). The PSN is a turbine nozzle that accelerates and swirls the air into the rotor. The exhaust duct collects the rotor discharge and routes it to a plenum which is then collected to a mass flow meter.

2.2.3.1 Inlet

The inlet flow path consists of two surfaces the hub and the shroud. The hub surface is a cylinder with a diameter that matches the rotor inlet hub diameter. The shroud consists of two cylinders one large to interface with Boeing's flow conditioner and one small to match up with the rotor inlet shroud. An ASME low beta bellmouth shape is used to transition between them. The inlet also contains a rig structural element a thick plate with holes in it (see Figure 2.2.3-1). The key dimensions of the inlet are shown in Figure 2.2.3-2. The inlet was designed to have minimal flow Mach numbers before entering the pre-swirl nozzle as shown in Figure 2.2.3-3.

Static pressure taps at the throat of the bellmouth along with a discharge coefficient calculation (based on Boeing Multiple Choked Venturi MCV measurements) allowed for an additional flow calculation (used during self-aspiration testing).

2.2.3.2 PSN

The PSN was designed by Concepts/NREC. A copy of the PSN design report is contained in the Chapter 2 Appendix. Ramgen selected the PSN to rotor spacing to be 1.5 duct heights to minimize rotor inlet distortion due to PSN wakes and end wall effects. Typical turbo-machinery spacing is around 0.5 duct heights but it is not unusual to see very large spacing (i.e. Turbofan Fan Rotor to Fan Stator spacing). During testing the PSN was moved much closer to the rotor and there was little to no impact on rotor only performance (see test results).

2.2.3.3 Exhaust

The exhaust flow path consisted of an annular duct with 18 radial holes that discharged into a plenum as shown in Figure 2.2.3-4. This configuration was chosen to be tolerant of all rotor exit flow angles and provide a uniform static pressure distribution at the exit of the rotor (via the 18 discharge holes). A moveable cylinder with holes was added to the exhaust system as a rig throttling mechanism. Unfortunately flow leakage around the cylinder was so large that the rig could not be throttled to design point. A valve was added downstream of the plenum for throttle and the rig throttle mechanism was left wide open. The 18 radial holes in the cylinder were sized to pass 2.5 times the exit corrected flow at design point. The stationary radial holes were sized to pass 4.0 times the exit corrected flow at design point.

A brief CFD effort was undertaken to insure the exhaust system could pass the flow at design point and provide

a uniform rotor exit static pressure distribution. A segment (1/18th of the exhaust flow path) of the exhaust system was simulated with the computational mesh shown in Figure 2.2.3-5. The inflow boundary condition corresponded to the rotor outflow mass flow and flow angle and the outflow of the radial hole set to a constant static pressure that corresponded to the estimated pressure in the exhaust plenum with downstream pressure losses taken into account.

The CFD results confirmed that the design could pass the design mass flow with very little rotor exit static pressure distortion as shown in and .

Note the high level of swirl at the inlet and how nearly all flow disturbances are near the radial outflow hole.

Note there is no rotor exit static pressure distortion.

Figure 2.2.3-1: Illustration of the basic features of the inlet system

Figure 2.2.3-2: Illustration showing the key dimensions of the inlet duct.

1-D, $C_p=1.4$, Design Point, No Blockage

$P_{tin}=22.05\text{psia}$, $T_{tin}=510\text{degR}$, $\dot{m}=3.128\text{lbm/sec}$

Figure 2.2.3-3: Illustration showing the inlet flow path design point Mach numbers at various locations.

Figure 2.2.3-4: Cross section showing details of the exhaust system.

Figure 2.2.3-5: Grid utilized for simulating the exhaust – 1/18th of exhaust flow path modeled.

Figure 2.2.3-6: Vector plot of velocity magnitude at on-design operating conditions.

Figure 2.2.3-7: Contour plot of static pressure illustrating the low degree of static pressure field distortion at the rotor exit produced by the exhaust hole

2.2.4 Rotor Aero Design

The aerodynamic design of the Rampressor rotor is broken up in the various discrete sections of the flow path that are designed either from the mean line design outputs (strake angle, pre-inlet height, inlet width, etc.), 2-D analytics or CFD (compression ramp profile), or 1-D analysis (rotor diffuser). The following sections of the rotor design detail how the final RP-1 design was arrived at through various methods.

2.2.4.1 Rotor Components

This section intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.2.4.2 Pre-Inlet Design

This section intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.2.4.3 Compression Ramp Design

This section intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.2.4.4 Analytical Rotor Diffuser Design

The rotor diffuser is defined as the area expanding section downstream of the inlet throat and serves to contain the normal shock and then diffuse the Mach 0.75 flow to the desired amount (typically around Mach 0.5). The rotor diffuser is composed of two sections: the portion just downstream of the minimum area which typically has very little area expansion and is intended to re-accelerate the flow to approximately Mach 1.3 from Mach 1.2 at the throat so that the normal shock can be stabilized in this section; the second component is the subsonic diffuser that diffuses the post normal shock further. The key issues involved in the design include:

- Normal Shock Location / Stability

- Total pressure recovery
- Subsonic Diffuser Design
  - Margin against separation
  - Viscous drag
  - Total pressure ratio
  - Discharge Mach No. Sensitivities

The analysis approach consisted of modeling the rotor diffuser as a 2-D planar type (see Figure 2.2.4-1) thus enabling the use of planar 2-D diffuser maps and test results of set expansion angle and area ratios to avoid separation or unsteady flow effects in addition to diffuser design handbooks and BL analysis to predict total pressure loss. A preliminary design was based on planar straight 2-D diffuser performance data for high Mach number inflows with the following inflow and outflow requirements:

- Diffuser inflow (post NS) Mach = 0.79
- Diffuser outflow Mach = 0.5 (selected to optimize system performance)

Preliminary design shows good margin against diffuser stall/separation effects as illustrated in Figure 2.2.4-2. Diffuser performance metrics include the pressure recovery, C_p

$$C_p = \frac{P_e - P_i}{\frac{1}{2} \rho \bar{u}_i^2} \quad (\text{incompressible flows})$$

$$C_p = \frac{P_e - P_i}{P_{oi} - P_i} \quad (\text{compressible flows})$$

$$C_{pi} = 1 - \frac{1}{AR^2} \quad (\text{based on Bernoulli's equation})$$

The maximum C_p achievable with modest inflow blockage as shown in Figure 2.2.4-3 and Figure 2.2.4-4 is approximately 0.65. Another required parameter is the velocity ratio, $\bar{u}_e / \bar{u}_i$, (de Haller Number):

- Typically not less than 0.72 for compressor cascades
- Lower values common in compressor outlet diffusers

The diffuser effectiveness is defined as

$$\eta = \frac{C_p}{C_{pi}},$$

While the loss coefficient, ϖ or in some texts denoted as ζ is defined as

$$\varpi \equiv \frac{P_{oi} - P_{oe}}{\frac{1}{2} \rho U_i^2} = \frac{\bar{u}_i^2 - \bar{u}_e^2}{U_i^2} - \eta C_{pi}.$$

The total pressure loss is defined using the loss coefficient as

$$dP_o = \frac{1}{2} \rho U_i^2 \varpi$$

Using the 1 degree and 5 degree equivalent diffuser angle geometry, the calculated parameters were:

- $C_p \sim 0.535$
- $\bar{u}_e / \bar{u}_i \sim 0.655$
- $\eta \sim 0.88 - 0.92$
- $\omega \sim 0.021$
- $dP_o \sim 0.994$

Thus the anticipated total pressure loss of the rotor diffuser is on the order of 1 percent.

Figure 2.2.4-1: Representation of Rampressor rotor diffuser as a conventional 2-D planar geometry for analysis

Figure 2.2.4-2: 2-D planar diffuser flow regimes

Figure 2.2.4-3: 2-D Planar diffuser performance characteristics

Figure 2.2.4-4: The limits of achievable C_p 's illustrated by 2-D diffuser flow regimes

Figure 2.2.4-5: Range of anticipated diffuser effectiveness for planar 2-D diffusers.

2.2.4.5 Rotating Exhaust

The rotating exhaust of the rotor consists of the section between the exit of the rotating diffuser and the outflow of the rotor. This section has a constant rim radius that is the same as the exit of the rotor diffuser. No particular analysis was used to design this component due to the constant annulus geometry. The only design feature of note is

that the constant rim radius portion of the rotor was extended downstream of the strake trailing edge in order to provide room for rotor flow path modifications which might require additional rotor axial length, thus all Rampressor 1 rotors have an extra long section of rotating exhaust.

2.2.4.6 Starting Analysis

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2.2.4.7 3-D Inviscid

The ideal 2-D profile of the compression ramp and subsonic diffuser sections was assembled into a basic rotor model to define the 3-D flow path. The compression ramp plus 1 degree and 5 degree subsonic diffuser profile for the on-design rotor is shown in Figure 2.2.4-6 and is denoted as “FP4”. The 3-D rotor flow path is shown in Figure 2.2.4-11. The 3-D compression ramp and diffuser geometries were generated by placing this profile on a helical flow path that is orthogonal to the strakes – i.e., the compression ramp geometry is a function of angle from the leading edge as well as axial distance from the leading edge. This produced compression ramp and diffuser surfaces that were aligned relative to the strakes and thus were intended to produce the 2-D shock structure between the strakes.

This 3-D geometry was then simulated via 3-D inviscid CFD to determine if the basic design worked well with 3-D effects. Adapco, maker of STAR-CD, was employed to run these 3-D inviscid simulations with a grid and boundary conditions set-up that was similar to the pre-inlet only simulations, except that the entire flow path was simulated as shown in Figure 2.2.4-12. Tip clearance was not simulated due to the additional grid requirements and the inviscid assumption of the simulation.

The flow path was initially solved with low back pressures to make sure the shock structure and flow rates were as desired. Five compression ramp geometries were simulated at low back pressures.

Ramps #1 and #2 were preliminary Mach 1.6 designs used to get an initial grid and CFD solution operating. Rotor Ramp #1 (see Figure 2.2.4-7) was designed using 2-D techniques, was over-contracted, and would not start numerically. The compression and diffusion surfaces had constant radii with axial position (non-helical surfaces) and the strake leading edge was swept back and the transition from the thin leading edge to the full strake thickness occurred on the compression ramp side of the strake (as opposed to the pre-inlet side). The geometry for Ramp #2 was arbitrarily reduced in radial value such that it would be easy to start numerically. The difference in radial profile is shown in Figure 2.2.4-8.

Ramp #3 strake geometry was identical to #2 except that the minimum area (throat) was moved upstream in the flow path (same radius however) in an attempt to have the shock reflection from the case wall intersect it as well as the rim compress and diffusion surfaces were made to be helical – i.e., the line of the start of the compression ramp and the throat, etc. were perpendicular to the strakes instead of being axial.

Ramp #4 had the same compression ramp geometry as #3, but the leading edge was radial (not swept back) and the transition to the full strake width from the thin leading edge was still on the compression ramp side. The solutions of Ramp #3 and #4 were compared to determine the effect of the sweep angle of the overall flow. It was decided to stay with the radial strake leading edge due to high pressure generated at the case wall where the strake meets the case.

Ramp #5 had the same radial leading edge as #4, but the cut back was on the pre-inlet side of the strake. In addition, the ramp and diffuser geometries were based on 2-D inviscid CFD simulations that produced compression and diffusion surfaces that had radius vs. angle (along a helical path parallel to the strakes) that had constant values – i.e., the compression ramps and diffuser rim were not a straight line, but rather curves defined by a constant rate of change of radius with angle. This combination of leading edge and compression/diffusion geometry seemed to give the closest flow field to the ideal 2-D flow.

Ramp configuration #5 was selected to perform back pressured simulations after evaluation of the shock structure on the compression ramp. The strake leading edge cut back placed on the pre-inlet side produced the best compression ramp shock structure, with the drawback being a higher strength leading edge shock that impinges on the pre-swirl nozzles (in the case of the CFD the inflow boundary condition).

The exit boundary static pressure on the Ramp #5 configuration was gradually increased starting at 75% of the theoretical value and achieving a value of 92% of the theoretical value. Above this value the normal shock would

push forward past the throat and the solution would unstart. The resulting static pressure distribution from the 92% case is shown Figure 2.2.4-13 and Figure 2.2.4-14 with the pressure scale compressed to highlight the lower pressure values at the inflow and compression ramp sections. Note the very good uniformity of the compression ramp static pressure distribution – i.e., the pressure variations run nearly orthogonal to the strake surfaces as desired. The leading edge does cause a disturbance on the compression ramp, but this is much less than the shock created by the leading edge on the upstream/pre-inlet side of the leading edge. In addition, Figure 2.2.4-13 shows the oblique shock waves reflect from the case wall and hit the rim at the throat/minimum area as desired.

Figure 2.2.4-6: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.2.4-7: Configuration for “Ramp #1” which included a raked back leading edge and shortened compression ramp. (NOTE: Strake diameters are shown larger in these models than in the actual design for ease in producing a CFD grid).

Figure 2.2.4-8: Profile differences between Ramp #1 and Ramp #2.

Figure 2.2.4-9: Configuration for “Ramp #3” with a swept back leading edge and the leading edge cut back on the compression ramp side of the strake. (NOTE: Strake diameters are shown larger in these models than in the actual design for ease in producing a CFD grid)

Figure 2.2.4-10: Configuration for “Ramp #4” with a radial leading edge and the leading edge cut back on the compression ramp side of the strake.

Figure 2.2.4-11: Solid model of Rampressor-1 flow path

Figure 2.2.4-12: Grid and boundary condition types used for 3-D inviscid simulation of rotor flow path.

Figure 2.2.4-13: Case wall static pressure and section cut through flow path of back pressured Mach 1.6 inviscid solution.

Figure 2.2.4-14: Computational domain boundary static pressure distribution of the Mach 1.6 3-D inviscid simulation of the rotor flow path.

2.2.4.8 3-D Viscous

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2.2.4.9 Final Rampressor-1 Flow Path Design

The final flow path design to be initially tested was but one flow path design used on a multiple-rotor testing strategy. It was known that the first rotor with all the flow path features desired for initial testing would not be available when all the static HW, peripheral systems, and test facility were ready. Thus a 6 rotor strategy was formulated to reduce risk and shorten the time to get useful data from the rotor with the full aerodynamic design. The multiple rotor strategy consisted of rotor geometries that add major flow path features in a stepwise fashion with the 6 rotors being denoted by their flow path (FP) features, thus being denoted as FP1 through FP5.

The FP1 and FP2 designs are illustrated in Figure 2.2.4-15 while the rim radius profile from the compression ramp to the end of the rotor diffuser is shown in Figure 2.2.4-16 for FP2, FP3, and FP4. The final FP4 full design is shown in Figure 2.2.4-17 with all the relevant features labeled. After initial testing with the FP1 and FP2 rotors (PSN characterization and system de-bugging) a testing strategy starting with assumed availability of FP on July 1 was implemented. This is shown in Figure 2.2.4-18 and represented a possible strategy first based on if FP4 could be aerodynamically started (supersonic flow past the throat). If it could not be started FP3 would be implemented while FP4a would be modified to correct any design issues. FP5 testing would happen as time permits, either due to

successful FP4 testing, or as an interim test plan while FP4a was being modified. By the end of the program only FP1, FP2, and FP4 were tested, with FP3 being in a partially machined state and FP4a and FP5 being dropped due to lack of funds.

Figure 2.2.4-15: Illustration of FP1 (a) and FP2 (b) rotor designs

Figure 2.2.4-16: This figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.2.4-17: This figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

FP1: No Features
FP2: Strakes Only
FP3: Easier To Start
FP4: Best Design
FP4a: 2 wk Turn-Around
FP5: Rim/Throat Slots

Figure 2.2.4-18: Potential test strategy subsequent to FP1/FP2 initial testing.

2.2.5 Static Core Test Rig

2.2.5.1 General Layout

The primary goal of the core rig static hardware was to provide a robust, flexible, low risk mechanical system with which to perform aero experiments. Thus the design incorporated the following attributes:

- Static hardware easily manufactured from readily available materials
- “off-the-shelf” components used where possible
- Rig easy to assemble/disassemble
- Ease of instrumentation access
- Anticipate and provide for “worst case” aero requirements
- Provide fast turn-around on disk flow path configuration

The general layout of the core rig is shown in cross section in Figure 2.2.5-1 with a 3-D cut-away shown in Figure 2.2.5-3 and additional features shown in Figure 2.2.5-4. The requirement of having high concentricity between the bearing journals, tip ring, and seal diameters drove the design to have critical radial and axial surfaces on the assembly as illustrated in Figure 2.2.5-6. The bearing bores were line-bored as an assembly to ensure straightness and concentricity. Due to the over-hung bearing design and the replaceable pre-swirl nozzle assembly, the load path between bearings was quite long, as shown in Figure 2.2.5-7, thus the assembly required significant stiffening. The bearings were pre-loaded using a spring system that was compressed via a fine thread bolt. Details of the bearing components are shown in Figure 2.2.5-8 (aft bearing shown, fwd bearing components similar). The core rig secondary flow systems are annotated in Figure 2.2.5-9 and show the bearing purge and wheel space purge flow paths.

2.2.5.2 Analysis

A thermal and structural finite element analysis was performed on the static core rig using ANSYS. The analyses results helped predict the deflections of various components of the static structure. The structure deflections were used for determining the hot to cold conversion, as well as to identify any interferences between different components. In addition, these analyses helped in identifying the load path of the test rig. Further more, these analyses helped in predicting the rig stiffness that was then used in the rotor dynamics analysis to predict the modes of the overall system. The boundary conditions for these analysis were based on the aero conditions predicted by the mean line design tool at a rotor inlet relative Mach Number of 1.6. Figure 2.2.5-11 through Figure 2.2.5-13 shows the results of these various analyses. Figure 2.2.5-11 shows the finite element model of the static structure. Figure 2.2.5-12 shows the predicted temperature distribution of the static structure at design point and Figure 2.2.5-13 shows the predicted deflections of the static structure due to thermal and pressure loads

Figure 2.2.5-1: Overall layout of Rampressor-1 test rig.

Figure 2.2.5-2: Detailed view of the annular exhaust valve ring and actuator arm.

Figure 2.2.5-3: Section of the core test rig assembly showing the 3-D view of the components.

Figure 2.2.5-4: Details of the core test rig forward housing weldment/assembly.

Figure 2.2.5-5: Drawing of the full cross section of the Rampressor-1 core test rig.

Figure 2.2.5-6: Illustration of precision radial and axial surfaces required to obtain the desired concentricity between the rotor and tip ring.

Figure 2.2.5-7: Illustration of the load path between bearings.

Figure 2.2.5-8: Detailed view of bearing layout and components (aft bearing)

Figure 2.2.5-9: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.2.5-10: Details of the thermally-actuated tip ring system.

Figure 2.2.5-11: Finite Element Model of the Rampressor 1 Static Core Rig

Figure 2.2.5-12: Predicted full load steady state Temperature distribution of the static structure

Figure 2.2.5-13: Predicted full load steady state deflections of the Rampressor 1 static core rig.

2.2.6 Core Rig Instrumentation

Mechanical health and aerodynamic performance instrumentation was incorporated into the Rampressor hardware design. The mechanical health instrumentation was required to ensure safe rotational operation and functionality of critical operating systems. The primary focus of the health monitoring was on the proper operation of the bearings. Bearing heat release and loading was monitored with a variety of methods. Oil supply pressure and temperature, oil out-flow pressure and temperature, buffer seal supply pressure, oil out-flow cavity pressures, and bearing race temperatures were monitored to ensure proper heat extraction from the rig and drive turbine bearings, Figure 2.2.6-1. Imbalance and bearing loads were monitored by tri-axially mounted velocimeters on the forward and aft rig housings, by a thrust load cell behind the aft bearing and by cavity static pressures. The summation of the cavity pressure is used to derive forward and aft side, rotor forces. Measurements were also taken to ensure proper operation of the throttling valve, the thrust balance system and the tip cooling system. Static pressures on the flow path and collector sides of the throttle valve were monitored to verify circumferentially uniform backpressure and flow, Figure 2.2.6-2. An encoder output the location of the throttle valve for comparison to its commanded position. Static temperature and pressure were monitored in wheel space cavities on the forward and aft sides of the rotor as part of the thrust balance and air buffering systems, Figure 2.2.6-3. Operation of the thermally controlled, tip clearance system was monitored with supply air, static temperatures and pressures, exhaust cavity pressures, and tip ring metal temperatures, Figure 2.2.6-4.

Figure 2.2.6-1: Lubrication System Pressures and Temperatures

Figure 2.2.6-2: Exhaust Section Static Pressures

Figure 2.2.6-3: Wheel Space Pressures and Temperatures

Figure 2.2.6-4: Tip Ring Metal Temperatures

Accurate measurement of aerodynamic performance parameters was critical for characterizing the rotor operation. Primary factors were the rotor inflow and exit conditions and the interaction of the rotating and static portions of the rotor flow path. Inflow conditions to the rotor were measured by static pressures and temperatures in the upstream plenum, static pressures through the pre-swirl nozzle blades, pre-swirl nozzle exit total and static pressures, entrance duct static pressures, and rotor entrance static pressures, Figure 2.2.6-5 through Figure 2.2.6-8. Inlet mass flow was measured with a critical flow venturi in the supply line. Rotor exit conditions were measured by a rotor exit total pressure and temperature probe, a flow angle probe, exit static pressures, and collector static temperatures. Rotor exit flow was measured with an ASME nozzle. Additional downstream exhaust static pressures existed for exhaust duct and throttle valve loss characterization. The most difficult measurements involved determination of flow conditions on the rotor. Sensors mounted in the stationary tip ring measured low and high frequency static pressure on the tip ring surface, tip ring metal temperature and tip clearance, Figure 2.2.6-9

and Figure 2.2.6-10. Operational speed was derived from rotor speed, as determined by drive turbine shaft speed and by coupling speed. Tooth detection on a six tooth gear yielded drive speed and a once per revolution pick-up on the drive to rig coupling provided redundant speed measurement and rotor feature location.

Figure 2.2.6-5: Pre-swirl Nozzle Static Pressures

Figure 2.2.6-6: Rotor Entrance Duct Static Pressures

Figure 2.2.6-7: Rotor Static Pressures

Exit Rake/Survey Plane Hub Statics 4-
Between Rakes
Exit Rake/Survey Plane Tip Statics 4

Figure 2.2.6-8: Rake and Probe Static Pressures

Figure 2.2.6-9: Rotor High Frequency Pressure Transducers

Figure 2.2.6-10: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.2.6.1 PCBs

The primary diagnostic tool for evaluating fluid condition on the rotor was the piezoelectric pressure transducer. Rotor relative measurements are difficult and expensive. A slip ring or telemetry system would have been required to ascertain conditions on the rotor hub. In lieu of these systems, high frequency pressure oscillations were observed on the tip shroud. Under ideal conditions, the coalescing shock system would produce well-defined static pressure regions on the stationary tip ring which travel with the rotor. The sensors measured the static pressure variation across these standing waves. Two types of sensors were mounted in the tip ring for this purpose. The sensors required resonant frequencies > 1 MHz. The sensing face diameter was smaller than the stroke width to avoid

creating a flow path over the strake. And the sensors operated over a wide temperature range, -100 to 350 degrees Fahrenheit. To avoid damage, the sensors were recessed 0.035 in. from the tip shroud, inner diameter. PCB 132A31 and PCB 115M18 sensors were employed. The 132A31 sensors were a time-of arrival sensor intended to locate strong shocks. The 115M18 sensors were slower decay sensors intended to determine the absolute magnitude of the pressure profile. The 132A31 probes were located at three circumferential locations with 120 degree symmetry. Axially the probes were positioned over the center of the throat. 115M18 probe were positioned at one circumferential location. Axially the probes resided over the throat quarter points. All probes were sampled at 1 MHz to resolve the shock location to ± 0.032 in. A once per revolution slot on the rotor served as a rotor position indicator and as the data acquisition trigger.

2.2.6.2 Performance Rakes

Exhaust duct surveys were performed to determine rotor exit total pressure, total temperature and flow angle profiles. Three types of probes were used. Initial surveys were performed using a single point, cobra probe to ascertain flow angle and total pressure profiles. Utilizing the cobra probe derived flow angle, the position was fixed for two 5 immersion, total pressure rakes. The total pressure rakes offered gross, continuous monitoring of the exit profile. Selectively, refined pressure and temperature profile data was collected with a performance probe. The performance probe was a single immersion, dual element probe. A 0.020 total pressure tube and a bare bead, K-type thermocouple were centered in a Kiel head, giving the probe pitch and yaw insensitivity to ± 30 degrees. Total pressure rake immersions also had Kiel heads. The performance probe and cobra probe were positioned using a Rotodata MS0835 actuator. The actuator allowed radial travel and rotation about the radial travel axis. The actuated, performance or cobra probe was positioned, in 120 deg symmetry with two total pressure rakes, at a survey plane located 0.6 duct heights downstream of the rotor exit plane. 4 tip and 4 hub, exhaust duct, static pressures were sampled in the rake and probe plane for calculation of exit Mach number profile.

Characterization of the pre-swirl nozzle was also performed using cobra probe surveys. Pre-swirl nozzle exit total pressure and flow angle profiles were used with exit duct static pressures and supply plenum static pressures to establish the non-dimensional, nozzle characterization term, omega-bar. $\text{Omega-bar} = (P_{t_{in}} - P_{t_{exit}}) / (P_{t_{exit}} - P_{s_{exit}})$. This term was used during testing, with and without static pressure reference, to determine the rotor inflow total pressure.

2.2.6.3 Key Pressure and Temps

Selection of temperature and pressure sensor type and installation is important for proper performance determination. Critical static temperatures in the inflow plenum and the exhaust collector were evaluated using platinum, resistance temperature detectors (RTDs), having a theoretical accuracy of ± 1.4 degrees Fahrenheit. Field calibration of these devices demonstrated an accuracy of ± 0.25 degrees Fahrenheit. Static temperature measurements were taken in low Mach regions, Mach 0.007 inflow plenum and Mach 0.19 exhaust collector. All other temperature measurements were collected with K-type thermocouples, having a accuracy of ± 3 degrees Fahrenheit.

Static pressure measurements in the flow path were fabricated as 0.020 sharp edge holes. The holes were wire plunged into the base material without surrounding material inconsistencies. Static pressure measurements taken below the flow path, in the wheel spaces and vacuum plenums, utilized thin wall installation. 0.040 instrument tube was installed through the component wall and polished flush with the surface. Pressure lines were routed to the exterior of the rig and sampled using the facility's Scanivalve Hyscan system. 0-15 and 0-50 psi modules were utilized for inflow and exhaust conditions respectively. Both modules have accuracies of $\pm 0.25\%$ full scale. Modules are calibrated prior to each half day test session.

Proper connection of pressure and temperature instrumentation was verified prior to testing. Static pressure sampling connections were bubble checked during initial installation. Thermocouple function was verified by ambient condition readings at initial connection. Critical thermocouple connects were checked with a heat gun to verify the absence of bi-metal junctions. Test level verification of instrument functionality was verified with a pressure lock-up test. The throttle valve was closed with only wheel space purge air on. The rig was allowed to pressurize to 17 psia. Pressure measurements in the rig were compared to the 17 psia nominal.

Proper function of instrumentation was verified at regular intervals. Calibration of the facility's Hyscan pressure system and thermocouple data acquisition system was performed prior to each half day test session to ensure the integrity of performance numbers. RTDs used to measure inflow and exhaust plenum temperatures were field calibrated twice during the test program. A two point calibration was utilized. The boiling and ice bath

temperatures of distilled water were compared to atmospherically corrected theoretical values. ± 0.25 degree Fahrenheit accuracy was demonstrated, Table 2.2.6.3-1. Total pressure cobra and performance probes were calibrated in Boeing's flow facility. The calibration validated probe recovery for 0.05 Mach and 0.1 Mach number increments from 0.1-0.9 Mach. All standalone, mechanical health transducers are subject to yearly calibration and were calibrated prior to testing.

Inflow and Exhaust RTD Calibration

Glass beaker of distilled water held at boiling point. Scans taken at 1 minute intervals

PREF, psia14.925

From "Thermodynamics an Engineering Approach", Cengel and Boles

Pres, psia14.696Tsat, degF211.99slope0.29231

15213.03Tsat at 14.925213.008

Tsat degR672.678

Scan #													
Ramgen Tagg	Boeing Tagg	Name	1	2	3	4	5	6	7	8	Tavg, deg	Tavg, deg	stnd dev
COL01MTi	T5401	Exhaust collector RTD	662.272	662.937	664.414	665.104	665.817	666.211	666.704	667.147	665.076		1.75879
COL02MTi	T5402	Exhaust collector RTD	672.142	672.340	672.266	672.364	672.414	672.290	672.340	672.315	672.309	672.309	0.08116
COL03MTi	T5403	Exhaust collector RTD	672.276	672.473	672.424	672.473	672.498	672.424	672.399	672.448	672.427	672.427	0.06899
COL04MTi	T5404	Exhaust collector RTD	672.334	672.507	672.457	672.507	672.531	672.432	672.482	672.482	672.467	672.467	0.06183
		Tavg, degR	669.756	670.064	670.390	670.612	670.815	670.839	670.981	671.098	670.570	672.401	
SID01MTi	T5501	Inlet RTD	672.393	672.615	672.541	672.639	672.664	672.566	672.541	672.566	672.566		0.08325
SID02MTi	T5502	Inlet RTD	672.461	672.683	672.658	672.733	672.782	672.683	672.634	672.683	672.665		0.09404
SID03MTi	T5503	Inlet RTD	672.494	672.691	672.691	672.716	672.716	672.666	672.593	672.642	672.651		0.07556
SID04MTi	T5504	Inlet RTD	672.406	672.579	672.554	672.579	672.629	672.530	672.530	672.554	672.545		0.0647
		Tavg, degR	672.439	672.642	672.611	672.667	672.698	672.611	672.575	672.611	672.607		
SED71MTi	T5170	Performance Probe TC	673.003	672.925	673.042	672.887	673.059	672.989	672.805	673.042	672.969		

Glass beaker of distilled water and ice, stirred continously during data collection. Scans taken 1 minute apart.

Scan #													
Ramgen Tagg	Boeing Tagg	Name	1	2	3	4	5	Tavg	stnd dev				
COL01MTi	T5401	Exhaust collector RTD	491.527	491.527	491.527	491.503	491.527	491.522	0.01073				
COL02MTi	T5402	Exhaust collector RTD	491.715	491.740	491.740	491.740	491.764	491.74	0.01733				
COL03MTi	T5403	Exhaust collector RTD	491.733	491.757	491.733	491.733	491.757	491.743	0.01315				
COL04MTi	T5404	Exhaust collector RTD	491.749	491.749	491.749	491.749	491.772	491.754	0.01029				
		Tavg	491.681	491.693	491.687	491.681	491.705	491.69					
SID01MTi	T5501	Inlet RTD	491.709	491.757	491.733	491.733	491.757	491.738	0.02008				
SID02MTi	T5502	Inlet RTD	491.759	491.759	491.759	491.759	491.759	491.759	0				
SID03MTi	T5503	Inlet RTD	491.713	491.737	491.713	491.713	491.737	491.723	0.01315				
SID04MTi	T5504	Inlet RTD	491.782	491.782	491.782	491.782	491.806	491.787	0.01073				
		Tavg	491.741	491.759	491.747	491.747	491.765	491.752					
SED71MTi	T5170	Performance Probe TC	491.582	491.434	491.657	491.454	491.355	491.496					

Table 2.2.6-1: RTD Calibration Data

2.2.7 Tip Clearance Control Ring

This section discusses the tip clearance control system used in Rampressor 1 test rig. The requirement for tip clearance was to maintain a 0.001 inch radial clearance at full load design point. The system used in the Rampressor 1 test rig was an impingement cooled thermally actuated tip clearance system. One of the key factors for achieving the requisite tip clearance levels was the material for the tip shroud ring. The analyses/procedures used to determine the material for the tip shroud ring is discussed here. Since the inlet temperature was expected to vary anywhere between 50 °F to 90 °F, the material that is chosen for minimum tip clearance at design point must be capable of holding minimum clearances at different inlet temperatures, at design point operation. In order to meet this objective, six different materials were analyzed by considering different inlet temperatures and different cooling air temperatures. The recommended material for meeting the tip clearance objective is Aluminum – 6061 – T6. The cooling air requirement for meeting the design point tip clearance objective is 1.5 lbm/s at 31 psia and –50 °F, in the impingement cavity (the cavity that is immediately upstream of the impingement plate). To address the off-design points, it is recommended that the cooling air temperature requirement be supplied at – 50 °F (based on extrapolation of the results at design point), to provide additional margin. Further details of the tip clearance system design are provided in the Chapter 2 Appendix.

2.2.8 Bearings

This section discusses the bearing system analysis and the rotor dynamics analysis that was performed for the Rampressor 1 test rig. The bearing system design required a minimum life of 100 hours at 21000 rpm (typical for

test rig operation) and the rotor dynamics analysis requirement was to ensure that the critical speed of the system is away from the design speed operation. This requirement ensured that the natural frequencies of the system were away from the design speed operation. The bearing system that was chosen for this purpose was a pair of NHBB 52100 steel 1108 size with a 15 degree contact angle. The bearing system had a preload spring in the forward side to account for thrust changes due to main flow path supersonic inlet un-start events. Furthermore, the bearing system was complemented by a thrust system (using pressures across the disk) to ensure that the bearings were always loaded. A detailed report of the bearing analysis is attached in the Chapter 2 Appendix

2.2.9 Internal Bleed Systems

2.2.9.1 Rotor Bleed

This section intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.2.9.2 Pre Swirl Nozzle (PSN) Bleed

There was a requirement for bleeding air immediately downstream of the PSN to provide a clean flow to the Rampressor rotor. The bleed flow was routed through three 0.25 inch pipes in the forward static structure to the peripheral system. However, during testing, it was determined that this bleed had very little effect on improving the aerodynamic efficiency of the Rampressor rotor. In the Appendix is included a report discussing the pre swirl nozzle bleed design.

Table 2.2.9-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Table 2.2.9-2: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.2.9-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.2.9-2: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.2.9-3: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.2.10 Monolithic Rotor Mechanical Design

The design speed for the monolithic rotor was 21000 rpm. At this speed, the rotor had to meet the low and high cycle fatigue requirements. The burst requirements for the disk were established as 125% of the maximum permissible speed. For the low cycle fatigue requirement, the von Mises stress in the rotor needs to be less than 80% of 0.2% yield strength and the principal stresses had to be less than 80% of ultimate tensile strength. The material used is Titanium 6242. The reason for this is because of strength and weight requirements. Furthermore, this is an easy material to machine and resists corrosion and erosion. For Ti 6242, the principal design stress limit was established as 100 ksi. Since there is strong vibratory excitement of the strake, the design for alternating stresses has to be less than 10ksi. Finally the burst margin of the disk establishes that the average disk tangential and radial stress have a safe margin on disk hub burst.

2.2.10.1 Stress Analysis

2D as well as 3D finite element analysis were performed on the rotor to ensure that the design met the above mentioned criteria. The 2D analysis was used for the disk and the 3D analyses were primarily used in for the rotor section that concentrates on the main flow path. All the analyses were performed using ANSYS. The models typically had about 800,000 elements and 1.2 million nodes. Extensive use of sub modeling was done to obtain accuracy in critical areas. A parametric model of the disk was set up using APDL in ANSYS. An optimization run using the APDL scripts performs the analysis, modifies the design based on the results of the analysis within the constraints imposed and carries out this loop till all the design criteria is met. Some of the analysis results were

- Average tangential stress in the disk was 14.1 ksi

- Average radial stress was 18 ksi
- Tangential burst margin was 2.7 and radial burst margin was 2.6
- Over speed was possible up to 26000 rpm. This speed was limited by the E-beam weld properties between the bleed plate and the rotor.

2.2.10.2 Heat Transfer Analysis

The thermal analysis was conducted using ANSYS. The boundary conditions were generated using the aero mean line predictions in conjunction with the results from the CFD analysis. A transient as well as a steady state analysis was performed to predict the rotor growth at various speeds. This was important in order to set the requisite tip clearance levels. Furthermore, the results from the thermal analysis were used in the structural analysis to predict the life cycle at the running temperatures. The model used for thermal analysis is shown in Figure 2.2.10-1. The predicted full load steady state temperature distribution is shown in Figure 2.2.10-2. The corresponding structural displacements at these various locations are shown in Figure 2.2.10-3. The transient radial growth at some of the critical locations (critical locations are specified in Figure 2.2.10-1) in the rotor is captured in Figure 2.2.10-4.

2.2.10.3 Rotordynamics

The rotor dynamics analysis was conducted to determine if the principal modes of the system were away from the design speed. It was also used to determine the maximum imbalance load that the system can sustain without causing vibration issues. The analysis included whirl speed of the entire system with damped and undamped critical speed analysis. A detailed report of the rotor dynamics analysis is attached in the Chapter 2 Appendix.

Figure 2.2.10-1: Rotor Model used for predicting thermal distribution and structural displacements.

Figure 2.2.10-2: Predicted full load steady state temperature distribution for the Rampressor 1 rotor

Figure 2.2.10-3: Predicted full load steady state displacements for the Rampressor 1 rotor

Figure 2.2.10-4: Predicted transient displacements for the Rampressor 1 rotor at critical locations.

2.2.11 Design Operating Conditions

The final predicted design operating conditions based on the MAC8 version of the mean line design code were:

- Air Supply Conditions
  - Total Pressure: 22.05 psia
  - Total Temperature: 50° Fahrenheit
- RPM
  - Physical Speed: 20,483 RPM
  - Corrected Speed: 20,656 RPM
- Flow Mach Numbers and Flow Angles
  - PSN Mean-Line Outflow: Mach 0.77 @ 73.29° from axial
  - Rotor-Relative Mean Line Pre-Inlet: 1.67
  - Rotor-Relative Mean Line Compression Ramp Inflow Mach: 1.6
  - Rotor Mean Line Outflow: Mach 0.27 @ -75.6° from axial (opposite direction from pre-swirl flow)
- Mass Flow Rates
  - Physical Inflow: 3.19 lbm/sec
  - Corrected Inflow: 2.11 lbm/sec

2.2.12 Peripherals

2.2.12.1 Drive

A Barbour-Stockwell radial inflow turbine was selected from a candidate group of pneumatic, steam and electric drives to power the Rampressor, Figure 2.2.12-1. The turbine's 1100 Hp and 50,000 rpm capability made it unit suitable for Rampressor experimentation as well as for future Ramgen projects. The turbine was fitted with a titanium rotor and ceramic hybrid bearings. Peak power output was designed for supply conditions of 10.5 lbm/sec air at 280 psig and 400 degrees Fahrenheit. A mist lubrication system was supplied with the unit. Bearing race TCs and drive rpm instrumentation was provided for monitoring drive health. The drive turbine has a stand alone controller with PC interface. The controller allows control using set speed or set valve position control. Drive operation can be tuned using operator supplied valve gain, acceleration and deceleration limits and over speed limits.

Figure 2.2.12-1: Barbour-Stockwell Turbine

2.2.12.2 Controls

Control of the Rampressor is performed primarily through safe operation of the drive turbine, oil supply and the thrust balance system. Within normal operating parameters, proper control of the bleed systems, the throttle valve and the tip clearance system can improve operational performance, but are not required for safe operation. All control resources are operable from the control room.

The most important control is the drive turbine air supply control. As a power consuming device the Rampressor is an inherently fail safe device. Loss of drive turbine supply air will result in deceleration of the rotor and reduction in stored energy. The drive turbine is controlled using a manufacturer supplied, control system. When provided with appropriate air resources the controller will allow the operator to input a target speed, acceleration rate, overspeed limit, and valve position gain. Direct valve position input was an unused, alternate control method. The drive speed feedback, used by the manufacturer's controller, was split for monitoring with Ramgen and facility control systems. Using this rpm signal, the facility was able to provide speed limitation through supply pressure control and rapid shutdown with a secondary, fast-acting valve in the drive line. Over pressure of the drive turbine supply line was prevented by system capacity and by a burst disk in the supply piping.

Health of the core rig was primarily monitored and controlled by Ramgen's control system. The Allen Bradley PLC5 system was responsible for operational pressure, temperature and thrust data acquisition and for thrust balance system and throttle, valve control. Thrust generated across the rotor was detected by a load cell mounted behind aft bearing. The thrust system relied on manual, valve position adjustment of back pressure valves on the forward and aft, wheel space cavity air exhaust legs. Cavity pressures were adjusted to maintain the indicated thrust of the load cell within the prescribed limits of the bearings. Throttle valve position was manually adjusted. Manual, fast-acting valves on the intake and exhaust plenums enable rapid depressurization of the rig. A 100 psig burst disk was supplied on the Rampressor air supply.

Additional operational control was performed by the facility. The facility provided manual control of the facility air system flows (4), drive air temperature, tip clearance air temperature, bleed system pressures (3), and survey actuator position and angle using its control systems. The plumbing diagram is shown in Figure 2.2.12-2.

Figure 2.2.12-2: Rampressor Plumbing Schematic

2.2.12.3 Data Acquisition

Responsibility for acquisition of instrument signals was the shared responsibility of Ramgen and the facility. Signals associated with rig health were collected by A/D cards in Ramgen's PLC. Signals associated with the bearings, the lube system, the thrust balance system, the tip clearance system, throttle valve pressures and the safety valves were sampled at 14 bit depth and 1 Hz continuously during testing. Four channels of the high frequency, over-rotor, pressure transducer signals were simultaneously acquired at 1 MHz on a stand alone personal computer. Acquisition of 7000 scans with 16 bit depth was operator initiated within a National Instruments LabView VI. 7000 scans represented 3 full rotor revolutions at full speed.

The facility provided data acquisition of the performance measurement parameters. Primary system signals included all Rampressor instrumentation on the facility's Hyscan Pressure system and K-Type TC models in addition to the facility PLC based control and fluid metering parameters and the Rotadata actuator output. All of the thermocouple measurements, 100 channels of pressure data and rough flow calculations were supplied to Ramgen at 0.2 Hz over a socket feed. Custom Agilent HPVee software received the signal and displayed raw data and calculated Rampressor performance parameters. Set-point data was also collected from the primary facility systems. At test director specified operating points, all facility data was sampled for 1000 scans and the average values were saved. After the test, set-point data was reduced. Thereafter all raw data and final flow calculations were transferred to Ramgen. The HPVee software then merged the set-point data with the 0.2 Hz data. In addition to its primary data acquisition system the facility also provided a four channel oscilloscope for acquisition of tip clearance sensor data. The clearance measurement system was sampled at 200 kHz or better and collected at 1 Hz. The

oscilloscope was interfaced to a stand-alone PC through a GPIB connection. Post test the clearance data was merged with the other facility data by the HPVVEE software.

Figure 2.2.12-3: Diagram of the Rampressor-1 data acquisition system.

2.2.12.4 Instrumentation

The facility provided a portion of the instrumentation used in the Rampressor test program. In addition to the pressure and temperature data acquisition systems, the facility provided mass flow metering and calculations for main air inflow and peripheral systems. All controlled legs of the facility air system were metered using critical flow venturis with stand alone pressure transducers. Lube system vent air, vacuum lines, and some wheel space purge flows were metered with AMSE nozzles and flow tubes. Total pressures for these systems were measured using the facility Hyscan pressure system and differential pressures were measured with stand alone transducers. Flow rates were calculated using Boeing proprietary reduction codes. These results were checked using ASME MFC-7M-1987 and MFC-3M-1989 standards.

The facility provided a laser alignment system to assist in pretest hardware alignment and a Charge Couple Device (CCD) imaging system to troubleshoot alignment during operation.

2.2.13 Lessons Learned

Improvements to several elements of the peripheral systems were identified during the course of the test program. The tip ring system should allow larger adjustment for eccentricity and have better incursion rate control. Elimination of the thermal element would also produce a more adiabatic system and consequently improve the performance measurements.

The thrust balance system should be designed to utilize a larger portion of the bearings thrust capability. Perturbations in the flow field resulted in thrust loads which initially exceeded the design operational limits. Development of a suitable control system or of a bearing scheme capable of carrying the full thrust load would improve operation.

Tip clearance measurement instrumentation should not share amplifiers or power supplies. Increased emphasis should be placed on sensor robustness. Provisions should be made to better measure cold build clearance. External access to the sensors should be provided for repairs during testing.

Variability or rapid change capability should be added to the PSN assembly. Blade stagger changes will likely be required to properly match pre-swirl nozzle and rotor mass capture.

The rotor proved insensitive to throttling location. Future rig builds could be simplified with throttling in the exhaust plumbing.

Air buffering should be used on the aft shaft seal. Abradable coated seal components should be easily removable for seal re-treatment.

Bearing race temperature measurement should be added. The measured oil temperature and bearing structure temperatures provided an indication of bearing operating condition, but lacked sufficient response time to serve as a preventive health tool. Vibration sensors provided the best operational health indication. These sensors should be added to the drive turbine.

2.3 Assembly

The Rampressor test rig was assembled at the Boeing test facility beginning in late May 2003. The forward assembly arrived from the vendor instrumented and partially assembled. The final assembly and instrument attachment was performed to support pre-swirl nozzle testing in early June. Final assembly and instrumentation of the aft housing was performed in parallel with pre-swirl nozzle testing. The first full rig assembly was performed to support late June testing with Rotor #1.

Installation and service of the Rampressor test article influenced the testing rate. A key feature of the test article was access to the rotor through the separation of a single fit between the forward and aft housing. This fit allowed quick access to the rotor, tip ring and bearings without large scale separation of the rig components. A limiting factor on test turn around was the volume of external service connections and alignment time required to ensure proper bearing operation. Figure 2.3.1-1 through Figure 2.3.1-6 depict the installation of the Rampressor.

2.3.1 Boeing Installation

Boeing Nozzle Test Facility (NTF) was selected as the location for Rampressor I testing because of its ability to meet the test requirements and provide margin for select conditions. The NTF offered four controllable, conditioned, air supply lines: primary, secondary, and 2 tertiary legs at 300 psig. The two tertiary lines had steam exchangers and the capability of operating at 1000 psig. Shop air was available at 100 psig. The facility had 4 controllable vacuum lines from a common plenum. For the Rampressor initial tests the primary air supply was used for the core rig flow and the secondary air supply used for the drive turbine. The two tertiary legs supplied the hot and cold legs of the tip ring conditioning system. The wheel space purge air was drawn from the shop air system so that it would be maintained in the event of a facility shutdown. Bleeds were routed to the facility vacuum system on independent legs. When the LN2 system was implemented in November the drive air supply line and the cold air supply leg of the tip ring conditioning system were interchanged. Facility installation photos can be found in Figure 2.3.1-1.

Installation into the facility began April 28th 2003. Peripheral systems and rig support hardware were integrated and tested throughout May. Assembly and checkout of Rig hardware took place in early June to support mid-month pre-swirl nozzle testing. The staged rig build continued until part speed spin testing began on June 25.

Figure 2.3.1-1: Test Cell Integration

Figure 2.3.1-2: Rampressor Inflow Plenum Assembly

Figure 2.3.1-3: Rampressor Assembly

Figure 2.3.1-4: Rampressor Installation (July 2003)

Figure 2.3.1-5: Rampressor Installation, Aft looking Forward (February 2004)

Figure 2.3.1-6: Rampressor Installation, Forward looking Aft (February 2004)

2.3.2 Rotors

Four monolithic rotors were fabricated for use during Rampressor testing: Rotor #1, a bald rotor; Rotor #2, a straked rotor without compression features; Rotor #3, a ready rotor; and Rotor #4 the final flow path rotor. Rotor #1 had complete, final geometry under the flow path. The outer diameter of the rotor was featureless and tapered to provide a smooth surface transition from the inflow to the exhaust duct height. The rotor was used for full speed mechanical systems check-out. The rotor was also required to prevent downstream separation during pre-swirl nozzle characterization. Rotor #4 was the compression test rotor. Rotor #3 was a partially machined rotor ready for quick turn-around modifications to the flow path. Initially this rotor was envisioned as a lower contraction ratio rotor which could replace Rotor #4 if starting difficulties were encountered. Rotor #3 was never used. The leading edge of all rotor's was chamfered 0.025 in. to allow for a manufacturing error in the hub diameter or the pre-swirl nozzle assembly.

2.3.3 Lessons Learned

During the Rampressor I test program, several hardware improvements were identified to increase test time. In general the rig should be as modular as possible. The requirement for access to most parts should be anticipated. However, emphasis should be placed on minimizing the number of connections required for access to the rotor and tip ring. The peripheral services should not need to be disconnected to reach the rotor. The rotor and tip ring require the most frequent inspection and refurbishment. Tight control of the bearing and its related fits is critical for proper running operation. Manufacture of these parts should be tightly control and rigorously inspected. To support the improve bearing fits the alignment should be as precise and as rigid as possible. Thermal variations in the support structure should be minimized. The drive and rig should be fixed in all six degrees of freedom, Allowing only radial growth. Movement of the fixed support structures should be minimized. Facility connections should be flexible to allow quick integration. As much rig pre-assembly and check-out as possible should be performed prior to arrival in the facility to maximize test time in the cell. Parallel test and assembly should continue to be leveraged.

2.4 Testing

Rampressor testing can be categorized into two categories:

- Pre-swirl nozzle (PSN) characterization tests, the purposes of these tests were to characterize the performance of the PSN to ensure it meets the rotor's inlet flow requirements and to measure the PSN total pressure loss so the rotor total pressure ratio and efficiency can be determined. These tests were static flow tests of the PSN at various inlet corrected flows. Radial surveys of the PSN exit conditions were made at various corrected flows so the mass average total pressure loss could be calculated and the profiles of flow angle and Mach number could be compared to design intent.
- Rotating component tests, the purposes of these tests were to measure a rotor(s) performance at various corrected speeds, throttle, and clearance. The emphasis of most of the testing was to discover the optimum speed, clearance, and throttle setting. FP4 the compression rotor was run with three different PSN configurations. The original PSN had 65 vanes and was located upstream of the rotor. The PSN solidity was adjusted (65 vanes to 72 vanes) to increase the PSN exit Mach number at a given inlet corrected flow and to see how sensitive FP4 performance was to PSN exit Mach number. The 72 vane PSN was also moved closer to the rotor (referred to as PSN Aft configuration) to see if FP4 performance changed dramatically with improved profiles of Mach number and total pressure.

Rampressor testing can also be broken down into several tests in each category

- PSN characterization tests
  - 1) 65 Vane PSN Fwd
  - 2) 72 Vane PSN Fwd
  - 3) 72 Vane PSN Aft
- Rotating component tests
  - 1) FP1
  - 2) FP2
  - 3) FP4 with 65 Vane PSN Fwd
  - 4) FP4 with 72 Vane PSN Fwd

- 5) FP4 with 72 Vane PSN Aft
- 6) FP4 with 72 Vane PSN Aft and Swept Leading Edge

2.4.1 Overall Testing Record & Schedule

2.4.1.1 By Month Record (Notable Events and Accomplishments)

April 2003

Ramgen began installation into Boeing's Nozzle Test Facility (NTF) April 28. Boeing's flow conditioner was positioned in the cell to provide the reference point for all Ramgen hardware. Ramgen's drive turbine was aligned to Boeing's diffuser.

May 2003

Installation of plumbing, support structure, instrumentation and controls wiring occurred during the first two weeks of May. Initial spin testing of the drive turbine was performed on May 12. Two days of shake down testing validated drive operation, but revealed speed control difficulty under no-load operation. Validation testing of the tip cooling system began May 15. Four days of component and system testing demonstrated that the vortex tubes did not perform per manufacturer's specifications and were inadequate for operation. Design of a replacement liquid nitrogen (LN2) based system was initiated. The instrumented forward housing was received May 22. Inspection and partial rig installation was performed culminating in initial pre-swirl nozzle (PSN) assembly tests on May 30. Flow of the PSN assembly without the downstream flow path resulted in separation of the flow upstream of the survey plane. Further PSN testing was postponed until the remaining rig components were available. The aft static hardware components were received during the final week of May.

June 2003

June 2 through June 18 the rig static components were inspected and assembled. The initial full rig installation began June 19. Rotor #1 was installed (bald rotor for mechanical system testing). Alignment, final instrument and fluid connection, and controls checkouts were performed to support PSN testing on June 24 and 25. Low speed testing of rotor #1 was performed on June 25. Vibrations were encountered briefly at 10k rpm. Spin testing was terminated for the remainder of the build. Lube system improvements were implemented. Higher mass flow rate PSN tests were started June 30.

July 2003

The PSN was tested from June 30 to July 7. Design physical flow, 3.25 lbm/sec and elevated physical flow, 7.25 and 8.0 lbm/sec, was explored in an effort to reach the PSN design corrected flow. A teardown was initiated July 8 to increase the throttle valve size. The throttle valve was resized to allow higher flows with the lower density exhaust air encountered in the absence of a compressing rotor. PSN testing was resumed July 15. Tank pressure and exhaust area were varied in an effort to achieve design corrected flow. July 15-22 surveys at the rotor entrance plane were performed to characterize the PSN. Survey data was gathered at 20, 30, 40 and 50 psia tank pressures. Survey data was replicated at 1.5 and 2.0 lbm/sec corrected flow with 25, 50, 75 and 100% full (tip and hub) and hub only PSN bleed. July 22 full speed operation was attempted. High vibrations encountered in June persisted and evidence of a forward bearing failure was discovered during post test inspection. The drive was decoupled and tested alone to 20,000 rpm. Before tear-down a capacity test was performed on the PSN assembly. Choked flow occurred at approximately 2.05 lbm/sec corrected flow. A tear down was initiated July 24 to replace the failed bearing and to correct all rotors for a PSN fabrication error resulting in a 0.016 in. forward facing step at the rotor entrance plane.

August 2003

Mechanical shake-down testing resumed August 7 with the reinstallation of rotor #1. 20,000 rpm was achieved and maintained for 10 min. Baseline torque was measured. August 7-8 the rig exhaust system capacity was characterized. During the course of exhaust valve characterization PSN surveys were performed at 1.75 and 1.94 lbm/sec corrected flow. On August 12 baseline vibration checks were performed at various speeds and flow rates. The drive control system settings were tuned. August 12-13 PSN assembly surveys were conducted at 0.5, 0.6, 0.7 and 0.77 Mach. August 14 the rig was torn down to install rotor #2 (straked rotor without flow path area change). During the tear down, the tip clearance hot and cold systems were statically tested. Windmill tests of rotor 2, performed August 21. Proper rotation of the rotor under forced aspiration was verified. Rotational speeds of 12,000

rpm were achieved with elevated tank pressures. During the tests, the throttle valve actuator failed to maintain valve position at elevated flow rates. The first powered flow tests were performed Aug. 22. 20,000 rpm operation with a supply pressure of 20 psia was achieved with operation of the tip heating, clearance monitoring and bleed systems. Further capacity checks were performed on rotor #2 with elevated tank pressures and exhaust area increases. 1.94 lbm/sec corrected flow was achieved. A tear-down began on Aug. 28 to install rotor #4 (full flow path rotor). A hydraulic actuator was installed in place of the original electric throttle valve actuator.

September 2003

Machining errors were found on the forward labyrinth seal of rotor #4. Tip clearance system was bench tested using the liquid nitrogen cooling system on September 5. Rotor #4 was reworked and installed for testing on Sept. 10. Clearance system development and rotor check out was performed during 80 and 90%, part speed testing of rotor #4. On Sept. 12 the tip ring retention system failed at 20,000 rpm. The tip ring was replaced and the retention system improved during a teardown from Sept. 15-22. During initial testing on Sept. 23 the drive line burst disk ruptured. An additional pressure relief valve was installed to support resumption of testing on October 1.

October 2003

On Oct. 1 drive stability problems were encountered at 20,000 rpm with elevated tank pressures, 45 psia. The test series continued with investigations into throttling and pre-swirl nozzle bleed effects. During testing on Oct. 6, the forward bearing of the torque meter failed. A drive no-load spin test, laser alignment, and vendor torque meter failure analysis were performed. Efforts were made to improve the drive train and operational alignment. A stiffer quill shaft was fabricated and lubrication provisions were added for both rig and drive spline interfaces. Testing resumed Oct. 16 with speed vs. throttle position tests. The drive slide was removed and replaced by fixed mounts for the drive. Testing continued through Oct. 31 with strong emphasis on rig and drive movement during testing. Influences of tip ring air supply temperature and rig plenum pressure were explored.

November 2003

Throttled testing continued in November. A constant tank pressure of 20 psia was set and throttle and clearance adjustments were made. 107% speed was determined to be the highest performance operating speed. Since full throttling was not possible with the designed valve, additional throttling was performed with restriction plates in the exhaust metering pipe run. It was discovered that the pressure build circuits in the bottled LN2 were incapable of supporting the required flow rate. Based on these early Nov. tests action was taken to expand the LN2 system capacity and to improve the deflection measurement through the use of Boeing's CCD camera system. The CCD camera system was first utilized on Nov. 13. Tank pressure and tip clearance air supply temperature effects were investigated for comparison with initial indicator readings. Both test sets supported the observations of previous indicator tests, rig movement during operation was 0.025 in or better. As a result of this deflection testing, a flexible coupling was installed to isolate Boeing's test diffuser and the Rampressor rig from the supply piping and the design for a rig table with trunion mounting for the rig was initiated. On Nov. 24-25 a tanker supplied LN2 line was installed, replacing the previously used, LN2 bottle manifold. Testing resumed Nov. 26 with 107% speed line data.

December 2003

Continued characterization of the 107% speed line. During testing on Dec. 2 the forward bearing failed. A teardown was initiated to repair and bearing and install a new tip ring. Drive turbine in-flow and out-flow temperature instrumentation was added to determine the mechanical work of the drive in the absence of a torque meter. Slower decay rate, over-rotor, differential pressure transducers were installed. Test resumed Dec. 17 with continued testing at 107% speed. Tests Dec. 18-22 with clearance and speed sensitivity. Normally aspirated tests were performed on Dec. 23.

January 2004

Testing resumed Jan. 7 with a series of exit surveys at previously defined best efficiency points. First performance probe (Pt and Tt) surveys. The highest rotor relative Mach number achieved is 1.48. The performance probe survey results demonstrated the error induced in the collector based, energy balance efficiency by the tip ring cooling. In order to increase the exit Mach number, a tear down is begun to increase the solidity of the PSN assembly. Blade count is changed from 65 to 72. Re-machining of the PSN blades, reinstallation and re-instrumentation consumes the remainder of the month. The case was modified to incorporate a PSN survey location between the PSN assembly and the rotor. Drive air was replumbed to accommodate an external facility system user. A butterfly valve was used in place of the restrictor plates to enable the full throttle range.

February 2004

Characterization of the 72 blade PSN is conducted Feb. 2-4 with rotor #1. Rotor #4 is reinstalled with ceramic hybrid ball bearings. Rig check out testing began Feb. 9. During 80% speed rig check-out Feb. 10, the outer race of the ceramic hybrid bearings expanded to fix its location in the journal. No load transfer to the trust sensor occurred under loads below 500 lbf. A thrust event similar to that of previous bearing failures occurred, prompting a teardown. The lube system was inspected and the rig was reassembled with new bearings for testing on Feb. 12. Thrust transfer problems persisted. Feb. 12-20 testing of 72 ct. PSN assembly with rotor #4. Clearance and speed were varied. Cobra and performance probe data taken. Inability to repeat data points in this test series and the failure of tip ring instrumentation prompted the installation of a new tip ring. Cobra probe survey preformed behind the PSN at full speed. Installation was completed Feb. 24. Tests without liquid nitrogen cooling were conducted to replicate early Feb. performance and explore speed line roll over. Installation of fine precision bonnet in the liquid nitrogen control valve.

March 2004

Liquid nitrogen cooling tests with the new tip ring, performed March 1 failed to replicate early Feb. results. A teardown was initiated to install the PSN Aft assembly which moved the PSN blade closer to the rotor. Comparison of the 72 ct. PSN survey data taken at the blade exit and the rotor entrance locations suggests this configuration will provide a cleaner rotor entrance flow. Testing Mar. 10-11 demonstrated overall system performance was improved, but rotor performance was unchanged. In an effort to improve the surface finish of the abraded tip ring, a ring coated with Metco 2042 was installed in place of the existing Metco 601 ring. Sleeves were also installed over the guide pins locating the tip ring. These sleeves reduced the point loading and consequential travel increase in the tip ring during operation. Tests of the new abradable were conducted Mar. 24-25. Removal of a Ramgen equipment from the test occurred over the remainder of the month.

Specific Tests of Interest:

- Peak performance search varying speed, throttle
  - 65 ct. PSN: 140, 141, 143, 145 & 151
  - 72 ct. PSN, tip ring #1: 176,177, 179 & 180-182
  - 72 ct. PSN, tip ring #2: 187, 189, 192 & 193
  - 72 ct. PSN Aft: 194-196
  - 72 ct. PSN Aft, Metco 2042 abradable: 194-196
- Targeted tests
  - 146: 65 ct. PSN, 80% speed line
  - 147: Normally aspirated
  - 149-150: Tank pressure insensitivity
  - 190: PCB data for peak performance speed line
  - 201: 80-110% speed lines

2.4.1.2 Test Hours

Figure 2.4.1-1 was created from Boeing NTF data files received after each day of testing. The total test time is defined as the summation of time between acquisition of the first data point and the last data point on a day when test data was acquired. This definition of total test time encompasses all types of testing. The Rampressor 1 test rig total test time was 304 hours, 20 minute, and 24 seconds.

Figure 2.4.1-2 was also created from Boeing NTF data files. Total rotating test time is defined has the time between the first scan taken where the RPM was greater than 1,000 and the last scan acquired when the RPM was greater than 1,000. The total rotating test time for all flowpaths (FP1, FP2, and FP4) was 146 hours, 26 minutes, and 48 seconds

Rampressor 1 Test Rig Total Test Time 304:20:24

Figure 2.4.1-1: Test time extracted from Boeing NTF raw data files

Rampressor 1 Test Rig Total Rotating Test Time 146:26:48

Figure 2.4.1-2: Rotating test time extracted from Boeing NTF raw data files

2.4.2 65 Vane PSN Fwd Characterization

The goals of PSN characterizations were to:

- Obtain the PSN total pressure loss vs. inlet corrected flow so the rotor inlet total pressure, and rotor pressure ratio and efficiency can be calculated during rotor testing. (A radial survey in front of a rotor is risky. The rotor vibrations induced by a strong once per revolution excitation and its harmonics can damage the rotor. Also the rotor will induce a strong 3 per rev. excitation of the survey probe. Near the end of the program a survey was done between the PSN and the rotor at the design point).
- Compare the total pressure loss to design intent
- Measure the exit flow angle and compare to design intent
- Measure the exit Mach number and compare to design intent
- Measure the impact of bleed on flow angle, Mach number, and total pressure loss

A radial survey probe was fabricated to measure total pressure and flow angle (Cobra probe). The radial survey probe was installed into a survey actuator mounted just downstream of the PSN trailing edge (See Figure 2.4.2-1). Total pressure and flow angle were measured at 19 locations from hub to shroud (every 5% span) for various inlet corrected flows. Mach number was calculated using total pressure and a static pressure radially interpolated from the hub and tip static pressures. To verify the Cobra probe was properly nulled and measuring total pressure correctly surveys were made using a Keil head probe. The total pressure profile measured using the Keil head probe exactly matched the total pressure profile measured using the Cobra probe indicating that the Cobra probe was properly nulled (Note the Keil head probe was larger so it could not acquire data close to the hub).

Figure 2.4.2-1: Survey Probe Setup

—○—	Run 8-Hub Bleed to ATM 1.77 lbm/sec
—×—	Run 9-Hub Bleed to ATM 1.62 lbm/sec
—+—	Run 10-Hub Bleed to ATM 1.47 lbm/sec
—*—	Run 11-Hub Bleed to ATM 1.23 lbm/sec
—□—	Run 12-Hub Bleed to ATM 0.909 lbm/sec
—◇—	Run 16-Hub Bleed to ATM 1.76 lbm/sec
—▽—	Run 21-Hub Bleed to ATM 1.75 lbm/sec
—△—	Run 23-Hub Bleed to ATM 1.92 lbm/sec
—○—	Run 24-Keil Probe/Hub Bleed to ATM 1.76 lbm/sec
—×—	Run 25-Keil Probe/Hub Bleed to ATM 1.92 lbm/sec
—○—	Run 43-NO Bleed 1.75 lbm/sec
—○—	Run 44-Hub Bleed Plenum to ATM 1.76 lbm/sec
—*—	Run 50-NO Bleed 2.00 lbm/sec
—×—	Run 51-NO Bleed 1.70 lbm/sec
—○—	Run 74-NO Bleed-NO FF Step 1.74 lbm/sec
—☆—	Run 75-NO Bleed-NO FF Step 1.94 lbm/sec

Figure 2.4.2-2: Legend for radial profile plots

Figure 2.4.2-3 is a plot of total pressure profiles at various inlet corrected mass flows and bleed. Note the Keil survey profiles (red curves) exactly match the Cobra survey profiles at the same mass flow.

Figure 2.4.2-3: Radial Profile of PSN Exit Total Pressure

The impact of bleed on all parameters was insignificant on a mass average basis. Bleed did have a small effect on the shape of the profiles in the endwall region. It was decided to ignore bleed when calculating rotor inlet conditions. If the impact of PSN bleed on rotor performance was significant it can be measured by comparing rotor exit surveys with and without PSN bleed.

The no-bleed total pressure survey profiles were mass averaged and an Omega Bar ($(P_{t\_inlet} - P_{t\_exit}) / (P_{t\_exit} - P_{s\_exit})$) was calculated and fit vs. inlet corrected flow (See Figure 2.4.2-4).

Pre-Swirl Nozzle Mass Averaged Pressure Loss

Figure 2.4.2-4: Mass Average Omega Bar vs. Inlet Corrected Flow

The PSN exit flow angle was measured directly using the Cobra probe. To verify the alignment of the probe to the centerline of the rig a continuity check was performed. PSN exit mass flow can be calculated using the Cobra probe measurements (P_t , angle), inlet total temperature, and the static pressure distribution. This calculated mass flow could then be compared to the mass flow measured by the choked flow nozzle in the air supply piping. Any discrepancy between these measurements needs to be resolved (continuity). Discrepancies were resolved by biasing the measured flow angle from the Cobra probe. This was done because the most accurate measurement of mass flow is calculated from the choked flow nozzle in the air supply piping and the most inaccurate measurement required for calculating mass flow using the survey data is the probe offset from the centerline. The Cobra probe was nulled in a flow facility. A flag was installed on the probe to indicate null but unfortunately the survey actuator did not have a means to use this flag to ensure the probe was lined up with the rig centerline alignment pin on the survey mount. Therefore the only way the Cobra probe could be aligned with the centerline of the rig was by eye. An individual would stand several feet away while another individual would rotate the probe until the null flag on the probe aligned with a surface (nominally) parallel to the centerline. After several attempts it was determined that the ability to repeat the centerline alignment was probably plus or minus 10 degrees. Therefore all flow angle profiles were biased (plus or minus) to ensure the mass flow calculated by integrating the data from the survey plane matched the choked flow nozzle in the air supply plumbing (minus bleed if any). The resulting continuity corrected flow angle profiles are shown in Figure 2.4.2-5.

Figure 2.4.2-5: Radial profiles of PSN exit continuity corrected flow angle

Another important parameter for the PSN is the exit Mach number. The Mach number is calculated using the inlet total temperature, Cobra probe center pressure, and the hub and tip static pressures linearly interpolated vs. radius. The profiles of Mach number are shown in Figure 2.4.2-6.

Figure 2.4.2-6: Radial profile of PSN exit Mach number

The PSN exit meanline Mach number and the mass average Mach number were plotted vs. inlet corrected flow so the corrected flow at the design intent exit Mach number could be determined. The PSN hit the design Mach number target at 1.94 lbm/sec slightly lower than the design corrected mass flow of 2.01 lbm/sec (See Figure 2.4.2-7)

Figure 2.4.2-7: Mass Average PSN Exit Mach number vs. inlet corrected flow

In the plots below the survey data is compared to design intent and three CFD models of the design. The design intent is a meanline target (solid black line). The CFD runs are shown using small filled symbols. The raw data is shown using large open symbols. Two profiles of raw data are shown because a survey was not acquired at the corrected mass flow where the PSN achieved the desired mass average Mach number (1.94 lbm/sec).

The measured Mach number profile shape is very similar to the CFD models of the PSN. The measured tip boundary layer is very small the hub boundary layer looks slightly larger in the measured data. (See Figure 2.4.2-8).

Figure 2.4.2-8: Radial profile of PSN exit Mach number vs. CFD runs and design intent

Figure 2.4.2-9 is a plot of flow angle profile compared to design intent and CFD runs. In Figure 2.4.2-9 the large filled symbols are the continuity corrected profiles. The large open symbols are the raw data. A minus six-degree bias in the measured flow angle was required to achieve continuity. Six degrees is well within the manual probe alignment window.)

Figure 2.4.2-9: Radial profile of PSN exit flow angle vs. CFD predictions and design intent

The Omega bar profile for the most part matches the CFD predictions except in the hub region where the measured loss is slightly higher than the predicted loss (see Figure 2.4.2-10)

Figure 2.4.2-10: Radial profile of PSN exit Omega Bar vs. CFD predictions

The PSN hit the design Mach number target at a flow slightly lower than the design flow (1.94 lbm/sec vs. 2.01

lbm/sec) but the exit absolute flow angle and loss levels were close to design intent. A decision was made to not modify the PSN to achieve the design Mach number at the design flow since the speed flow characteristics of the rotor were not known. It turned out to be a wise decision because the rotors corrected flow at the peak efficiency point (minimum clearance, optimum bleed) was only 1.78 lbm/sec.

2.4.3 FP2 & FP4 Proof Testing

2.4.4 FP1 Results Synopsis

The purpose of FP1 testing was to check out the mechanical operation of the rig. Three issues were uncovered - two were resolved completely, one continued to affect testing.

The first issue uncovered was an issue that was specific to FP1 only. Turbine operation and speed control was very difficult. This issue was resolved to a certain extent after several (nervous) hours of operation. The only torque FP1 exerted on the turbine was due to windage and bearing drag (very small). While the turbine control was capable of being tuned to handle a no load situation it became apparent that lessons learned from tuning the turbine for FP1 could not be applied to other rotors where the torque was a strong function of speed, inlet total pressure, bleed, clearance, and throttle.

The second issue uncovered was the capacity of the fwd and aft oil drains was not large enough to handle the oil and the buffer air. Increasing the size of the oil drain line and adding an oil sump to the oil system resolved the issue.

A bearing failure occurred during FP1 testing. Subsequent inspection of the failed bearing determined that the bearing balls were out of specification – oversize. The resultant decreased clearance resulted in runaway thermal behavior and subsequent ball failure. Fortunately the failure was graceful, accompanied by a readily-noticed increase in vibration so the rig was halted before damage to additional components could occur. Ceramic ball bearings were ordered with 100% inspection requirement to eliminate this problem from future testing and ensure graceful failure under all conditions.

2.4.5 FP2 Results Synopsis

FP2 was a rotor with strakes and no compression ramp. The purpose of FP2 testing was to:

- Check out the cap probe clearance measurement system.
- Practice running the drive turbine with a load.
- Practice running the test facility main air system.
- Practice running the exhaust throttle valve.
- Practice balancing the rotor thrust
- Practice nulling the rotor exit survey cobra probe

The first test performed focused on the main air system. The main air was slowly ramped up to 1 lbm/sec. FP2 windmilled in the desired direction and the rotor was spun up to 12,000RPM.

The next test involved the main air system and the drive turbine. The rotor was windmilled up to 8,000RPM. The constant speed target of the drive turbine control was set to 8,300RPM. The drive turbine was engaged and the RPM quickly stabilized at 8,300 RPM. The drive turbine control was much easier to tune with a load.

The final test was a basic compressor test. The rig was operated at various corrected speeds and data was acquired at various throttle settings (see Figure 2.4.5-1). It was decided not to vary the clearance to preserve the tip ring abradable and avoid damaging the rotor. All systems performed well up to 100% speed. The rotor exit survey data from the cobra probe matched the fixed rake Kiel head data when the exit flow angle was within the Kiel head flow angle tolerance. All data was acquired with a heated tip ring so clearance was open. It was decided to move on to FP4 compression rotor testing.

Figure 2.4.5-1 is a plot of FP2 performance. The fixed rake data was slightly lower than the mid-span Cobra probe data. This was expected since the rakes have more immersions close to the endwalls.

Figure 2.4.5-1: FP2 Performance Map

2.4.6 FP4 Results Synopsis

2.4.6.1 Initial Test Results

Initial testing was done with the tip heat on (open clearance). The maximum P/P achieved was a little more than 1.4. The measured clearance was between 5-10 mils which is nominal for most compressors. Figure 2.4.6-1 is FP4 compressor performance map (all speeds) with open clearance.

**FP4
Open Clearance Data**

Figure 2.4.6-1: Early FP4 performance (Open Clearance)

Figure 2.4.6-2 is a FP4 open clearance compressor performance map with the design point plotted.

**FP4
Open Clearance Data**

Figure 2.4.6-2: Early FP4 data with design point

In early November it was decided to close down the clearance and the dramatic impact of clearance on performance became evident. Figure 2.4.6-3 shows the dramatic impact of clearance on performance. Reducing clearance at a constant corrected speed (107%) dramatically increased the pressure ratio of the rotor and the amount

of flow the rotor pumped.

Figure 2.4.6-3: FP4 performance with reduced clearance

Once the clearance was closed down and the impact of a leaky exhaust valve overcome, the compressor was throttled up to a P/P much closer to design. Figure 2.4.6-4 shows that the rotor was capable of reaching the design point pressure ratio (110% speed tight clearance).

FP4

Figure 2.4.6-4: FP4 performance showing the impact of clearance and throttle

Figure 2.4.6-5 is a plot of corrected flow vs. pressure ratio and illustrates typical FP4 pressure ratio mass flow operation. An operational procedure was established to ensure the behavior of the rig could be monitored at several repeated conditions. The first step in the procedure was to open all the rig exhaust valves. The inlet total pressure in the inlet tank was set to 20 psia. The resulting airflow windmilled the rig to around 8,000 RPM. The next step was to charge the line upstream of the drive turbine with hot air. Then the drive turbine was started with an RPM setpoint that resulted in a corrected speed of 80% (around 15,000 RPM). Once the drive stabilized at 80% corrected speed the rotor was started. In Figure 2.4.6-5 starting is noted so the performance difference at 80% speed can be seen. The rotor could then be spun up to the desired speed, throttled, and clearance could be adjusted.

Figure 2.4.6-5: FP4 typical speed flow P/P operation

Figure 2.4.6-6 shows the dramatic impact of clearance on performance and throttle. The tighter the clearance the more vertical the throttle line. High pressure ratio data points could only be achieved at tight clearance.

Figure 2.4.6-6: FP4 Throttle clearance sensitivity

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After sensitivity tests were concluded an effort was undertaken to discover the optimum settings for all variables. Figure 2.4.6-7 represents all the data acquired during one shift of testing. The "+" symbols are real time updates and the red circles are saved data scans. This plot indicates that most of the data acquired, analyzed, and

viewed during testing is not recorded.

Figure 2.4.6-7: FP4 search for optimum operating point

It was decided that the peak efficiency point would be the data point with the highest performance probe based efficiency with the performance probe parked mid span. The performance probe temperature was used to calculate exit temperature because the exit collector RTD's were tainted by the heat transfer of the chilled rotor shroud (see Figure 2.4.6-10). Figure 2.4.6-8 is a rotor only P/P vs. efficiency plot showing the best efficiency point calculated by the performance probe. Main flow thermal and energy balance efficiency is also shown. The main flow thermal and energy balance efficiencies are calculated using the exhaust collector RTD's, which the chilled rotor shroud, was clearly influencing. The turbine based efficiency was set to zero when the turbine flow measurement nozzle was not choked. Choking the turbine flow nozzle was very difficult operationally to achieve.

Figure 2.4.6-8: FP4 Rotor P/P vs. Efficiency for best operating point

Radial rotor exit surveys at the peak efficiency point show the profile of pressure ratio is reasonably flat (see Figure 2.4.6-9).

Figure 2.4.6-9: FP4 Rotor Exit P/P profile at peak efficiency

The rotor exit DT/T ($(T_{\text{exit}} - T_{\text{inlet}})/T_{\text{inlet}}$) profile (see Figure 2.4.6-10) clearly indicates that the chilled rotor

shroud is impacting the airflow temperature (primarily at the tip).

Figure 2.4.6-10: FP4 Rotor Exit DT/T profile

The rotor exit efficiency profile (see Figure 2.4.6-11) also clearly shows the chilled rotor shroud is impacting the exit temperature.

Figure 2.4.6-11: FP4 Rotor Exit Efficiency Profile

2.4.6.2 72 vane PSN characterization

In an attempt to improve FP4 performance two modifications were made to the PSN. The first modification was the blade count. The blade count was increased from 65 to 72 so the PSN exit Mach number would be higher at a lower corrected flow. The new meanline target corrected flow for the PSN was 1.78 lbm/sec corrected. The test results of the 72 vane PSN are compared to the 65 vane PSN below. Figure 2.4.6-12 is the legend for the plots. All of the 65 vane PSN surveys are shown with black lines and symbols all of the 72 vane PSN surveys are shown with red symbols.

⊖	Run 43 Forward Facing Step Mach 0.59
✖	Run 74 NO Forward Facing Step Mach 0.58
+	Run 75 Design Point Mach 0.73
*	Run 79 Mach 0.5-Flow 1.60
⊕	Run 80 Mach 0.6-Flow 1.80
▽	Run 81 Mach 0.7-Flow 1.92
△	Run 82 Max Mach-Flow 1.99
◁	Run 50 Mach 0.83
▷	Run 51 Mach 0.56
★	Run 11 Mach 0.35
⊗	Run 12 Mach 0.25
⊖	Run 155 72 Vanes-Mach 0.63-Flow 1.74
✖	Run 156 72 Vanes-Mach 0.68-Flow 1.79
+	Run 157 72 Vanes-Mach 0.71-Flow 1.82
*	Run 158 72 Vanes-Mach 0.73-Flow 1.84
⊕	Run 159 72 Vanes-Mach 0.47-Flow 1.49
▽	Run 160 72 Vanes-Mach 0.74-Flow 1.85
△	Run 161 72 Vanes-Mach 0.73-Flow 1.84
◁	Run 162 72 Vanes-Mach 0.52-Flow 1.59
▷	Run 164 72 Vanes-Mach 0.27-Flow 0.96
⊗	Run 165 72 Vanes-Mach 0.75-Flow 1.86

Figure 2.4.6-12: Legend for PSN exit profile plots comparing 65 vanes (Black) with 72 vanes (Red)

The PSN exit total pressure profiles for the 72 vane PSN are very similar to the 65 vane PSN (see Figure 2.4.6-13)

Figure 2.4.6-13: Exit total pressure profile 65 vane PSN (black) vs. 72 vane PSN (red)

The 72 vane PSN Mach number profile was also very similar to the 65 vane PSN profile (see Figure 2.4.6-14)

Figure 2.4.6-14: PSN exit Mach number profile 65 vane PSN (black) vs. 72 vane PSN (red)

The meanline and mass average Mach number was increased at a lower inlet corrected flow. The target

meanline Mach number (the star symbols above each data point) was achieved at a flow slightly higher than the desired flow (1.81 lbm/sec vs. 1.78 lbm/sec see Figure 2.4.6-15)

Figure 2.4.6-15 Meanline and Mass Average PSN exit Mach number vs. inlet corrected flow 65 vane PSN (black) vs. 72 vane PSN (red)

The Omega Bar of the 72 vane PSN was slightly higher than the 65 vane PSN as expected. Mostly in the endwall region (see Figure 2.4.6-16)

Figure 2.4.6-16: PSN exit Omega Bar profile 65 vane PSN (black) vs. 72 vane PSN (red)

A new curve of inlet corrected flow vs. Omega bar was created and used to determine rotor inlet conditions (see Figure 2.4.6-17)

Figure 2.4.6-17: 72 vane PSN Omega Bar curve fit

2.4.6.3 Characterization of the 72 vane PSN Aft configuration

In another attempt to improve FP4 performance a survey was acquired closer to the PSN trailing edge. If the PSN exit conditions were significantly improved the PSN could be installed closer to the rotor (referred to as the PSN Aft configuration). Figure 2.4.6-18 is the legend for the PSN survey plots. Data from the nominal PSN configuration (PSN Fwd) are plotted using red lines and symbols. Data from the PSN Aft configuration are plotted using green line and symbols.

Figure 2.4.6-18: Survey legend for PSN Exit Surveys. Red curves survey is at the rotor leading edge. Green curves the survey is upstream of the rotor leading edge

The PSN exit total pressure profile and the PSN exit Mach number profile were dramatically different closer to the PSN trailing edge (see Figure 2.4.6-19 and Figure 2.4.6-20)

Figure 2.4.6-19: PSN exit total pressure profile at rotor leading edge (red) and upstream of rotor leading edge (green)

Figure 2.4.6-20: PSN exit Mach number profile at rotor leading edge (red) and upstream of rotor leading edge (green)

The meanline and mass average Mach number at a given inlet corrected flow was slightly higher (see Figure 2.4.6-21).

Figure 2.4.6-21: PSN meanline and mass average exit Mach number vs. inlet corrected flow for survey at rotor leading edge (red) and upstream of rotor leading edge (green)

The PSN loss levels were significantly lower as indicated by the profiles in Figure 2.4.6-22. The losses in both endwalls were improved dramatically and the overall loss level was significantly less (see Figure 2.4.6-22).

Figure 2.4.6-22: PSN exit Omega Bar profile at rotor leading edge (red) and upstream of rotor leading edge (green)

Figure 2.4.6-23: Mass average PSN Omega Bar at rotor leading edge (red) and upstream of rotor leading edge (green)

Based on the improvement in the Mach number profile and the loss levels a plan was enacted to test FP4 with the PSN installed closer to the rotor (PSN AFT configuration).

2.4.6.4 FP4 performance with 65 vane PSN Fwd, 72 vane PSN Fwd, and 72 vane PSN Aft

FP4 was re-installed and data was acquired to see if FP4 rotor only performance was strongly influenced by:

- PSN Mach number at a given inlet corrected flow (72 vane PSN vs. 65 Vane PSN)
- PSN exit pressure and Mach number profiles (PSN Fwd vs. PSN Aft)

It was very difficult to exactly replicate the previous 65 vane PSN Fwd data. Acquiring data points at the same or better clearance proved to be difficult due to the cold build clearance and the concentricity of the cold build clearance. Eventually enough data was acquired to prove that a small increase in the PSN exit Mach number at a given inlet corrected flow had very little impact on FP4 rotor only performance. The 72 vane PSN Aft configuration was not tested at all levels of clearance (especially tight clearance) but early indications (open clearance data) were that the improvement in PSN exit pressure and Mach number profiles did not have a strong influence on FP4 rotor only performance.

Figure 2.4.6-25 is a plot of all the data acquired for all configurations. At this scale it is clear that no PSN configuration strongly influences FP4 total pressure ratio vs. inlet corrected flow performance.

Figure 2.4.6-24: Legend for FP4 performance plots

Figure 2.4.6-25: FP4 Rotor only performance map for various PSN configurations

A zoomed in view of the 107% corrected speed data (Figure 2.4.6-26) shows that peak pressure ratio for the 72 vane PSN Fwd configuration is slightly higher than the 65 vane PSN Fwd configuration. The open clearance 72 vane PSN Aft data also shows a slight improvement in peak pressure ratio.

Figure 2.4.6-26: FP4 Pressure Ratio vs. Inlet Corrected Flow 107% Speed, 65 vane PSN Fwd (black), 72 vane PSN Fwd (red), 72 vane PSN Aft (green)

The improvement in peak pressure ratio did not result in an improvement in rotor only efficiency. Figure 2.4.6-27 is a zoomed in plot of the 107% speed thermal efficiency characteristic. Figure 2.4.6-28 is a zoomed in plot of the 107% speed net rotor efficiency (also referred to as energy balance efficiency) based on the performance probe measurements. Both figures show that the best 65 vane PSN Fwd data points could be repeated with the 72 vane PSN Aft configuration. The 72 vane PSN Fwd configuration was never tested at an equivalent clearance due to mechanical issues (abradable failure)

Figure 2.4.6-27: FP4 Rotor only thermal efficiency 107% Speed, 65 vane PSN Fwd (black), 72 vane PSN Fwd, (red) 72 vane PSN Aft (green)

Figure 2.4.6-28: Net Rotor Efficiency base on Performance Probe 65 Vane PSN Fwd (black) and 72 Vane PSN Fwd (red)

2.4.7 Rotor Leading Edge Incident

All FP4 strake leading edges were damaged during Run 196. Run 196 was an attempt to obtain minimum clearance data for the 72 vane count-PSN Aft configuration. Prior to Run 196 the thrust balance system was not functioning properly due to the aft outer race of the bearing sticking. In the past the sticking problem usually went away after a few hours of running. This time it did not. Testing was terminated and a meeting was held with all of Ramgen engineering to discuss the data acquired so far and to decide if the rig should be torn down to fix the thrust problem. A decision was made by the entire Ramgen engineering team to continue running (at risk) to obtain the minimum clearance data point for the current configuration rather than taking multiple days to tear down the rig and troubleshoot. Wheel space pressures were set to levels previously run to ensure the aft bearing was being loaded properly.

The rig was running at 107% corrected speed and the throttle was set to achieve peak efficiency with the tip heat off. The LN2 cooling system was brought on line and the tip shroud was being cooled to reduce clearance. The tip shroud had just reached a thermal equilibrium point. Boeing was able to maintain a relatively constant tip ring cooling air temperature (4.0 degF plus or minus 2degF). Based on Cap probe readings and prior borescope inspections the rotor was rubbing in the leading edge and trailing edge locations. Deeper at the leading edge. Prior to taking data the rig was throttled to ensure the rotor was operating at peak efficiency. At first the rig was throttled down (valve open). Rotor P/P and efficiency dropped. Then the rig was throttled past the peak P/P and efficiency point achieved with tip heat off. The performance of the rotor dropped rapidly (P/P dropped, Corrected Flow dropped) indicating an un-start. At the same instant one of the vibration sensors tripped (> 0.7 inch/sec), the indicated thrust saturated (> 1000 lbf), and all of the Cap probes stopped working. The rig also speeded up slightly.

The vibrations quickly returned to normal levels. The Cap probes did not recover. The indicated thrust was still saturated. The rotor flow and P/P settled out at a much lower P/P and Flow. Data was acquired and the rig was shut down in the normal fashion (tip heat on, open throttle, slow decel).

The Figure 2.4.7-1 shows the thrust sensor saturation that occurred during the un-start. Figure 2.4.7-2 shows what the vibration sensors detected. Figure 2.4.7-3 illustrates the compressor map taken during Run 196 and shows how radically the performance changed after the event.

After the event the rig was borescoped and a decision was made to teardown the rig since all the abrasible was gone and the tips of the strakes were damaged as shown in Figure 2.4.7-4. The tip ring abrasible was found to be severely worn (in some cases down to the aluminum ring material) as shown in Figure 2.4.7-5 and Figure 2.4.7-6.

After reviewing all of the data and inspecting the damage a most likely scenario was developed to explain the damage. It apparently started with a rotor un-start at 107% (usually benign), followed by the Aft bearing outer race coming free. Wheel space pressures along with rotor un-start pressures force the rotor aft suddenly (Thrust >1000lbf). Resulting static case vibration (>0.7 inch/sec) or fwd bearing unload causes a sudden hard rub. The rub could be radial as well as axial. Clearances are already tight (cryo is on) and the rotor is running in a trench especially near the leading edge. The thin section of titanium strake catches fire due to rubbing. The titanium fire extinguishes once it reaches a thick section. The fire along with the hard rub consumes most of the abrasible.

Based on the scenario above the following lessons were learned.

1. The rig should not be run if careful control of the thrust balance system is not available.
2. It is possible to start the ultra-thin leading edges of the strakes on fire.
3. If the strake catches fire it extinguishes quickly.
4. The rig can be operated safely during and after a strake fire.

Repair options for the monolithic rotor consisted of re-building the strake leading edges via welding or laser powder deposition or re-machining the remaining material such that the leading edge had a sweep back angle. Due to time constraints, the re-machining of the existing material option was implemented with the resulting new leading edge geometry showing in Figure 2.4.7-7, which is identical for all three leading edges. All subsequent tests utilized this rotor leading edge geometry.

Figure 2.4.7-1: Run 196 diagnostic parameter traces during unstart event and subsequent rotor leading edge fire.

031104 Run 196 Vibration

Figure 2.4.7-2: Run 196 velocity and rotor speed traces during the rotor leading edge fire event.

Figure 2.4.7-3: Run 196 rotor pressure ratio vs. corrected flow map during rotor leading edge fire event.

Figure 2.4.7-4: Photograph of one of the rotor leading edges illustrating the typical amount of material removal due the Run 196 event.

Figure 2.4.7-5: Post Run 196 rotor LE fire event inspection of the tip ring abradable illustrating the amount of material removed by the rotor strake tips.

Figure 2.4.7-6: Photograph of tip ring abradable wear after Run 196 rotor LE fire event.

Figure 2.4.7-7: Photograph of modified rotor leading edge (typical of all three) post leading edge fire event. All runs after 196 utilized this leading edge geometry.

2.4.8 Compressor Mapping

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2.5 Post Test Analysis

2.5.1 CFD vs. Raw and Processed Experiment Data

2.5.1.1 Full Rotor CFD Modeling

A full 3-D viscous STAR-CD rotor model was assembled which included the pre-swirl nozzle design, bleed characteristics, and tip gap modeling capability. The model is shown in Figure 2.5.1-1 and, unlike previous rotor modeling efforts, the entire rotor flow path – pre-inlet, compression ramp, diffuser, exhaust, and tip gap – were modeled.

2.5.1.2 Comparison to Raw and Process Experiment Data

Shortly after Rampressor 1 testing was complete the results of a CFD model of the 72 vane count PSN and FP4 were compared to rig data. The CFD model was run at two different clearances 2 mil gap and a 4 mil gap. The CFD inlet pressures and temperature were set to rig inlet conditions. The RPM was set to 107% corrected speed. The bleed in the CFD model was set to the optimum bleed levels found during rig test. The CFD model calculates a mass flow, pressure ratio and efficiency. So in the plots below the CFD is predicting the X and Y-axis values.

In the plots below the CFD data is compared to the raw rig data. The raw rig data is all data acquired at 107% corrected speed.

As the plots indicated the CFD results definitely lie within the range of data and the CFD results tend to line up best with the 72 vane count PSN.

In the plots below the CFD data is compared to processed data. The speed line shape in the processed data was acquired at optimum bleed settings and a relatively tight clearance. The speed line was then scaled to the best operating point (optimum bleed, optimum clearance) indicated with a star on the plot.

The plots above clearly show that the CFD predictions come very close to the measured performance. The inlet corrected flow at 4mil clearance matches the measured corrected flow. The pressure ratio predicted by the CFD is also very close to the measured pressure ratio. The CFD prediction of efficiency is slightly lower than the measured efficiency but given the difficulties of measuring adiabatic efficiency in the rig with all the heat transfer at the shroud the comparison is quite good.

Figure 2.5.1-1: Illustration of 3-D viscous STAR-CD full rotor model.

Figure 2.5.1-2: Typical Mach contour results at mid-inlet height from full 3-D viscous STAR-CD rotor model.

Figure 2.5.1-3: Comparison of CFD to cobra and performance probe experimental data.

Figure 2.5.1-4: Comparison of CFD to performance probe experimental data.

Figure 2.5.1-5: Comparison of CFD using performance map.

Figure 2.5.1-6: Comparison of 2 mil and 4 mil CFD speed lines and experimental data

Figure 2.5.1-7: Combined rotor pressure ratio and gas path efficiency versus corrected mass flow for 2 mil and 4 mil tip gaps: CFD model at various back pressures and test rig peak efficiency point.

2.5.2 Self Aspiration

To determine if FP4 would operate without the ram effect of the Boeing main air system a shift of testing was performed without Boeing's main air system. Boeing's main air system was disconnected just upstream of the flow conditioner. The inlet corrected flow was calculated using the rig bellmouth static pressures, inlet total temperature and inlet total pressure. The effective area of the bellmouth was calculated using data acquired from all the previous runs. Essentially the bellmouth was calibrated to the Boeing main air MCV's.

80% corrected speed data indicated that FP4 did not need the Boeing main air system to operate. As expected the rotor was throttled up slightly. Note in the figure below how nicely the red points line up with the speed lines indicated by the green points.

More data was acquired at high speed and it backed up the observations made at low speed. The Rampressor operates like all other compressors: the rotor pumps the flow.

Figure 2.5.2-1: Graph showing self-aspiration capability.

Figure 2.5.2-2: Rampressor does not require elevated pressure air supply to operate properly.

2.6 Segmented Rotor Overview

A segmented design was chosen to minimize k_t effects due to hoop load in bleed hole regions, leading to higher

operating speeds of rotor, thus allowing a reduced PSN Mach number which would reduce PSN distortion/loss. Also, this design would generate higher rotor pressure ratio.

2.6.1 Goals

The goals of the segmented rotor (RP-1.6S) effort included:

- To design a robust, low risk, fast turn-around mechanical system to replicate Flow Path 4.
- Learn about designing and building a segmented Rampressor flow path

2.6.2 Objectives

The objectives of the Rampressor 1.6S effort include:

- Design the rotor to operate at a relative Mach # of 1.6.
- Initially operate the rotor at 20,000 RPM to enable a direct comparison between the monolithic and segmented performance
- Mechanical design to enable minimization of pre-swirl Mach number: aero design speed: 27,230 RPM
- Minimize static hardware changes to reduce cost and lead times.

2.7 Segmented Rotor Aero Design

Increased rotor speed and the decreased in the amount of pre-swirl (to maintain the same rotor-relative Mach number) serves several potentially advantageous functions:

- Reduce rotor inflow distortion and loss
- Increase rotor only total pressure ratio
- Increase rotor-only (not necessarily stage) efficiency

The high PSN outflow Mach number (0.77) of the Rampressor-1 design was a result of the desire to maintain a Mach 1.6 rotor-relative flow while reducing the rotor rim speed as much as possible to reduce stresses in the monolithic rotor design. The biggest downside of this high pre-swirl design is the possibility of having a highly distorted flow exiting the pre-swirl nozzles which adversely affects the rotor performance (efficiency, pressure ratio, etc.). Thus a design with much lower pre-swirl Mach and outflow angles was required to accompany the segmented rotor design.

The aerodynamic re-design required for a higher RPM segmented rotor that maintains a Mach 1.6 relative flow consists of performing a mean line design to determine the RPM vs. PSN Mach requirements of the fixed geometry rotor (strake angle and radii), a detailed design of the pre-swirl nozzles to produce the desired Mach number and swirl angle, and an update of the rotor flow path design due to limitations on the bleed flow rates and placement imposed by the segmented rotor design relative to the monolithic rotor design.

2.7.1 Mean Line Design

For a fixed rotor geometry, i.e., strake angle and flow path for Mach 1.6 relative operation, the only variable (besides supply air total temperature) that determines rotation rate is the amount of pre-swirl. Thus a parametric study of the effect of varying pre-swirl Mach number ($M_{2.4}$) was performed to determine how low a pre-swirl Mach number to design the system. The lower the pre-swirl the higher the RPM up to the point where the flow out of the pre-swirl nozzles is purely axial and the maximum design RPM is achieved – this is equivalent to zero pre-swirl. The pre-swirl was varied over the range of $0.212 < M_{2.4} < 0.74$ with the minimum value corresponding to zero pre-swirl and the maximum value corresponding to the current Rampressor-1 monolithic rotor design value. The results of the study are shown in Figure 2.7.3-1 which indicates the relationship between pre-swirl outflow Mach and RPM which maintains zero incidence on the strake leading edge. Thus at zero pre-swirl (Mach 0.212) the theoretical maximum design rotor speed is 38,547 RPM (100% design speed at 50 F air total temperature). The only limiting factor is the RPM capability of the segmented rotor which is based on rotor dynamics or disc/segment stresses.

A rotordynamics analysis was performed by Barbour Stockwell, Inc. (BSI) with the segmented rotor which

determined the first critical occurs at approximately 32,000 RPM. BSI recommended that the rotor never exceed 29,000 RPM for any length of time, thus setting the absolute upper limit. Rotor/segment stresses were not exceeded at this speed, so the rotor dynamic critical speed set the upper design limit. From this upper limit a design speed is determined with a mechanical safety margin plus aerodynamic margin. A speed of 27,250 RPM (100% design speed) was chosen based on the parameters explained in Figure 2.7.3-2. A 2.5% aero margin plus a 4 % mechanical over-speed margin were utilized. The mechanical over-speed margin is required due to the possibility of sudden rotor power requirement decrease during an unstart event. A sudden resistive torque removal from the drive turbine could make the system RPM suddenly accelerate before the drive control system could close the turbine air supply valve and reduce the power production. A very conservative approach to calculate the amount of over-speed is to assume that all of the torque requirement of the rotor is lost. This analysis was performed by BSI based on the following assumptions:

- 30K RPM
- 355 HP load instantly removed from drive
- 370 lbm-in² rotor inertia
- 0.5 sec valve closure (nothing happens for ½ sec, then instant close)

A maximum over-speed of 32,500 RPM is calculated which corresponds to a 8.3% speed increase. Thus if 29,000 RPM is the maximum the maximum speed is well below 27,000 RPM. A more realistic analysis is based on:

- Non-Instant Removal of Aero Torque
- Rotor unstart or inlet starting (not back pressured) does not produce significant torque changes (based on CFD & analysis)
- The only way feasible for quick aero unloading is if Boeing air supply was quickly cut off:
  - >100 ft. of 8” pipe + tank to blow down
  - Air control valve is slow (~1 sec to close)
- 27.25K RPM HP is actually less than the 355 HP used for calculations (at ~4 lbm/s physical flow)
  - This HP requirement may increase depending on the final physical flow rate
- BSI Drive Has Faster Than 0.5s Valve Response
  - More like 300 – 400 msec per conversations with BSI
- Drive would not provide full torque during sensing and valve closure transient

Based on these assumptions, a 4% over-speed was determined to be adequate due to the calculations indicating that even one-half of the rotor power requirement could be instantly lost (high unlikely) at 355 HP and the rotation rate would not exceed 29,000 RPM starting at 102.5% of the design speed.

Thus the pre-swirl nozzle design Mach number was determined to be 0.477. The associated rotor parameters at these conditions are shown in Table 2.7.3-2 along with the Rampressor-1, 29,000 RPM, and 31,000 RPM operating conditions for reference. Thrust, torque, and power requirements for the Rampressor-1, 27,250 RPM, and 29,000 RPM conditions are shown in Table 2.7.3-3 through Table 2.7.3-5. These tabulated values are used for power/torque requirements at various rotor operating states (unstarted/low back pressure, unstarted/high back pressure, started/low back pressure, and started/high back pressure).

2.7.2 PSN Design

Once the PSN outflow Mach number and angle were determined by the mean line design analysis and the rotation rate limits, the PSN could be designed. This was accomplished by subcontracting the PSN design to Adapco which used STAR-CD solver coupled with Mode Frontier optimization software to arrive at an “optimum” design for the PSN.

- Model included Hub & tip fillets (0.030” radius)
- 1.5 % minimum leading edge radius

- 1.2 % trailing edge radius
- Varied Parameters included:
  - Axial chord length
  - Blade count
  - Maximum camber
  - Location of maximum camber
  - Maximum blade thickness
  - Stack Axis vs. Radius
  - Uniform loading length of the blade (from LE to TE)
  - Static pressure @ exit
  - Setting angle vs. Radius

The STAR-CD analysis type was steady state which employed a variation of the well known SIMPLE algorithm. This is a segregated, implicit solution algorithm.. The discretization was an upwind difference scheme which used multidimensional first-order accurate spatial discretization for stability. The flow & turbulence modeling used compressible air flow at 50 degrees R enters the model. Density is varied using the ideal gas law (pressure-based algorithm) and viscosity varied with temperature according to Sutherland's law. A low Reynolds number Spalart-Almaris turbulence model was applied. The specified geometry and operating conditions were:

- $R_{Hub} = 5.036$ in.
- $R_{Tip} = 5.714$ in.
- $\alpha = 63.21^\circ$
- $T_{o,Inlet} = 50$ F
- Mass Flow = 3.935 lbm/s
- $P_{o,Inlet} = \sim 23$ psia
- $P_{s,Exit} = 19.2$ psia
- $M_{Exit} = 0.477$

The optimization parameters and their priority were set by RPS as:

- Deviation from target swirl angle dist. – *minimize std. deviation*
- Total Pressure Loss: ϖ – *minimize mean value*
- Deviation from target Mach profile – *minimize std. deviation*
- Wake mixing length – *(keep below 2 duct heights)*

Initially, three meshes were used to perform a grid resolution study utilizing the Rampressor-1 PSN design parameters (Mach 0.77 outflow, 73.3 degrees swirl angle). The meshes differed primarily in the number of radial cell rows (streamlines):

- Coarse: 33 streamlines(32 cell rows), 254 K cells
- Medium: 50 streamlines(49 cell rows), 377 K cells
- Fine: 65 streamlines (64 cell rows), 501 K cells

An illustration of the grid and boundary conditions utilized for these simulations is shown in Figure 2.7.3-3. The study (sample results shown in Figure 2.7.3-4) determined that there was no significant difference between the Medium and Fine meshes and that the Coarse mesh showed significant differences from the higher resolution simulations. Thus for all PSN design/optimization simulations for the 1.6S operating conditions the Medium

resolution grid was utilized.

The final solution arrived at by Adapco is shown in Table 2.7.3-6 which lists the high level/1-D design goals vs. what the final solution was able to obtain. The differences were deemed acceptable by RPS and the PSN design was frozen and the final PSN blade shape is illustrated in Figure 2.7.3-5. Volume and mean line section contour plots for Mach number, total pressure and static pressure are shown in Figure 2.7.3-6. Although the contour plots are very useful, and the 1-D averaged values are used to measure how close the high level design goals were achieved, circumferentially-averaged radial variations in the outflow properties are most useful to determine how acceptable the PSN design actually is. An illustration of the circumferential averaging of properties (mass or mass-flow-rate weighted depending on the property) is shown in Figure 2.7.3-7. The circumferentially averaged radial profiles of pressure loss, exit angle, exit Mach, and exit total pressure of the final design are shown in Figure 2.7.3-8.

2.7.3 Segmented Rotor Flow Path Design

An aerodynamic re-design of the segmented flow path is required due to the different placement of bleed features between segmented and monolithic rotors as well as the difference in the PSN outflow characteristics.

2.7.3.1 Segmented Flow Path Design

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2.7.3.2 Slotted Throat Segmented Rotor Design

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Figure 2.7.3-1: Plot of pre-swirl Mach # vs. rotor RPM for the Rampressor-1 rotor design.

M _{2.4}	PS angle (deg)	RPM	Rotor only PR static	Rotor only PR total	M _{2.7}	Rotor exit flow angle (deg)
0.21216	0.919482636	38547.96	3.44	5.16	0.82	85.34
0.25	31.94777939	35347.42	3.44	4.61	0.71	84.65
0.3	44.99982536	33315.66	3.44	4.27	0.64	84.10
0.35	52.69243864	31608.33	3.44	4.00	0.59	83.55
0.4	57.97213064	30030.77	3.44	3.76	0.54	82.95
0.45	61.87440095	28522.9	3.44	3.53	0.49	82.27
0.5	64.89582793	27058.82	3.44	3.32	0.45	81.47
0.55	67.31305786	25625.65	3.44	3.13	0.40	80.52
0.6	69.29512294	24216.52	3.44	2.95	0.36	79.34
0.65	70.95213246	22827.68	3.44	2.77	0.32	77.87
0.7	72.35931355	21457.13	3.44	2.61	0.27	75.94
0.74	73.34155368	20373.2	3.44	2.49	0.24	73.93

Table 2.7.3-1: Values of pre-swirl outflow angle, rotor RPM, rotor total pressure, absolute outflow Mach and angle as a function of pre-swirl Mach number.

Figure 2.7.3-2: Illustration of the origin of the 27,250 RPM operating point.

Tip Radius	inches	5.714
Hub Radius	inches	5.036
Physical/Actual Mass Flow	lbm/sec	3.935
Inflow Total Temperature	deg. F.	50.00
Inflow Total Pressure	psia	23
Inlet Flow angle	deg. from axial	0
Inlet Mach #	M2.3	0.194
Exit Mach #	M2.4	0.477
Exit Swirl Angle	deg. from axial	63.21

		Rampressor 1.6S		Rampressor 1	
		27.25K RPM	~29K RPM	~20.5K RPM	~31K RPM
Physical Speed	RPM	27,250	29,025	20,483	31,002
Corrected Speed	Nc	27,481	29,271	20,656	31,264
Corrected Mass Flow	Mc	2.493	2.573	2.109	2.649
Physical/Actual Mass Flow	lbm/sec	3.935	4.062	3.191	4.181
Tip Radius	inches	5.714	5.714	5.714	5.714
Tip Speed	ft/sec	1,359	1,447	1,021	1,546
Pre-Swirl Mach #	M2.4	0.477	0.415	0.770	0.350
Pre-Swirl Angle	deg. from axial	63.21	58.83	73.29	52.14
Rotor Static Pressure Ratio	out/in	3.38	3.38	3.65	3.38
Rotor Static Temperature Ratio	out/in	1.45	1.45	1.48	1.45
Rotor Total Pressure Ratio	out/in	3.41	3.68	2.59	4.00
Rotor Total Temperature Ratio	out/in	1.46	1.47	1.39	1.49
Rig Inflow Total Temperature	deg. F.	50.00	50.00	50.00	50.00
Rig Inflow Total Pressure	psia	23	23	22.05	23

		Rampressor 1.6S		Rampressor 1	
		27.25K RPM	~29K RPM	~20.5K RPM	~31K RPM
Rotor Pre-Inlet Static Pressure	psia	19.20	19.92	14.60	20.60
Rotor Pre-Inlet Total Pressure	psia	22.43	22.43	21.61	22.43
Rotor Pre-Inlet Static Temperature	deg. F.	27.82	33.00	-4.13	37.79
Specific Heat Ratio (@exit)	"Gamma"	1.399	1.399	1.400	1.398
Outflow Mach Number	M2.7	0.488	0.546	0.266	0.611
Outflow Flow Angle	deg. from axial	-82.19	-83.02	-75.55	-83.77
Outflow Total Temp.	deg. F.	281.44	297.60	226.19	315.44
Outflow Static Temp.	deg. F.	248.10	255.46	216.70	262.17
Outflow Total Pressure	psia	76.41	82.49	55.92	89.64
Outflow Static Pressure	psia	64.97	67.41	53.25	69.73

Note: Outflow is rotor exit plane (stations 2.6 and 2.7)

Table 2.7.3-2: Rotor design parameters for various Rampressor rotor operating conditions.

	Condition	P _{2.5}	P _{2.5}	V _{2.6}	V _{2.6}	mdot-2.5	mdot-2.6	Net Thrust	Net Torque	Power Req.
Label	AR = 2.2/M=1.6	psia	psia	ft/s	ft/s	lbm/s	lbm/s	lbf	ft-lbf	hp
1	Un-started with min back P (large mdot in)	15.02	12.02	1745.68	1798.71	0.9804	0.9546	38.49	-1.80	-7.1789391
1a	Un-started with min back P (mdot const.)	15.02	12.02	1745.68	1798.71	0.9546	0.9546	39.05	-1.75	-6.9795241
4	Un-started with max back P (large mdot in)	15.02	47.94	1745.68	663.23	0.9804	0.9462	-870.39	44.51	177.51921
4a	Un-started with max back P (mdot const)	15.02	47.94	1745.68	663.23	0.9462	0.9462	-869.66	42.97	171.377228
2	Started with min back P	15.02	11.49	1745.68	1858.76	0.9804	0.9804	53.01	-4.22	-16.830624
3	Started with max back P	15.02	51.65	1745.68	653.58	0.9804	0.9804	-962.71	44.95	179.274061

Table 2.7.3-3: Various Rampressor-1 operating parameters – 20,500 RPM

	Condition	P _{2.5}	P _{2.6}	V _{2.5}	V _{2.6}	mdot _{2.5}	mdot _{2.6}	Net Thrust	Net Torque	Power Req.
Label	AR = 2.2/M=1.6	psia	psia	ft/s	ft/s	lbm/s	lbm/s	lbf	ft-lbf	hp
1	Un-started with min back P	19.20	18.70	1753.17	1713.41	1.2976	1.2087	-8.10	9.61	49.85
4	Un-started with max back P	19.20	60.09	1753.17	731.29	1.2976	1.1980	-991.73	62.03	321.85
2	Started with min back P	19.20	17.84	1753.17	1804.30	1.2976	1.2976	15.26	-1.62	-8.41
3	Started with max back P	19.20	64.97	1753.17	651.24	1.2976	1.2976	-1106.40	63.68	330.40
3	Started with max back P (mean line)	19.20	64.97	1753.17	651.24	1.354	1.22			317.23

Table 2.7.3-4: Various Rampressor 1.6S operating parameters - 27,250 RPM.

	Condition	P _{2.5}	P _{2.6}	V _{2.5}	V _{2.6}	mdot _{2.5}	mdot _{2.6}	Net Thrust	Net Torque	Power Req.
Label	AR = 2.2/M=1.6	psia	psia	ft/s	ft/s	lbm/s	lbm/s	lbf	ft-lbf	hp
1	Un-started with min back P	19.79	18.87	1748.19	1722.60	1.3193	1.2133	1.00	10.12	55.86
4	Un-started with max back P	19.79	60.64	1748.19	735.22	1.3193	1.2026	-991.66	63.03	348.00
4	Un-started with max back P (CFD result)	22.04	65.21	1152.82	770.55	1.3070	1.2650	-1047.68	61.96	342.14
2	Started with min back P	19.79	18.24	1748.19	1813.98	1.3193	1.3193	19.56	-2.44	-13.50
2	Started with min back P (CFD result)	20.06	17.31	1733.31	1775.64	1.2985	1.2712	73.67	-0.84	-4.66
3	Started with max back P	19.79	67.41	1748.19	654.56	1.3193	1.3193	-1150.40	64.38	355.50
3	Started with max back P (mean line)	19.79	67.41	1748.19	654.56	1.354	1.22			349.84

Table 2.7.3-5: Various Rampressor 1.6S operating parameters - 29,000 RPM.

Figure 2.7.3-3: Illustration of the grid types and boundary conditions used for PSN simulations.

Figure 2.7.3-4: Rampressor-1 PSN exit Mach and static pressure results for 3 different radial grid resolutions (Mach 0.77 pre-swirl design).

	Final Design	Goal	% Error
Number of Blades	68	-	-
Mass Flow (lbm/s)	3.962	3.935	+0.7
Exit Angle (@ meanline)	64.01	63.65	+0.5
Exit Mach (@ meanline)	0.496	0.477	+4.1
$P_{stat,exit}$ (mean)	19.196	19.2	-
$P_{tot,exit}$	22.71	22.43	+1.3

Table 2.7.3-6: Comparison of goal vs. final design optimization parameters.

Figure 2.7.3-5: Final PSN blade shape arrived at using optimization tools.

(a)

(b)

(c)

Figure 2.7.3-6: Volume and mean line cut contour plots of Mach number (a), static pressure (b), and total pressure (c) from the PSN designed for 27,350 RPM operation of the segmented rotor.

Figure 2.7.3-7: Illustration of how circumferential mass averaging is done at the outflow plane of the computation to arrive at radial profiles of flow properties.

Figure 2.7.3-8: Circumferentially averaged radial profiles of pressure loss, exit angle, exit Mach, and exit total pressure for the final Rampressor 1.6S PSN design.

2.8 Segmented Rotor Mechanical Design

The Rampressor 1.6S rotor mechanical design comprises of the disk, segments, cover plates, bolts, shaft and the bearing system. Of these components, the shafts and the bearing system were identical to the Rampressor 1 design and the remaining required a new design. The design speed of the Rampressor 1.6S was 27250 rpm with an over speed limit at 29000 rpm.

2.8.1 Disk

A 20 degree sector of the Rampressor 1.6S rotor system is shown in Figure 2.8.1-1. This disk has disk posts,

axial dovetail slots and bolt holes. The basic 2D Axisymmetric shape of the disk was developed in Ansys using the APDL scripts. The shape of the disk is parameterized using design variables like the coordinates of 'keypoints'. The objective of the optimization is to determine the minimum-weight disk to meet all the design objectives, as before. Since the speed of for this design was higher than that of the Rampressor 1 program, a blisk design was not possible with all the bleed holes. A more traditional rotor design with blades called segments, in this case and a disk was appropriate. Such a design allows the overall weight of the rotor system to be reduced by around 16 % compared to the Rampressor 1 rotor. Since the ability of the design to run at higher speeds is inversely proportional to the weight of the disk, but a non-linear relation, this particular disk design was chosen for achieving higher rotational speeds. Views of the rotor and one flow path set of segments is shown in Figure 2.8.1-2.

Figure 2.8.1-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.8.1-2: Views of the rotor and segments which incorporate a single aerodynamic flow path.

2.8.2 Cover Plates

The cover plate is the component that retains the segments in place. The cover plate is retained to the disk by means of bolts. Figure 2.8.1-1 shows the cover plate configuration in the 20 degree sector. Two different configurations, one retained by a snap ring (Figure 2.8.2-1) and the other retained by bolts (Figure 2.8.2-2), were analyzed for the cover plate. The snap ring design configuration analyzed had a high local contact region and since the bolted design configuration did not have any such issues, it was decided to go with the bolted design configuration. It is to be noted that given additional time, a snap ring design could have been made to work adequately. An axisymmetric analysis of the cover plate along with the disk was performed in ANSYS and the results of this analysis for the two configurations are shown in Figure 2.8.2-3 and Figure 2.8.2-4.

Figure 2.8.2-1: Cover Plate held by a retaining ring (snap ring)

Figure 2.8.2-2: Cover Plate held by a fastener (bolt)

Figure 2.8.2-3: Cover plate ANSYS analysis results for the snap ring configuration

Figure 2.8.2-4: Cover plate ANSYS analysis results for the bolted configuration

2.8.3 Rotordynamics

Based on the reduction in the mass of the rotor system, the principal moment of inertia of the rotor system decreased by 27%. This dual reduction helped in increasing the critical speeds of the rotor system allowing higher speed operating for the Rampressor 1.6 rotor system. The rotor dynamics analysis results in a rigid body mode at 35980 RPM and a compressor only damped mode of 32980 rpm. The whirl speed modes existed at 33000 rpm resulting in bearing loads excess of 8000 lbs. This will result in bearing failure at these speeds and hence the maximum safe operational speed of the system was set at 29000 rpm. The Chapter 2 Appendix contains the full rotor dynamics analysis report.

2.8.4 Segments

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2.9 Segmented Rotor Rig Integration

2.9.1 Cross Section With Segmented Rotor

A cross section of the Rampressor 1.6S rig is shown in Figure 2.9.1-1. Comparing this cross section with the Rampressor 1 cross section, excepting the wheel space shroud design and the rotor components mentioned earlier, all other component designs are identical between the two test rigs. Figure 2.9.1-2 shows the assembled rotor.

Figure 2.9.1-1: Cross section of the Rampressor 1.6S test rig.

Figure 2.9.1-2: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.10 Segmented Rotor Manufacturing, Assembly, and Tip Treatment

2.10.1 Manufacturing

The manufacturing of the rotating components for the Rampressor-1.6S consisted of the disc, segments, and cover plates (see Figure 2.10.1-1) with the assembly constituting the rotor. We shall only cover the disc and segments due to the cover plate being a relatively straightforward part.

The rotor disc and segments were manufactured from one of the Ti-6-2-4-2 forgings (Figure 2.10.1-2) originally intended for one of the Rampressor-1 monolithic rotors. Several of these forgings already have the disc shafts friction welded and were rough machined to the dimensions shown in Figure 2.10.1-3. To conserve the material as well as to take advantage of the superior forging mechanical properties and grain structure, the forging had an outer ring cut out via wire EDM to form the base material for the segments while the remainder disc with stub shafts became the rotor disc. This is illustrated in Figure 2.10.1-4, Figure 2.10.1-5, and Figure 2.10.1-6.

2.10.1.1 Disc

The disc was manufactured by Swiss Wire from the center portion of the forging left with the stub shafts after the outer ring for the segments was removed (see Figure 2.10.1-7). The rotor surfaces were precision turned before machining of the dovetail slots – in particular the lab seals and bearing journals. The dovetail slots were then machined via wire EDM to obtain the required slot geometry tolerance as well as concentricity relative to the bearing journals.

2.10.1.2 Segments

The segments were manufactured by AMT (see Figure 2.10.1-8) from the material cut out of the periphery of the same forging used for the rotor disc. The manufacturing process consisted of wire EDM of a common profile for sets of segments (dovetail, web, and stake), multi-axis machining to remove the rest of the main material, and then plunge EDM. Images of segments at various stages of the manufacturing process are shown in Figure 2.10.1-9.

Figure 2.10.1-1: Major components of the segmented rotor – disc, cover plate, and segments.

Figure 2.10.1-2: Rampressor-1 titanium forging used for 1.6S rotor (shaft stubs not yet attached via friction welding).

Figure 2.10.1-5: Drawing of segments and disc material to be taken out of the disc forging.

Figure 2.10.1-6: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.1-7: Photograph of Rampressor-16.S rotor disc during final manufacture.

Figure 2.10.1-8: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.1-9 This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.10.2 Rotor Assembly

2.10.2.1 Segment assembly

The following are the steps for segment assembly prior to installation onto the disc:

5. Assemble each flow path separately [flow path segments marked A,B,C and numbered 1-12 on the mail dovetail]
6. Starting with number 1 of each flowpath bend and install seals in both strakes and platform of the double strake segments and the strake and the high pressure side[aft] of the single strake segments –[Figure 2.10.2-1 and Figure 2.10.2-2]
7. Seals are never to be proud of the segment seal slots
8. Dry fit in adjacent segments
9. Carefully remove seal and add skim coat of RTV and reassemble
10. Continue to install seals until the entire flow path-1thru 12- is assembled.
11. Place strake alignment ring into tool [Figure 2.10.2-3]
12. Place flow paths on the assembly tool high pressure side up and align segment 1A with tool match mark 1A
13. Install last three seal sets and fit entire assembly together bringing the dovetails into contact with the half pin stops of the tool

2.10.2.2 Full rotor assembly

The following steps are for final assemble of the full rotor assembly once the segments have had seal strips installed:

1. Once all three flow paths are staged against the half pins of the tool, lower the rotor on to the tool engaging all the dovetails of the segments [Figure 2.10.2-4]
2. When the rotor is in place ensure that the segment dovetails are in full contact with the rotor dovetail slots. Set this assembly aside for the RTV to set up.
3. Place both cover plates in containers and cover with dry ice and let cool for one hour
4. Install a thin bead of RTV radially at the intersection of the segment dovetails and the rotor dovetail slots and the outside edge of the segments [Figure 2.10.2-5]
5. After plates have reached temperature, place the aft plate on the rotor assembly aligning plate match mark 1A with segment 1A and tool match-mark 1A
6. Install 5 bolts at the marked tool locations (the heads of the bolts should be on the **XX** side of the rotor)-these will tie the plate, rotor assembly and tool together
7. Remove the entire assembly from the tool base ring and place cover plate ring onto main tool ring
8. Invert rotor assembly and place on tool
9. Install the low pressure seals into segments with a small drop of RTV on the interior end
10. Remove half-pin tool plate from assembly
11. Repeat steps 4-5 using the forward cover plate
12. Install all assembly bolts and torque to 130 inch/pounds [Figure 2.10.2-6]
13. Let assembly come to room temperature and re-torque to 150inch/pounds.

Figure 2.10.2-1: Photograph of seal slot material (0.010" thick x 0.050" wide titanium) and an example bent seal prior to installation.

Figure 2.10.2-2: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.2-3: Photograph of rotor disassembled rotor assembly too..

Figure 2.10.2-4: Photograph of alignment markers and alignment pins on the rotor assembly tool.

Figure 2.10.2-5: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.2-6: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

2.10.3 Tip Machining

Prior to final machining, all 12 segments were analyzed at the anticipated rig condition (300F, 27931 rpm) to calculate the segment strake tip deflection profiles (due to CF and thermal loads) to determine the potential non-uniformity of the strake tip diameter. The numbering scheme of the 12 segments is shown in Figure 2.10.3-1. The resulting deflection profiles for all 12 segments at the anticipated operating condition are shown in Figure 2.10.3-2. Illustrations of the 3-D deflections of select segments are shown in Figure 2.10.3-3 which also indicate the tangential deflection in addition to the radial deflections. It can easily be seen that:

- Radial displacements appear to be very close to linear across strake tip
- Radial displacements of segments 1->5 are somewhat uniform across strake tip with deltas from inlet-to-exit less than 0.5 thousandths of an inch.
- Radial displacements of segments 6 - 12 become increasingly less uniform.
- Segments 8, 11 & 12 are the worst with differential radial displacements of 0.007 inch, 0.0065 inch & 0.0095 inch respectively.

With the desired maximum tip clearance between any of the strake tips and the tip ring being on the order of a couple of thousandths of an inch, these tip deflections at full speed were unacceptable. Therefore high speed grinding or individual pre-machining of the segment strake tips is required if the desired uniform tip clearance is to be achieved.

2.10.3.1 High Speed Tip Grinding

One option to achieve the desired consistent and concentric segment strake tip radius is to spin the assembled

segmented rotor at a high enough speed to deflect the segments and at this speed grind the rotor diameter to the correct/desired diameter. Prior to this final grinding process excess material would be left on the strake tips such that after grinding the correct diameter would be achieved without radius steps occurring between segments.

Typically turbine or compressor rotors are ground at high enough speed to ensure that the blade attachments are sufficiently seated or the blades have “un-wrapped” to their full extent. In addition, each blade is essentially identical and thus will have nearly the same degree of radial growth at RPM or the same radial growth with rotation rate characteristics. Thus grinding speeds do not necessarily have to be at the operational RPM. The Rampressor segment tip deflections at full speed without thermal effects (CF effects of disc plus segments only) are shown in Figure 2.10.3-4 to illustrate the different deflections of each segment. Typically grinding (rotor) rotation rates are a few thousand RPM at most, depending on the compressor/turbine design. Thus OEM tip grinding machines do not have the capability to spin a rotor greater than 3-5,000 RPM. As a result a study was performed to determine the minimum (if it existed) tip grinding speed that would produce the desired consistent segment tip radius at full operational speed. This study calculated (using ANSYS w/o using “Geometrically Non-Linear” settings to speed calculation times) the full speed (27,931 RPM) segment strake tip deflections as a function of grinding RPM. The results are shown in Figure 2.10.3-5 and indicate that to produce the desired consistent strake tip radius the tip grinding needs to be done essentially at full operational RPM. This is not necessarily true for those segments that do not have an excessive amount of twisting or tipping (i.e., segments 1-7), but those segments with two strakes prevent low speed grinding.

Since the use of an “OEM”/existing tip grinding machine was not possible a feasibility study was performed to determine if a custom tip grinding apparatus, capable of full rotor rotation speed grinding, could be constructed. The two key unknowns of this approach were: Is the material removal process (tip grinding) at these high strake tip speeds (~1,400 ft/s) physically feasible; and can a suitable machine/fixture/rig be constructed in a cost and time acceptable manner.

The grinding process unknowns included:

- Physics of grinding anything at 400m/s+ relative wheel speeds: Typical grinding or machining processes don’t even come close to these relative speeds
- What type of machining process to use: Abrasive (grinding) or single point (cutting)
- What type of abrasive to use: diamond, CBN, vitrified, etc.
- What type of abrasive geometry to use: *Stationary block of abrasive* (Simple to set up and control, but wear and heating issues thought to be significant); *Moving wheel* (distributed wear/reduced heating, but wheel speed control may be an issue)
- Coolant requirements: *Liquid* (imbalance and vacuum issues); *Gas* (this has been done, but is cooling/lubrication capability adequate)

The issue of grinding/material removal physics was investigated and it was found that “super-abrasive” or “ultra high speed” grinding processes, which employ diamond- or cubic boron nitride (CBN)-plated wheels have been demonstrated at work piece/wheel relative speeds of upwards of ~450 m/s (~1,500 ft/s) to produce net shape contours or cuts in hardened steels. At these relative speeds, the cutting physics are fairly straightforward (in fact sometimes advantageous compared with lower relative speeds) with the primary challenge being work piece/grinding wheel thermal management, grinding wheel structural integrity, and grinding machine/spindle/grinding wheel rotordynamics. Several researchers and high speed grinding companies were contacted as potential consultants and all indicated that the configuration of rotating the rotor at high speeds with a relatively stationary abrasive surface should not be a problem (technical issues certainly, but no fundamental show stoppers).

Three options/approaches were conceived to produce a machine or fixture to machine the strake tips at full speed:

- Construct a custom fixture/machine which would be capable of spinning the rotor at the desired full speed (~28,000 RPM) and performing the grinding process
- Utilize and/or adapt the Rampressor-1 test rig to perform the tip grinding process

- Utilize a combination of abradable materials on the strike tips and tip ring to simultaneously “machine” the strikes to a consistent diameter while also abrading or machining the tip ring to the desired concentricity and diameter

The last option was eventually discarded due to the inability to determine the correct combination of abradable materials. The second option was also not pursued due to: risk of damaging/destroying the test rig during the machining process due to mechanical issues or titanium fire; cost of modifying the test rig to accommodate the abrasive material; cost of taking up the test cell time to develop and perform the tip grinding; and the schedule delay produced from tying-up the test rig to develop and perform the tip grinding (no other testing or use of the test rig could be performed during this time). The first option was pursued with two companies: Test Devices Inc. (TDI) and Kinemax Engineering, PLLC.

TDI proposed building a grinding fixture to operate inside one of their spin pits that are typically used for turbomachinery proof testing and fatigue (structural and thermal loading) testing. A rough sketch of their proposed configuration is shown in Figure 2.10.3-6 and has the fixture which holds both the rotor (via high speed bearings) and the abrasive surface to the lid of the spin pit. The TDI approach was

- Rigid cylindrical fixture mounting rotor/bearings/grinding feature
- Vacuum for minimal aero loading (torque/axial) & metal fire elimination
- Spin pit for safety
- Existing high speed air drive turbine (50,000+ RPM)

With implementation issues which included:

- Bearing Lubrication (air/oil mist not practical in vacuum pit)
  - TDI has designed drip oil systems for use in the vacuum pit
- Bearing Axial Loads Generated by Grinding Process
  - Not known, but should not be large
- Torque @ Speed (Power) Requirements
  - Difficult to predict
  - They believe the power requirement will be small compared to their capability
  - Very small amount of material to be taken off at a time => small torque required
- Use of a grinding block vs. something similar to Kinemax (cup grinding wheel)
  - Vacuum-capable spindle required, but does not have to spin at very high speeds so a custom design may be doable for low cost.
  - Grinding block/spindle actuation/adjustment needs to be done as simply as possible for cost and schedule
- What type of abrasive: CBN or diamond?
- Cooling: Cold gas jet a possibility (been done before)
  - They actively pump out the spin pit during operation so they can inject a certain amount of cold gas for cooling: TBD how much or how effective it will be

The Kinemax approach, illustrated in Figure 2.10.3-7, was to assemble rented existing machinery components onto a standard grinding/machine bed, spin the rotor at full speed using a high speed machine spindle, and utilize a cupped plated grinding wheel which would be 2-axis actuated and be spun with another machine spindle. Implementation issues included:

- Potential unbalance due to fluid build-up
- Power required at speed to overcome aero torque

- Axial forces on bearings due to aero forces
- Grinding wheel spin-up prevention if grinding wheel spun in like direction to minimize effective tip speed
- Containment if something goes wrong
- Rotordynamics – Rig/Fixture Stiffness

The Kinemax approach was not pursued due to:

- Kinemax did not have good grasp of containment issue (scary!!)
- Bid very low – big underestimation of time/complexity of process
- Difficulty in predicting aerodynamic torque and axial force loadings
- Kinemax not a grinding expert
- Kinemax not a rotor dynamics expert
- Facility not owned/not controlled
- Titanium Fire Issues – High speed grinding in air w/o coolant

The TDI approach was further pursued for design, cost, and schedule estimates to determine if the high speed grinding approach was preferable to the individual strake tip machining process. Although RPS believed the TDI approach because of the inherent safety capability of performing the grinding process inside a spin pit, TDI experience with high speed bearings, rotordynamics, and instrumentation, as well as their general high level of work ethic, this approach was set aside in the short term to pursue the tip machining option due to:

- Schedule (potentially up to ~18 wks to get a ground rotor)
- Cost (~\$225K)
- Lack of High Speed Grinding Data/Experience (technical risk)
- Existing Viable Option of Predict/Machine/Measure with Western/TDI

2.10.3.2 Strake Tip Machining

The method of strake tip pre-machining is based on the assumption that the amount of segment tip deflection could be accurately modeled to within the desired tip gap tolerances and that each segment strake tip could be machined with the deflection removed (a classic “hot-to-cold” compensation) such that under CF and thermal loading a consistent strake tip diameter across all segments could be achieved. An illustration of this analysis and manufacturing process is shown in Figure 2.10.3-8. The post analysis prediction of the radial dimension profiles of the individual segment strakes is shown in Figure 2.10.3-9 and predicts that under CF loading only, all the segments will have the desired minimum variation in rotor diameters (well under 0.002”).

A fixture was designed to hold the rotor plus at least one flow path set of segments so a wire EDM process could individually machine the strake tip of each segment. This is illustrated in Figure 2.10.3-10 and shows how the rotor disc itself is part of the machining fixture, thus ensuring that the segment tips radii will be as concentric as possible relative to the rotor bearing journals. The actual machining fixture is shown in Figure 2.10.3-11.

The final machining process consisted of installing every-other segment in the rotor in a specific disc location/dovetail slot (that way any tolerance build-up would be taken care of when the segment is utilized/installed in that some slot) so that the wire of the EDM process would not contact an adjacent segment while removing material from any individual segment. The rotor with every-other segments installed is shown in Figure 2.10.3-12. Thus two set-ups were required, the first with 18 segments, the second with the remaining 18 segments. The segment dovetails were held firmly against the disc slots by using a set of aluminum wedges between the bottom of the disc slot and bottom of the segment dovetails.

The EDM machine has on-machine CMM capability and thus radial dimension measurements were made of each segment during the machining process without removing the machining fixture from the machine or the

rotor+segments from the machining fixture. The results of the segment strake tip radial profile measurements are shown in Figure 2.10.3-13 and Figure 2.10.3-14. The data indicates that, in general, the machining process was successful at producing the desired, non-CF-loaded tip radius profile prior that will produce the desired consistent strake tip radii at full rotation rate (21,250 RPM). Most of the segment radii were within +/- 0.0005" of the desired radial dimension. The exception was one of the segment #9 parts which had become loose during the machining process and thus too much material was removed on both the forward and aft strake tips, thus producing the large variation in radius from the desired dimension. However, this deviation was determined to be acceptable for initial testing of the segmented rotor.

Figure 2.10.3-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.3-2: Radial deflection profiles for each segment strake tip due to strake thermal, strake CF, disc thermal, and disc CF loading. Note segments 8 through 12 have two strakes (fwd and aft).

Figure 2.10.3-3: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.3-4: Segment 1-12 stroke tip radial deflection/grow due to disc and segment CF loading only (room temperature): Segments 1-12 FWD stroke tip deflections (top) and segments 8-12 AFT stroke tip deflections (bottom).

Figure 2.10.3-5: Effect of high speed tip grinding RPM on the resultant full speed (27,931 RPM) strake segment tip deflections – CF only due to disc and segment deflections (room temperature).

Figure 2.10.3-6: Sketch of the Test Devices, Inc. (TDI) high speed grinding approach using one of their existing spin pits.

Figure 2.10.3-7" Illustration of the Kinemax Engineering high speed grinding approach which utilized off-the-shelf components to perform the process in the bed of a convention machine base.

Figure 2.10.3-8: Illustration of the segment/disc analysis and segment stake tip machining process.

Figure 2.10.3-9: Predicted strake tip radial profiles at room temperature full speed (21,250) operating conditions.

Figure 2.10.3-10: Drawing of segment tip machining fixture with segmented rotor installed.

Figure 2.10.3-11: Photographs of the segmented rotor machining fixture prior to use.

Figure 2.10.3-12: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Rampressor 1.6s Segment Strake Tips
Radial Deviation rel nominal using data from CMM inspections
FINAL CUT

Figure 2.10.3-13: Chart of segment strake tip deviation from desired radius for each set of segments (36 in all).

Figure 2.10.3-14: Additional chart of segment strake tip deviation from desired radius for each set of segments (36 in all).

2.10.4 Tip Measurement

A test plan was developed to accurately measure the performance of the segmented rotor at operating speed. This plan utilizes data acquisition to record rotor RPM, upper bearing journal displacement, lower bearing journal displacement, a 1 per revolution signal (not shown), stroke tip displacement (upper RC20 probe), segment rim displacement (lower RC20 probe), and a signal to monitor segment leading and trailing edges (either of the RC20 probes). Test Devices, Inc. (TDI) was chosen as the facility for proof testing and tip measurement of the segmented rotor. This facility was fully capable of supporting all of our data acquisition and peripheral needs. The specifications of the measurement are as follows:

- 4 channels will be supplied by Test Devices and simultaneously recorded by Ramgen:
  - A 1 per revolution signal that will come from a probe that reads a mark placed on the arbor. The mark on the arbor will be aligned with a TBD feature on the rotor. This signal will be used to determine which segment is being seen by Ramgen's probes as well as serving as a trigger/timing signal for Ramgen's data acquisition system. The frequency response of this probe is 50 kHz. The voltage scale is from 0-10 V.
  - An upper bearing journal probe signal. The frequency response of this probe is at least 10 kHz. The output of this probe is in the 0-10 VDC range. Test Devices to provide calibration data so that Ramgen can convert this voltage to engineering units (i.e., mils gap).
  - A lower bearing journal probe signal. The frequency response of this probe is at least 10 kHz. The output of this probe is in the 0-10 VDC range. Test Devices to provide calibration data so that Ramgen can convert this voltage to engineering units (i.e., mils gap).
  - An RPM signal. The output of this probe is in the 0-10 VDC range.
- Test Devices will provide a low speed video camera that will monitor the rotor and test fixture while it is running in the spin pit. Video will be recorded.
- 3 channels will be recorded by Ramgen Power Systems and will NOT be recorded by Test Devices
  - Stroke measuring "RC" probe signal.
  - Stroke measuring "D" probe signal.
  - Segment measuring "RC" probe signal.

Figure 2.10.4-1: Tip Measurement Schematic – Side View

The RC20 probes are made by Philtec, Inc. and have both reflectance compensated and reflectance dependent capabilities. The frequency response of the reflectance compensated output is 350 KHz and is used for all displacement measurements. The reflectance dependent output is capable of 1 MHz responses and has been used to monitor the segments leading and trailing edges. In order to most effectively use the RC20 probes, the alignment

guidelines outlined in Figure 2.10.4-2 were used. As shown below, brackets were also fabricated in an attempt to aid alignment and support of the probes.

- Mount the probes so that the alignment flat is parallel to the rotor axis of rotation – failure of proper alignment will minimize voltage spiking between segments
- Mount the sensor so that it is perpendicular to within one degree of the target surface

Figure 2.10.4-2: RC20 Probe Alignment

In-house calibration was performed in order to maximize the signal to noise ratio of the RC20 outputs. A sample calibration is shown in the following figure. The “H” signal is reflectance compensated and the “D” signal is reflectance dependent. As seen on the calibration curve, for this RC20 probe to operate in the linear range of the reflectance compensated signal, it can be more than 30 mils away from the object it is measuring. However, the reflectance dependent signal peaks around 18 mils for this individual probe. This calibration curve was produced from a titanium specimen representative of the segmented rotor.

Figure 2.10.4-3: RC20 Probe Calibration for Serial Number 1394

The high speed part of the data acquisition system consisted of a National Instruments, model PCI 6110 data acquisition card with a BNC 2110 model analog to digital board was used with LabVIEW, version 6.0 software. For low speed measurements, a WaveBook 512 model was used with WaveView software, version 7.14.16. Integration of both systems for a common time base and common trigger point was accomplished through the use of the 1 per revolution signal.

Two translation stages were implemented for the measurement process. Since the stroke tip measuring probe must translate along the axial length of the rotor, a Newport, model 433-I, 60 mm translating stage along with a Zaber Technologies, Inc., model KT-LA60-SV, 60 mm linear actuator were used to get a comprehensive survey of the stroke tips radius profile. To ensure safety of the RC20 probes while the rotor is not at operating speed, a Zaber Technologies, Inc., model TSB-28, 28 mm translation stage along with a model KT-LA28-SV, 28 mm linear actuator were used to move the probes away from the rotor, as shown in Figure 2.10.4-3. In this figure, the vertical arrow indicates the path of the stroke tip measuring probe and the horizontal arrow indicates the translation of both probes moving away from the rotor.

Figure 2.10.4-4: Translation Path of RC20 Probes

The goal of the measurement process was to readings at three different axial locations per segment. One reading is to be near the leading edge of the segment, one at the middle of the segment, and one near the trailing edge of the segment. The axial locations (in inches) and corresponding segment numbers are shown below. The axial locations were measured from the leading edge of the segment, not from the leading edge of the strake. The test plan for measuring the segmented rotor was defined by the following:

- All strake tip measurements are to be made at a rotor rpm of 21,250 under ambient conditions unless otherwise specified.
- The first series of tests are to measure the radial growth at the middle of each of the segment's strake at 21,250 rpm.
- Dwell at 21,250 rpm. Measure and record strake tip radial growth as well as other peripheral test data for the forward strake of segment 8, for at least 5 revolutions.
- Translate the probe such that it is in the middle of segment 9's forward strake. Repeat the measurement process as mentioned in step 3.
- Repeat step 4 for strakes of all segments, both forward and aft strakes.
- The second series of tests involves measuring the radial growths at the leading edge of each segment strake tip. Follow steps (3) through (5) for this series of tests.
- The third series of tests involves measuring the radial growths at the trailing edge of each segment strake tip. Follow steps (3) through (5) for this series of tests.
- Repeat steps (2) through (7) at least 3 times for repeatability.

Segment	LE	Middle	TE
8fwd	0.08	0.10	0.13
9fwd	0.17	0.21	0.25
10fwd	0.30	0.33	0.36
11fwd	0.45	0.48	0.51
12fwd	0.60	0.63	0.66
1	0.74	0.76	0.78
2	0.87	0.89	0.90
3	0.99	1.01	1.03
4	1.12	1.14	1.16
5	1.24	1.26	1.28
6	1.37	1.39	1.41
7	1.49	1.51	1.53
8aft	1.62	1.64	1.66
9aft	1.73	1.76	1.78
10aft	1.86	1.88	1.92
11aft	1.99	2.01	2.05
12aft	2.15	2.18	2.21

Figure 2.10.4-5: Axial Locations for Segment Measurement

Figure 2.10.4-6: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 2.10.4-7: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Analysis of the results from testing indicated a number of the segments are tipping at operating speed. The following figure shows leading edge, middle, and trailing edge data along each of the three strake profiles. Notice the significant change in radius along forward and aft segments (segments with two strakes on them), while segments one through seven (segments with one strake) seem to be much more uniform (less tipping) along the strake profile.

Figure 2.10.4-8: Measurement Data for each Strake Profile (A, B, C)

To determine the effect that tipping segments would have on the performance of the actual test rig, the equivalent gap produced by the non-uniform radial profiles of each strake was required. From the equivalent gap, total leakage across the strake tip can then be found. As shown in the following two figures, Rout symbolizes the depth of the abradable groove created by the strake at different angular location. That is, each strake can contribute to the abradable groove, depending on the strakes radius at a given point of contact. Note that the angle also travels along the axis of rotation (the strake is a helical geometry). By knowing the profile of the abradable groove, the corresponding area gap can then be calculated for each profile at each angular location. Also, ΔR_{in} is the measured strake profile, $d\theta$ is the change in angle, and R_{i_ave} is the average radius calculated by taking the Riemann Sum of the total gap created between the abradable groove and the strake tip at each individual angular location. This calculation was performed for each strake profile, as shown below.

Figure 2.10.4-9: Designation of Nomenclature

Figure 2.10.4-10: Normalized Data Per Flow Path (A, B, C)

Once the abradable gap for each profile was determined, the calculation was then taken one step further to estimate the equivalent gap created by all three profiles. For the below figure, "AveGap" is the average gap created by averaging all three profiles. "MaxGap" is the maximum gap possible, produced by subtracting the smallest radius from the largest radius at each angular location from the entire data set (includes all three stroke profiles). "EquivGap" is a single value that estimates the equivalent gap across the entire rotor between the stroke tip and the abradable.

Figure 2.10.4-11: Normalized Data for Entire Flow Path

As shown by the previous figures, the data produced by this high speed measurement technique is highly repeatable and has captured the degree of resolution required for in depth analysis of the strake tip profiles. Analysis of the tip measurement data revealed an equivalent tip gap near 2.29 mils. Balancing of the segments is paramount in minimized segment tipping. With the single strake segments tipping much less than the dual straked segments, it is evident that more care must be taken to minimize this effect. To better understand why this is happening, more testing is necessary. Possible tests include operating the rotor without seals and RTV between the segments. This would clearly show how the segments interact with each other. Test Devices, Inc. proved to be quite proficient for this type of testing and further testing of the segmented rotor would be encouraged to take place at their facility.

2.10.5 Proof Testing

Proof testing was performed by Test Devices, Inc. (TDI). The process of proof testing consisted of pressing an arbor onto the segmented rotor, balancing the rotor and arbor, and then proof testing. The arbor was manually pressed on to the rotor by chilling the rotor with liquid nitrogen and heating the arbor. This procedure is extremely efficient and relatively simple.

Figure 2.10.5-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Balancing the rotor and arbor proved to be the most intensive phase of the proof test. The segments can easily shift relative to each other and much care must be taken in balancing. It was found that driving the rotor from the smallest possible radius provided the most repeatable and reliable readings. Before balancing, the segments were thoroughly cleaned and individually inspected. All excess RTV was removed in an attempt to create a uniform RTV bead across the segmented rotor.

Figure 2.10.5-2: Balancing the Arbor and Rotor at Test Devices, Inc.

Once balancing was completed, the rotor and arbor were then attached to the spin pit lid. The proof test plan followed these guidelines:

- One cycle up to 23,375 RPM @ ambient temperature
- Record vibration data up and down cycle as well as during dwell time
- Dwell time of two (2) minutes at maximum rpm (23,375 RPM)
- Low speed video through cycle

The proof test of the segmented rotor confirmed the integrity of the disk and rotor. As shown in the figure, the rotor appears to be well balanced and resonant speeds were minimal near the operating speed range.

Figure 2.10.5-3: Proof Test Configuration

Figure 2.10.5-4: Proof Test Results

CHAPTER 2 APPENDIX

Appendix

Summary:

This document summarizes the analyses/procedures used to determine the material for the tip shroud ring. Since the inlet temperature can vary anywhere between 50 °F to 90 °F, the material that is chosen for minimum tip clearance at design point must be capable of holding minimum clearances at different inlet temperatures, at design point operation. In order to meet this objective, six different materials were analyzed by considering different inlet temperatures and different cooling air temperatures. The recommended material for meeting the tip clearance objective is Aluminum – 6061 – T6. The cooling air requirement for meeting the design point tip clearance objective is 1.5 lb_m/s at 31 psia and –50 °F, in the impingement cavity (the cavity that is immediately upstream of the impingement plate). To address the off-design points, it is recommended that the cooling air temperature requirement be supplied at – 50 °F (based on extrapolation of the results at design point), to provide additional margin.

Objective:

The primary objective is to maintain minimum tip clearance at full load steady state operation. At the same time, the temperature gradient in the tip shroud should be kept reasonable to maintain the concentricity with the rotor tips.

Assumptions:

The nominal radial cold build clearance between the rotor tip and the tip shroud is assumed to be 0.005 inches. Any eccentricity is assumed to be addressed with the help of rub tolerance coatings applied on the tip shroud. Meanline aero conditions were used as inputs to generate the boundary conditions.

Discussion:

The different materials analyzed with the corresponding inlet and cooling air conditions is shown in the Table 1.

The results of the thermal / displacement analyses carried out at steady state (full load, 100% speed) condition using Ansys are shown in the Figures 1 through 44 (refer the powerpoint file attached at the end of this document). Figures 1 through 20 show the expected thermal gradients at steady state for the static structure, and Figures 21 through 40 show the expected displacements at steady state for the static structure. Figures 41 through 44 show the thermal and displacement results for the rotor. All temperatures reported in the above mentioned figures are in degrees Fahrenheit and all displacements are in inches.

The rig inlet temperature, the material analyzed for the tip shroud and the cooling air temperature are reported at the bottom left hand corner of each of the attached figures.

The plots are zoomed in to that part of the static structure that has the most immediate effect on the tip clearance.

Table 1. Analyses performed for tip shroud material selection.

Tip Shroud Materials	Rig Inlet Conditions (deg. F)	Cooling Air Temperatures (deg. F)
INCO 903	90	50
		-50
	50	50
		-50
WC-6	-	-
		-
	50	50
		-50
D5B	90	50
		-50
	50	50
		-50
Al - 6061 -T6	-	-
		-
	50	-25
		-50
A536	90	50
		-50
	50	50
		-50
17-4PH	90	50
		-50
	50	50
		-50

For a 90 °F inlet day, the rotor tip is expected to grow about 0.007 inches radially (Figure 41) and since the cold build clearance is 0.005 inches, for a line to line contact, the tip shroud can grow at most 0.003 inches for 0.001 inch tip clearance, which from the results is achieved by almost all the materials that have been considered. The caveat to this is that the thermal gradient across the tip shroud becomes an issue and hence, A536 would be the preferred tip shroud material. For a 50 °F inlet day, not only does the aft temperature go down but the centrifugal growth of the disk reduces too, as the speed drops. This makes achieving the tip clearance goal that much more challenging. Though the analyses indicate that a 50 °F cooling air is capable of causing a 0.002 inch tip clearance gap, when A536 material is used for tip clearance (compare figures 30 and 42), there is no margin for error on issues such as tolerance stack ups etc., in this design. In order to alleviate such concerns, a much cooler air source is required. In addition, a material that has a better thermal expansion is preferred because this ensures that a slight reduction/increase in metal temperature produces a relatively higher yield on the radial contraction/growth of the tip shroud ring. Aluminum 6061 addresses these material concerns and hence is the preferred material for the tip shroud ring.

Figure 1. Predicted Metal Temperature for a 90 °F Inlet day, Material 17-4PH, -50 °F Cooling Air.

Figure 2. Predicted Metal Temperature for a 90 °F Inlet day, Material A536, -50 °F Cooling Air.

Figure 3. Predicted Metal Temperature for a 90 °F Inlet day, Material D5B, -50 °F Cooling Air.

Figure 4. Predicted Metal Temperature for a 90 °F Inlet day, Material INCO 903, -50 °F Cooling Air.

Figure 5. Predicted Metal Temperature for a 90 °F Inlet day, INCO 903, 50 °F Cooling Air.

Figure 6. Predicted Metal Temperature for a 90 °F Inlet day, D5B, 50 °F Cooling Air.

Figure 7. Predicted Metal Temperature for a 90 °F Inlet day, A 536, 50 °F Cooling Air.

Figure 8. Predicted Metal Temperature for a 90 °F Inlet day, 17-4PH, 50 °F Cooling Air.

Figure 9. Predicted Radial Displacement for a 90 °F Inlet day, Material 17-4PH, -50 °F Cooling Air.

Figure 10. Predicted Radial Displacement for a 90 °F Inlet day, Material D5B, -50 °F Cooling Air.

Figure 11. Predicted Radial Displacement for a 90 °F Inlet day, Material A 536, -50 °F Cooling Air.

Figure 12. Predicted Radial Displacement for a 90 °F Inlet day, Material INCO 903, -50 °F Cooling Air.

Figure 13. Predicted Radial Displacement for a 90 °F Inlet day, Material INCO 903, 50 °F Cooling Air.

Figure 14. Predicted Radial Displacement for a 90 °F Inlet day, Material D5B, 50 °F Cooling Air.

Figure 15. Predicted Radial Displacement for a 90 °F Inlet day, Material A 536, 50 °F Cooling Air.

Figure 16. Predicted Radial Displacement for a 90 °F Inlet day, Material 17-4 PH, 50 °F Cooling Air.

Figure 17. Predicted Metal Temperature for a 50 °F Inlet day, WC 6, 50 °F Cooling Air.

Figure 18. Predicted Metal Temperature for a 50 °F Inlet day, INCO 903, 50 °F Cooling Air.

Figure 19. Predicted Metal Temperature for a 50 °F Inlet day, A536, 50 °F Cooling Air.

Figure 20. Predicted Metal Temperature for a 50 °F Inlet day, D5B, 50 °F Cooling Air.

Figure 21. Predicted Metal Temperature for a 50 °F Inlet day, 17-4PH, 50 °F Cooling Air.

Figure 22. Predicted Metal Temperature for a 50 °F Inlet day, 17-4PH, -50 °F Cooling Air.

Figure 23. Predicted Metal Temperature for a 50 °F Inlet day, WC 6, -50 °F Cooling Air.

Figure 24. Predicted Metal Temperature for a 50 °F Inlet day, INCO 903, -50 °F Cooling Air.

Figure 25. Predicted Metal Temperature for a 50 °F Inlet day, D5B, -50 °F Cooling Air.

Figure 26. Predicted Metal Temperature for a 50 °F Inlet day, D5B,- 50 °F Cooling Air.

Figure 27. Predicted Metal Temperature for a 50 °F Inlet day, A536, -50 °F Cooling Air.

Figure 28. Predicted Radial Displacement for a 50 °F Inlet day, Material 17-4PH, 50 °F Cooling Air.

Figure 29. Predicted Radial Displacement for a 50 °F Inlet day, Material 17-4PH, -50 °F Cooling Air.

Figure 30. Predicted Radial Displacement for a 50 °F Inlet day, Material A536, 50 °F Cooling Air.

Figure 31. Predicted Radial Displacement for a 50 °F Inlet day, Material D5B, -50 °F Cooling Air.

Figure 32. Predicted Radial Displacement for a 50 °F Inlet day, Material D5B, 50 °F Cooling Air.

Figure 33. Predicted Radial Displacement for a 50 °F Inlet day, Material INCO 903, 50 °F Cooling Air.

Figure 34. Predicted Radial Displacement for a 50 °F Inlet day, Material INCO 903, -50 °F Cooling Air.

Figure 35. Predicted Radial Displacement for a 50 °F Inlet day, Material WC 6, -50 °F Cooling Air.

Figure 36. Predicted Radial Displacement for a 50 °F Inlet day, Material WC-6, 50 °F Cooling Air.

Figure 37. Predicted Metal Temperature for a 50 °F Inlet day, Al 6061, -25 °F Cooling Air.

Figure 38. Predicted Metal Temperature for a 50 °F Inlet day, Al 6061, -50 °F Cooling Air.

Figure 39. Radial Growth for a 50 °F Inlet day, Material Al 6061, -25 °F Cooling Air.

Figure 40. Radial Growth for a 50 °F Inlet day, Material Al 6061, -50 °F Cooling Air.

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Figure 42. Predicted Transient Radial Growth for a 90 °F Inlet day.

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Energy Policy Act of 1992**

Tip Clearance Control - Summary

Active Tip Clearance Control (ATCC)

- Final Requirements from the Facility
  - 1.0 lb_m/s 30 psia, ~220 °F Compressed Air Source for Heating the Tip Shroud.
  - ~ 1.5 lb_m/s 30 psia, -30 °F Compressed Air Source for Cooling the Tip Shroud
    - *Note: Above conditions are in the Plenum immediately upstream of the impingement shields.*
    - *Includes a 25 % margin on top of what is required.*

Analysis Performed

- 90 °F Inlet Conditions.
- 50 °F Inlet Conditions.

Design Challenges

- Worst Case for tip shroud cooling is for a 50 °F Inlet day.
- Also need to consider off-design operating conditions.

Thermal Analysis

- **Tip Ring Heat Input**

- Calculate the aero conditions based on the absolute reference frame. (remove the $r\omega$ term from the data predicted by Ryan's performance prediction indicates)
  - Compared it with the CFD simulation plots that Ryan generated from the adapco inviscid sims.
- Heat load is calculated based on existing correlations for flow over a flat plate and the local adiabatic film temperature.
  - Compared this against a duct flow calc & swirling flow calc.

- **Tip Ring Cooling Load**

- Calculated based on data available for impingement cooling schemes.
  - High degree of confidence on this data, from prior experience.

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the Energy Policy Act of 1992**

Flow Path HTC's Applied To The Tip Shroud

Estimated Uncertainty Of Heat Load In Different Regions

- Estimated error based on comparison with duct flow, swirling flow and other flat plate theories.

Tip Shroud Cold Side Model

- **Impingement Cool the back side so that we get as uniform a temperature as possible on the tip shroud.**
  - Important to minimize the thermal gradient across the tip shroud.
    - So choose a material with suitable thermal conductivity.
  - **Minimize radial growth distortions.**
    - So choose a material with good thermal coefficient of expansion.

Tip Shroud Materials Considered

Tip Shroud Materials	Rig Inlet Conditions (deg. F)	Cooling Air Temperatures (deg. F)
INCO 903	90	50
		-90
	60	50
		-90
WC-6	-	-
		-
	50	50
		-90
Q5B	90	50
		-90
	50	50
		-90
Al - 6061-T6	-	-
		-
	50	-25
		-90
A536	90	50
		-90
	50	50
		-90
17-4PH	90	50
		-90
	50	50
		-90

Tip Shroud Hot Side Heater Tape Estimate

Assumptions

- Switch on the heater about 150 seconds before the rig operates or before the rig is shut down.

Heater Requirement Calculation

- Estimate the mass to be heated.
- Estimate the tip shroud temperature required to avoid interference.
- Using $Q = mC_p(\Delta T)$, we can estimate the heater power requirement and with an additional 30% margin, it comes to about a 1.2KW heater.

Concerns with the heater

- Possible contact resistance issues might interfere with the cooling mechanism.

Recommended only if there is huge hassle with plumbing the hot air lines.

Tip Clearance Control Analysis Results

- **Active Tip Clearance Control (ATCC)**
  - ***Assumption***
    - ***Cold Build Tip Gap of 0.005 inches (Nominal).***
      - ***Gap between the strake tip and the abradable coating on the tip shroud surface.***
  - **Analysis based on a 50 °F Inlet Day.**
    - **2D Axisymmetric.**
    - **Titanium Rotor**
    - **Al 6061 Tip Shroud**
    - **Impingement Heating/Cooling**

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Policy Act of 1992**

Static Structure Model

Tip Clearance Control

- Transient Starting Cycle that was analyzed.

Time (hrs)	Time (secs)	Rotor RPM
0.000	0	0
0.008	29	4075
0.016	59	4075
0.024	88	8150
0.033	118	8150
0.043	153	13000
0.051	183	13000
0.066	236	20373
0.132	476	20373
0.299	1076	20373
1.327	4776	20373
1.331	4791	20373
1.364	4911	8000
1.420	5111	2000
1.503	5411	0

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Full Load Static Structure Temperature Distribution

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Full Load Static Structure Radial Growth Distribution

Transient Rotor Radial Growth Distribution

1

POST26

RotorTip
ODRimSeal
IDRimSeal
hubSeal
DiskCore
RotorRim

ANSYS

MAR 25 2003

10:32:32

TIME AT THE END OF THIS LOAD STEP =%

Transient Static Structure Radial Growth Distribution

Estimated Transient Strake Tip / Tip Shroud Radial Growth Profile

Estimated Transient Tip Gap Profile

Tip Clearance Control Analysis Results

- **0.005 inch nominal cold radial gap**
  - **results in an interference of 0.002 inches to 0.004 inches at full load operation.**
    - **The forward part of the tip shroud contracts more than the aft part of the tip shroud due to the temperature gradient along the axial direction of the tip shroud.**
      - **Heat load from the flow path changes as we go from the inlet to the diffuser section of the rotor flow path.**
  - **This difference in growths + eccentricity / stack up tolerances etc., will be taken care of by the abradable coating applied on the tip shroud.**

RAMGEN SUPERSONIC COMPRESSOR **RIG BEARING ANALYSES**

SUMMARY

Computer analyses with the "Jones" and, to a lesser extent, with the "Shaberth" computer codes were carried out to assess the viability of the use of NHBB 52100 steel 1108-size angular contact bearings in the Ramgen supersonic compressor rig. These studies showed that the maximum total load on a single bearing at the initial operating speed of 21,000 rpm was about 1200 lb. to give a minimum of 100 hr. B.10 life. At 28,034 rpm this load dropped slightly to about 1100 lb.

If the rotor thrust load is maintained as low as possible and both bearings are subjected to as high a bearing system preload consistent with the total load limit (on the #2 bearing), the (statistical) life of the two-bearing system is, however, reduced below 100 hours. The 1000 lb.+ impact loads experienced during operation were found acceptable, producing Hertzian contact stresses below the bearing material elastic limit in compression. Bearing radial stiffness data were developed in an effort to help explain an earlier bearing failure and to ensure that different bearing/operation situations were satisfactory in that respect.

To obtain higher load/preload capability or B.10 lives, the substitution of bearings made of better material would be advisable. The use of "hybrid" bearings (with silicon nitride balls) would not be necessary. The latter have, however, proved repeatedly in operation involving failure situations to produce much more benign failures, including much less collateral damage, than the more usual all-steel versions.

PROCEDURE

With the cross-section drawing of the compressor rig, models for the "Jones" and, later, the "Shaberth" computer codes were formulated. The "Jones" code is particularly suited to evaluating preloaded bearing systems with added thrust and radial loads. It does not consider lubrication effects (as does "Shaberth"). The effect of different factors affecting computed life, such as material, material processing, and lubrication, were input as a modified fatigue constant to provide a "tidy" printout of each solution. The #1 was nominated as the "floating" bearing next to the preload spring. The code also produces computed system radial and axial stiffnesses the former of which were extracted as extremely useful for rotor dynamics calculations. (Inclusion of lubrication effects have been found over the years not to affect significantly stiffnesses computed with the "Jones" code.) Mean Hertzian stresses calculated at the inner race (I/R) and outer race (O/R) contacts of both bearings were also extracted, inspected at each speed/loading condition for the maximum value in each case and this converted to the "peak" value by multiplying by 1.273 (in ball bearings) for comparison to the material elastic limit of about 330,000 psi.

To ensure that the life factors mentioned above were as accurate as possible, NHBB was consulted for the material processing of these particular bearings and a basic model of the bearing system was derived for the computer code "Shaberth" to permit derivation of a lubrication life factor resulting from the oil film thicknesses in the I/R and O/R contacts mentioned above. Operating temperature of all components of each bearing was assumed to be the reported 160F with oil-in temperature 100F at 21,000 rpm. It was also reported that there was no significant external heat input into the bearing system. At 28,034 rpm the temperatures were estimated to be 200F and 110F respectively.

Cases for various thrust loads and preloads up to 1500 lb. for Conditions 1 and 2 (21000 and 28034 rpm respectively) were studied to establish values that could be tolerated for a minimum of 100 hours life. This is interpreted as the "B.10" life - the life in hours under specified loading and

operating conditions which 90% of a population of nominally identical bearings will exceed. Stated alternatively, this is the life by which 10% will have failed, in classical fatigue - usually more or less a heavy-load phenomenon. Some cases were run at the 25 degree contact angle of the bearings used initially, one of which failed, primarily to develop radial stiffnesses for comparison with those of bearings which have operated successfully with 15 degrees.

Additionally, a case was run at 21,000 rpm with 200 lb. preload and 2500 lb. thrust to study the effect of such an impact load. For very high loads care has to be taken not to cause the ball-race contacts to run over the edges of the raceways. Both inner and outer contact angles increase with load and for 2700 lb. total load the contact area extreme angles computed were 44.7 degrees at the inner and 43.4 at the outer. NHBB bearing drawing dimensions suggest that the maximum (allowable) angles are about 60 degrees at the inner and 55 at the outer.

Further, a ("hybrid") bearing of Cronidur 30 through-hardened stainless steel with silicon nitride balls at 28,034 rpm under the 200 lb. preload and 200 to 1500 lb. thrust, with the same geometry, was studied using the "Jones" code to evaluate stiffness characteristics. As expected, the hybrid was substantially stiffer than the all-steel bearing at each load condition.

These computed data are shown in Tables 1 to 4.

Failure scenarios and their detection were discussed.

CONCLUSIONS

1. Up to a total load on the (more heavily loaded) #2 bearing, that is both preload + thrust load, of 1200 lb. a B.10 life equal to or greater than 100 hr. is obtained.
2. When the bearing system preload is increased, the life of the #1 bearing is reduced commensurately and the life of the two bearings considered as a system is statistically less than the lower of the two, constituting an overall reliability factor. For a system B.10 life of 100 hr. with a thrust load of 200 lb., a preload only a little more than 800 lb. can be sustained.
3. Up to a total load on the #2 bearing, preload + thrust load, of 1700 lb. the greatest "peak" Hertzian contact stress, which occurred at the I/R, is equal to or less than the 52100 steel compressive yield stress of about 330,000 psi and the effect of any such impact stress should be benign.
4. For an impact (thrust) load of 2500 lb., the above stress criterion is exceeded by about 46,000 psi. However, the load-contact areas do not run over the (theoretical) edges of the raceways and the overload should be tolerated. Cage stresses could be problematical if any shaft misalignment occurs during the impact loading. This would result in high ball to cage loads which vary from ball to ball and create significant "hoop" tensile stresses in the cage.
5. Because the impact stresses under consideration are of only a few seconds' duration, they will have an insignificant effect on computed B.10 life.
6. Significant impact load could cause fracture of the bearing cage and result in bearing failure but, particularly since this has not happened in operation to date, evidently the present load magnitude is insufficient to result in this.
7. A radial stiffness of the #1 bearing of 475,000 psi with 200 lb. preload at 25 degrees contact angle and 21,000 rpm was found substantially less than 1.2×10^6 psi for a bearing with a 15 degree contact angle. This would tend to dictate a minimum preload of 300 or 400 lb. for operation of 15 degree bearings at 28,034 rpm because a 200 lb. preload at that speed produces a stiffness of only 882,500 psi. These findings are presented for their possible effects on rotor dynamics.
8. At the Condition 2 higher speed of 28,034 rpm the allowable load for 100 hr. minimum life was a little lower: about 1100 lb. total, preload + thrust.
9. At Condition 2 the impact load limit for no significant effect on life and failure was the same,

1700 lb. total preload + thrust.

10. At condition 2 the system B.10 life of 100 hr. preload, as in (2) above, is reduced to a little less than 800 lb.
11. If a total of preload + thrust (on the more heavily loaded #2 bearing) more than outlined above is required, it would be advisable to procure bearings of improved material, for example the Barden/FAG-Germany *Cronidur 30* through-hardened steel version supposedly readily available with both steel and silicon nitride balls. The B.10 life would be at least 3.33 times the NHBB bearings and the compressive yield stress would be at least 20,000 or 30,000 psi greater.
12. The greater stiffness characteristics of a hybrid bearing, otherwise identical to an all-steel version, allow geometrical re-optimization. By reducing the stiffnesses to existing values, increased lives can be obtained.
13. Equipment/scenarios related to failure might include the following.
  - (a) Install chip detectors, ideally electric, real time-indicating, in both bearing oil drain lines to permit detection of any bearing magnetic material particles which might be generated by fatigue failure of the raceway. If, on the other hand, a skidding-type failure were to occur, generally because of too little load at high speed, very fine "fuzz" (wear) particles would be produced and probably detected only by inspection at regular intervals to permit recording the frequency/amount and categorize them as increasing (a problem) or decreasing (not a problem). Too little load in a preloaded system can result when the bearing which is supposed to float axially hangs up and upsets the load sharing and when the thrust load exceeds the preload in the wrong direction and the whole system moves axially. Obviously though, if the "upset" is drastic, the failure will be "drastic" too.
  - (b) Affix thermocouples to the outer races of both bearings, either to the outside diameter or the end face, to permit continuous recording of operating temperatures to permit establishing values for normal operation, whether there is any gradual increase with operating time, either during the course of a single run or with time. A pronounced temperature spike, for instance, most likely is indicative of a raceway fatigue failure (and would be accompanied by debris on the affected chip detector as well as increased vibration.) A longer-term increase in operating temperature, on the other hand, could indicate a wear-type failure (and should also show up as a parallel increase in vibration.) The characterization as "long" *can* be minutes rather than hours with high speed bearings with light enough loads. Redundancy is desirable.
  - (c) Install pairs of proximity probes, at 90 degrees, at "live" spots, say, at the extreme ends of the shaft, and in a position to detect disc axial movement. The shaft probes can be X-Y displayed on an oscilloscope to provide a ready indication of rotor motion. The disc probe will indicate if the preload arrangement is behaving as intended and can permit "calibration" of the impact load by comparing its output to computed bearing displacement data.
  - (d) Arrange continuous recording of velocimeter output for bearing "health" determination also.
  - (e) Install comparatively coarse strainers (in addition to existing much finer oil system filtration) in oil drain lines to catch any (large) non-magnetic debris such as would be generated during bearing failure by break-up of the cage or (in the very unlikely event) ceramic balls in a "hybrid" bearing.
  - (f) Save debris removed from the oil system filter(s) and record date, hr. of operation, etc. for each sample for correlation with other reliability data.
  - (g) While the analyses carried out show no indication that "hybrid" bearings would

provide any life advantage here without re-optimization of the geometry, there has been much evidence developed over the years in this author's personal experience which shows that "hybrid" bearings (with silicon nitride balls) substantially improve failure behavior, particularly of preloaded systems. Even when the cage has suffered multiple fractures and completely disappeared during a bearing total failure, the ceramic balls continue to provide adequate radial support for the rotor to avoid much of the collateral damage usually associated with the mangled, mishapen, welded mass remaining of an all-steel bearing.

Tables 1 through 4 follow on pages 5 and 6.

TABLE 1 - RAMGEN SUPERSONIC COMPRESSOR RIG BEARINGS ANALYSIS

Preload lb.	Radial Load lb.	Thrust Load lb.	Life #1 hr.	Load #2 lb.	Life #2 hr.	System Life hr.	Hertz Stress #2 I/R		Radial Stiffnesses lb/in			
							Mean, psi	Max, psi	Vertical	#1 Horiz	Vertical	#2 Horiz
CONDITION 1 - 21,000 RPM			NHBB 108-SIZE 52100 BRG									
200	45	200	16402	400	2289	2080	169110	215277	1.18 E6	1.18 E6	1.75 E6	1.75 E6
200	45	400	16425	600	740	720	191190	243385	1.18 E6	1.18 E6	2.05 E6	2.05 E6
200	45	600	16444	800	337	333	208260	265115	1.18 E6	1.18 E6	2.24 E6	2.24 E6
200	45	800	16460	1000	184	183	222340	283039	1.18 E6	1.18 E6	2.39 E6	2.39 E6
200	45	1000	16474	1200	113	113	234430	298429	1.18 E6	1.18 E6	2.52 E6	2.52 E6
200	45	1200	16487	1400	75.1	74.8	247070	314520	1.18 E6	1.18 E6	2.62 E6	2.62 E6
200	45	1500	16504	1700	44.9	44.9	259040	329758	1.18 E6	1.18 E6	2.76 E6	2.76 E6
200	45	2500	16550	2700	13.4	13.4	295140	375713	1.18 E6	1.18 E6	3.08 E6	3.08 E6
400	45	200	2296	600	740	591	191210	243410	1.76 E6	1.76 E6	2.05 E6	2.05 E6
500	45	200	1230	700	484	368	200210	254867	1.92 E6	1.92 E6	2.16 E6	2.16 E6
600	45	200	742	800	337	246	208290	265153	2.05 E6	2.05 E6	2.25 E6	2.25 E6
800	45	200	338	1000	184	127	222370	283077	2.25 E6	2.25 E6	2.40 E6	2.40 E6
1000	45	200	185	1200	113	74.9	234460	298468	2.40 E6	2.40 E6	2.53 E6	2.53 E6
1200	45	200	113	1400	75	48.2	245100	312012	2.53 E6	2.53 E6	2.63 E6	2.63 E6
1500	45	200	62.7	1700	44.9	28	259080	329809	2.68 E6	2.68 E6	2.77 E6	2.77 E6
CONDITION 2 - 28,034 RPM			NHBB 108-SIZE 52100 BRG									
200	37	200	11786	400	1769	1595	165570	210771	8.82 E5	8.82 E5	1.54 E6	1.54 E6
200	37	400	11802	600	569	552	188060	239400	8.82 E5	8.82 E5	1.91 E6	1.91 E6
200	37	600	11815	800	256	253	205460	261551	8.82 E5	8.82 E5	2.15 E6	2.15 E6
200	37	800	11826	1000	139	138	219800	279805	8.82 E5	8.82 E5	2.33 E6	2.33 E6
200	37	1000	11835	1200	84.7	84.4	232090	295451	8.82 E5	8.82 E5	2.46 E6	2.46 E6
200	37	1200	11844	1400	55.9	55.8	242890	309199	8.82 E5	8.82 E5	2.58 E6	2.58 E6
200	37	1500	11855	1700	33.3	33.3	257070	327250	8.82 E5	8.82 E5	2.72 E6	2.72 E6
400	37	200	1782	600	569	455	188070	239413	1.54 E6	1.54 E6	1.92 E6	1.92 E6
600	37	200	573	800	256	188	205480	261576	1.92 E6	1.92 E6	2.16 E6	2.16 E6
800	37	200	258	1000	139	96.3	219820	279831	2.16 E6	2.16 E6	2.34 E6	2.34 E6
1000	37	200	140	1200	84.6	56.3	232110	295476	2.34 E6	2.34 E6	2.48 E6	2.48 E6
1200	37	200	85.3	1400	55.9	36.1	242920	309237	2.48 E6	2.48 E6	2.60 E6	2.60 E6
1500	37	200	46.8	1700	33.2	20.8	257100	327288	2.65 E6	2.65 E6	2.74 E6	2.74 E6
CONDITION 2 - 28,034 RPM			HYBRID 108-SIZE BRG									
			<i>Lives and stresses approximate</i>									
200	37	200	17298	400	2390	2174	194150	247153	1.49 E6	1.49 E6	2.08 E6	2.08 E6
200	37	400	17320	600	774	753	219450	279360	1.49 E6	1.49 E6	2.39 E6	2.39 E6
200	37	600	17339	800	353	349	239000	304247	1.49 E6	1.49 E6	2.60 E6	2.60 E6
200	37	800	17355	1000	193	192	255150	324806	1.49 E6	1.49 E6	2.77 E6	2.77 E6
200	37	1000	17369	1200	119	118	269010	342450	1.49 E6	1.49 E6	2.91 E6	2.91 E6
200	37	1200	17382	1400	78.8	78.6	281220	357993	1.49 E6	1.49 E6	3.03 E6	3.03 E6
200	37	1500	17399	1700	47.2	47.1	297260	378412	1.49 E6	1.49 E6	3.19 E6	3.19 E6

**TABLE 2 - # 2 BEARING CONTACT ANGLES
UNDER 2700 LB. LOAD**

	Inner Race	Outer Race
Upper Extremity	44.7	43.4
Mean Contact Angle	26.4	25.3
Lower Extremity	8.2	7.3

**TABLE 3 - LUBRICATION LIFE MULTIPLYING
FACTORS FROM "SHABERTH"**

Speed rpm	Preload lb	#1 Bearing		#2 Bearing	
		Outer Race	Inner Race	Outer Race	Inner Race
21000	200	2.65	2.62	2.64	2.61
21000	1500	2.61	2.57	2.61	2.57
28034	200	2.57	2.52	2.55	2.49
28034	1500	2.49	2.41	2.49	2.40

Lubricant properties per MIL-PRF-23699 turbine oil

**TABLE 4 - MATERIAL LIFE MULTIPLYING
FOR "JONES" & "SHABERTH" CODES**

52100 Bearing Steel per AMS 6440	3
Consumable Vacuum Degas (CVD) Processing	1.5

Pre Swirl Nozzle Bleed Design

Summary:

This document summarizes the maximum amount of bleed flow that can be extracted at design point in the exit region of the pre-swirl nozzle. This is expected to provide the aero group with the limitations of each of the different designs so that a final design configuration is chosen, by considering both the mechanical design limitations as well as the aero design limitations.

Discussion:

Inputs from Aero [1] that were made use of for this analysis are

Pt = 21.6 psia.

Tt = 50 °F.

Ps = 15.02 psia.

M = 0.74

Pre-swirl ID bleed mass flow rate = 0.087 lbm/s.

Pre-swirl OD bleed mass flow rate = 0.069 lbm/s.

A brief analytical trade study was conducted to estimate the maximum mass flow rate that can be bled from the pre-swirl region, while maintaining the same (exhaust) bleed duct dimensions throughout the study (3 pipes of 0.3" diameter each). Only one of the two walls (Inner Diameter, in this case) was considered for this study. The parameters that were varied were

- (i) the number of bleed holes.
- (ii) the angle of the holes.
- (iii) the Mach Number in the bleed duct.

Since only one wall was considered for this study, the way to interpret the results from this study is to multiply the reported bleed mass flow rate and the number of holes (see attached table) by two to arrive at the overall bleed capability that is built into the current rig layout.

As is evident from the results, to meet the pre-swirl ID bleed requirement of 0.087 lbm/s bleed requirement condition, we would need a large number of 20° angled holes (see the last three rows of the attached Table) and in addition, the exhaust bleed duct on the ID will have to be operating at choked conditions (this poses a major challenge).

The attached table also suggests that with the current rig out, bleeding approximately 0.040 lbm/s on the ID can be achieved, while keeping the bleed duct Mach Numbers around 0.5.

Conclusion:

The purpose of this document is to establish the bounds within which the pre-swirl nozzle bleed flow can be varied. It is recommended that the aero/mechanical group decide on a suitable design based on the alternatives provided.

Table 3-1. Estimated PSN bleed flows on the ID of the static flow path, forward of the rotor.

Approximate Mach Number in the Bleed Duct	Number of Holes	20 Degree Holes		40 Degree Holes		90 Degree Holes	
		Estimated Mass flow rate that can be bled (lbm/s)	Approximate Static Pressure at the exit of the bleed duct	Estimated Mass flow rate that can be bled (lbm/s)	Approximate Static Pressure at the exit of the bleed duct	Estimated Mass flow rate that can be bled (lbm/s)	Approximate Static Pressure at the exit of the bleed duct
-0.2	400	0.0159	13.4	0.0145	12.1	0.0136	11.3
	800	0.0160	13.6	0.0146	12.2	0.0139	11.6
	1200	0.0160	13.6	0.0147	12.3	0.0140	11.7
	1600	0.0160	13.6	0.0147	12.3	0.0141	11.7
-0.3	400	0.0236	12.8	0.0216	11.5	0.0200	10.5
	800	0.0239	13.0	0.0218	11.7	0.0207	11.0
	1200	0.0240	13.0	0.0219	11.8	0.0209	11.1
	1600	0.0240	13.1	0.0220	11.8	0.0210	11.2
-0.5	400	0.0385	11.5	0.0352	10.4	0.0316	9.2
	800	0.0395	11.8	0.0361	10.7	0.0337	9.9
	1200	0.0398	11.9	0.0364	10.8	0.0343	10.1
	1600	0.0399	12.0	0.0365	10.8	0.0346	10.2
-0.7	400	0.0526	9.9	0.0480	8.9	0.0401	7.3
	800	0.0548	10.3	0.0500	9.3	0.0454	8.4
	1200	0.0554	10.4	0.0506	9.4	0.0470	8.7
	1600	0.0556	10.5	0.0508	9.5	0.0477	8.8
1	400	0.0792	6.5	0.0649	6.4	0.0515	5.4
	800	0.0859	7.2	0.0703	6.9	0.0627	6.6
	1200	0.0874	7.3	0.0716	7.1	0.0657	7.0
	1600	0.0880	7.4	0.0721	7.1	0.0671	7.1

**Pages 2A-78 through 2A-125 intentionally left blank under the provisions of
the Energy Policy Act of 1992**

RAMPRESSOR™ ROTOR MECHANICAL DESIGN

- ☑ Four aero design rotor candidates have been analyzed so far
  - 105,000 rpm design with a pressure ratio of 16:1
  - 40,507 rpm design with a pressure ratio of 11:1
  - 49,000 rpm design with a pressure ratio of 15:1
  - 20,375 rpm design with a pressure ratio of 3.5:1

- ☑ Helped understand the design challenges, narrow the design space and at the same time meet the Rampressor™ Design Objectives.

- ☑ The following presentation will cover the details of the Mechanical Design Requirements, Stress Analysis methodology and results for the 20,375 rpm, 3.5:1 pressure ratio design.

MECHANICAL DESIGN CRITERIA

Low Cycle Fatigue Requirements

- **30,000 cycle design life.**

The Equivalent stress in the rotor to be less than 80% of the 0.2% yield strength and principal stress to be less than the 80% of the UTS. Typical LCF critical areas are disk bore, webs, holes and notches. For Ti 6242 the principal / equivalent stress design limit is 100 ksi.

High Cycle Fatigue Requirements

- **10 million cycle design life.**

The strake is subjected to strong vibratory excitation sources and should be designed with adequate alternating stress margin and with minimum background stress. For Ti 6242, based on Goodman Diagram the vibratory stress margin is to be less than 10 ksi if mean stress is 100 ksi.

Burst Requirements

- **125% of maximum permissible speed.**

The average disk tangential and radial stresses must be low enough to provide this safe margin on disk hub burst or rim separation.

RAMPRESSOR™ DESIGN CYCLE

Pertinent Cycle Parameters at MDP

- Max. Mechanical Speed : 20,375 rpm
- Condition : Standard Day (90F), S.L
- Rampressor Total Pressure Ratio: 2.5
- Rampressor Inlet Temp.: 35 F
- Rampressor Exit Temp.: 290 F

MATERIAL SELECTION

- Extensive material selection study carried out.
- Strength and weight requirements lead to Titanium alloy.
- Ti-6Al-2V-4Zr-2Mo has proven reliability, easy to machine and good corrosion and erosion resistance.

Ti 6242 Strength Properties

Properties at 200F

E = 17.4E6 psi

α = 5.2 μ in/in

ρ = 0.163 lb/in³

Elongation limit = 11 %

Temp. usage limit = 1000F

ANALYSIS METHODS

2D Axisymmetric Analysis with Point Mass

- Axisymmetric areas are simulated using Plane 42 elements with keyopt(3)=1
- Point Mass magnitude and CG location was calculated from heaviest rim segment. Simulated using Mass 21 element with keyopt(3)=4
- This method has a quick turn-around time.
- Good at predicting disk stresses and deflections.

**Pages 2A-131 through 2A-135 intentionally left blank under the provisions
of the Energy Policy Act of 1992**

Disk Shape Optimization

- Shape of the disk below the flow-path was parameterized.
- The Design Variables are the coordinates of key points which determine the shape of the disk.
- Ansys Optimizer performs analysis-evaluation-modification cycles until the design constraints are achieved.
- Ansys Optimizer used to minimize stresses in critical areas of disk to meet LCF requirements.

**Pages 2A-137 through 2A-173 intentionally left blank under the provisions
of the Energy Policy Act of 1992**

DISK BURST MARGIN CALCULATIONS

Avg. tangential stress in disk (s_{avg}) : 14.12 ksi

Avg. radial stress (S_{rad}) : 18 ksi at radius at a radius of 4 in.

Avg. bulk temperature of disk : 75 F

**Material Utilization Factor (MUF) : 0.85 for tangential burst
: 1.0 for radial burst**

Tangential Burst Margin : $\sqrt{s_{ult} (MUF) / s_{avg}} = 2.7$ (~ 55,000 rpm)

Radial Burst Margin : $\sqrt{s_{ult} (MUF) / s_{avg}} = 2.6$ (~ 53,000 rpm)

- **Disk stresses meet bulk burst margin requirements.**
- **The disk can be safely oversped to ~26,000 rpm.**
  - **Most of rotor < 60 ksi. O.K. for peak < 110 ksi. This would allow limited plastic deformation.**
  - **Temperature gradient will decrease peak stress.**

STRAKE VIBRATION ANALYSIS : FIXED ROOT CAMPBELL PLOT

- Fundamental mode > 10,000 hz.

STRAKE VIBRATION ANALYSIS : MODE SHAPES

GOODMAN PLOT FOR TI 6242 AT 200F

- Allowable alternating stress capability decreases with increase in mean stress.
- Controlling the mean stress < 100 ksi, leads to good vibratory stress margin.

RISKS AND CONCLUSIONS

RISKS

- **Low risk**

Rotor material properties should be cross-checked with test data. FE model should be updated.

- **Medium Risk**

Stress concentrations due to abusive machining or part not to print.

Need for good process control and checks / controls.

- **High Risk**

E-beam weld is susceptible to have cracks in the weld. Need for good control on welding process / vendor certification.

ROTOR/BEARING SYSTEM REVIEW

Barbour Stockwell, Inc.

Report no. 10189

Prepared for:

Ramgen Power Systems

Bellevue, WA

Prepared by:

Barbour Stockwell, Inc

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January 30, 2003

1. Introduction

BSi is currently manufacturing an air turbine drive system for Ramgen Power Systems for the purpose of testing a new compressor. In addition to this effort BSi has proposed, at the request of Ramgen, the performance of the following analytical work in support of the design of the compressor and the complete test rig.

- Task 1 – Rotor / Bearing System Review
- Task 2 – Dynamic Analysis

This document is a report of the results of the Task 1 effort. It includes a preliminary dynamic review of the compressor rotor bearing system. Deliverables also include design recommendations and the design of a coupling between the BSi turbine and a third party torque meter.

2. Compressor Rotor / Bearing System Review

Ramgen has provided to BSi the geometry of the compressor to be tested. From this data a model was created for the Dyrobes Rotordynamic Software package that was used to predict undamped critical speeds given the characteristics of the bearings employed. The model configuration from Dyrobes is shown in Exhibit 1.

Two ball bearing models were considered: New Hampshire Ball Bearing (NHBB) Part Nos. ME-2108 and ME-1108. These bearings are both 40 mm bore with identical external dimensions and have the same ball complement. The difference lies in the contact angle. The ME-2108 has a contact angle of 25° while the ME-1108 has an angle of 15°. The ME-2108 has a higher thrust capability. This comes, however, at the expense of radial stiffness. Stiffness at 20,000 rpm was provided by NHBB for both bearings. The values are presented below together with the resulting critical speed.

Bearing P/N	Preload (lbf)	Stiffness (Lbf/in.)	Critical Speed (rpm)
ME-2108	150	167,000	15,658
ME-1108	150	677,000	29,867

Given the rig design point speed of 19618 rpm, the use of the ME-2108 generates a potential problem. NHBB has suggested a preload of 200 lbf which would raise the stiffness of the ME-2108 to 286,000 lbf/in. Unfortunately, this only raises the critical speed to 20,221 rpm; alarmingly close to the design point.

The more rigorous rotordynamic analysis, which includes damping, proposed for Task 2 will serve to lower the critical speeds but the effect is expected to be small; probably less than 10 percent. These analyses will also indicate just how serious the problem of a critical in the operating range will be. A critical in the operating range is not necessarily fatal. Nonetheless, it is our recommendation that the ME-1108 be employed instead of the ME-2108. The Dyrobes output, which illustrates the mode shapes as well as the critical speeds, is presented in Exhibits 2, 3, and 4.

It should be noted that at speed the centrifugal ball loading will generate an axial force to oppose the preload. A preload sufficient to insure that it is not exceeded by the axial component of the ball loading is advisable to avoid the skidding that will result as the balls start running off the design contact angle.

To calculate the centrifugal ball load it is necessary to determine the rotational speed of the balls and the cage. The characteristic frequencies, including the cage frequency, of both bearings considered is presented vs. speed in Exhibits 5 and 6. As shown, there is little difference in the cage frequency between the two bearings. At a frequency of 145 Hz the 17 balls generate a centrifugal force of 170 lbf. With the 15° and 25° contact angles, the axial components are 46 lbf and 80 lbf, respectively. Therefore, the 150 lbf preload is more than adequate.

Please keep in mind that the table of characteristic frequencies can be a useful diagnostic tool. With the knowledge of these frequencies, monitoring of the spectral content of the signals from the recommended vibration instrumentation will allow detection and identification of various bearing defects.

3. Jack Shaft Analysis / Design

3.1. Splines

3.1.1. Turbine End

The involute spline on the turbine end of the jack shaft was designed for the output of the turbine when supplied with 280 psig / 400°F air. The torque vs. speed curve is presented as Exhibit 7. The spline selected is a 24/48 – 25 tooth design with a pitch diameter 1.042. Output from our spline analysis program, Exhibit 8, illustrates that the spline is well suited for this requirement. Fatigue loading is minimal and the spline can accept intermittent shock loads. The bursting stresses of the internal spline are low enough to allow a 0.125 in. wall thickness. Note that it is recommended that the bursting tensile stresses do not exceed 45 ksi for the 4340 material selected (33 – 38 Rc).

3.1.2. Torquemeter End

The capability of the spline at the torquemeter end was predicted for the operating conditions specified above. The results are displayed in Exhibit 9. As shown, the .8 in. pitch diameter spline does not have the capacity to transmit the torque under these conditions. The analysis was redone for the output of the turbine when supplied with 100 psig / 75°F air (See Exhibit 10). This is the air supply needed to meet the revised design point of 300 HP at 22,000 rpm. As shown in Exhibit 11, the spline has ample capacity to transmit the torque that will be generated with 100 psig / 75°F air.

3.2. Jack Shaft

Clearance requirements dictate a distance of approximately 6 inches between the torquemeter and turbine spline ends. A quill diameter of 0.75 in. was selected. The stall torque generated by the turbine with a 280 psig / 400°F air supply is 184 ft.lbf. From theory of elasticity we calculate a shear stress of 26 ksi which is an equivalent von Mises stress of 45 ksi. Typical yield strength at the minimum specified hardness is in excess of 125 ksi. Minimum factor of safety on yield is therefore 2.75.

The natural frequency of the jack shaft, P/N 9658-003, was calculated with the assumption that the splines would behave as pinned attachments. The first natural frequency is approximately 60,000 rpm.

The detailed manufacturing drawing of the jack shaft is presented as Exhibit 12.

4. Instrumentation

The employment of thermocouples at each bearing is strongly recommended. Provision should be made such that the thermocouples positively contact the bearing outer ring. Temperatures should be continuously monitored during operation with a limit of 85°C (185°F).

It is also suggested that accelerometers be installed on the housings in which the bearings are installed. The accelerometers should be as close to the plane of rotation of the bearing as possible. For future reference, the bearing defects associated with the characteristic frequencies are as follows.

- Outer Ball Pass Frequency – Defect on Outer Race
- Inner Ball Pass Frequency – Defect on Inner Race
- Ball Spin Frequency – Ball Defect
- Cage Frequency – Worn Cage

5. Design Recommendations

5.1. Bearings

- Use ME-1108 Bearings
- 150 lbf preload is satisfactory

5.2. Jack Shaft

- Limit operation to the 300 HP / 22,000 rpm torque curve.
- Misalignment between the turbine and torquemeter shafts must not exceed .003 in.

5.3. Instrumentation

- Thermocouples on the outer ring of each bearing.
- Accelerometers in the plane of rotation of each bearing.

EXHIBIT 1
DYROBES COMPRESSOR MODEL

EXHIBIT 2
UNDAMPED CRITICAL SPEED – ME1108 / 150 LBF PRELOAD

EXHIBIT 3
UNDAMPED CRITICAL SPEED – ME2108 / 150 LBf PRELOAD

EXHIBIT 4
 UNDAMPED CRITICAL SPEED - ME2108 / 200 LBf PRELOAD

EXHIBIT 5

Bearing Frequency Analysis

BEARING SPECIFICATIONS

Ball Diameter	0.313
Pitch Diameter	2.134
Number of Balls	17
Contact Angle	15 deg
Maximum Speed	50000 rpm

	SPEED									
DATA OUTPUT	5000	10000	15000	20000	25000	30000	35000	40000	45000	50000
Outer Ball Pass Frequency (Hz)	608	1216	1824	2433	3041	3649	4257	4865	5473	6081
Inner Race Ball Pass Frequency (Hz)	809	1617	2426	3234	4043	4851	5660	6468	7277	8085
Ball Spin Frequency (Hz)	279	558	836	1115	1394	1673	1952	2231	2509	2788
Cage Frequency (Hz)	36	72	107	143	179	215	250	286	322	358

EXHIBIT 6

Bearing Frequency Analysis

BEARING SPECIFICATIONS

Ball Diameter	0.313
Pitch Diameter	2.134
Number of Balls	17
Contact Angle	25 deg
Maximum Speed	50000 rpm

	SPEED									
DATA OUTPUT	5000	10000	15000	20000	25000	30000	35000	40000	45000	50000
Outer Ball Pass Frequency (Hz)	614	1229	1843	2457	3072	3686	4300	4915	5529	6143
Inner Race Ball Pass Frequency (Hz)	802	1605	2407	3209	4012	4814	5616	6419	7221	8023
Ball Spin Frequency (Hz)	280	559	839	1118	1398	1677	1957	2236	2516	2795
Cage Frequency (Hz)	36	72	108	145	181	217	253	289	325	361

**BSi RADIAL INFLOW TURBINE DESIGNED FOR 280 PSIG / 400F AIR
10.5 LBm/SEC. AT 280 PSIG / 400F**

EXHIBIT 8

Spline Data		
Maximum Operating Speed	50000 rpm	
D_{re} (Minor Dia)	0.94 in	
D_{ri} (Major Dia)	1.135 in	
D (Pitch Dia)	1.041 in	
N (Number of Teeth)	25	
Length	0.75 in	
D_h (Shaft ID - External)	0 in	
T_w (Radial Wall Thickness Internal Spline)	0.125 in	
T (Actual Tooth Thickness)	0.0654 in	
Pressure Angle	30 degrees	
K_f (Fatigue Factor)	1	
K_a (Application Factor)	1.5	
K_m (Misalignment Factor)	1	
K_w (Wear Factor)	0.5	
Application Data		
Torque	2184 in-lb	
Stress Results		
External Spline - Root - Shear Stress	20088 psi	40ksi max 4340 33-38 Rc
Pitch Dia - Shear Stress	10265 psi	40ksi max 4340 33-38 Rc
Compressive Stress on Teeth Sides	31 psi	12ksi max 4340 33-38 Rc
INTERNAL SPLINE		
Centrifugal Tensile Stress	6865 psi	
Beam Loading Tensile Stress	7166 psi	
Radial Tensile - Internal Spline	4113 psi	
Bursting Tensile Stress Total	23782 psi	45ksi max 4340 33-38 Rc

K_f (Fatigue Factor)		
Cycles	Reversing	Unidirection
1.00E+03	1.8	1.8
1.00E+04	1	1
1.00E+05	0.4	0.5
1.00E+06	0.3	0.4
1.00E+07	0.2	0.3
K_a (Application Factor)		
Load		
	Uniform	1
	Light Shock	1.2
	Intermittent Shock	1.5
	Heavy Shock	1.8
K_m (Misalignment Factor)		
Misalign	Face Width	
in/in	0.5	1
0.001	1	1
0.002	1	1
0.004	1	1.5
0.008	1.5	2
K_w (Wear Factor)		
Rev of Spline	K_w	
1.00E+04		4
1.00E+05		2.8
1.00E+06		2
1.00E+07		1.4
1.00E+08		1
1.00E+09		0.7
1.00E+10		0.5

EXHIBIT 9

T Meter Spline 1 Data		
Maximum Operating Speed	30000 rpm	
D_m (Minor Dia)	0.719 in	
D_n (Major Dia)	0.872 in	
D (Pitch Dia)	0.8 in	
N (Number of Teeth)	16	
Length	0.5 in	
D_h (Shaft ID - External)	0 in	
T_w (Radial Wall Thickness Internal Spline)	0.061 in	
T (Actual Tooth Thickness)	0.0812 in	
Pressure Angle	30 degrees	
K_f (Fatigue Factor)	1	
K_a (Application Factor)	1.5	
K_m (Misalignment Factor)	1	
K_w (Wear Factor)	0.5	
Application Data		
Torque	2184 in-lb	
Stress Results		
External Spline - Root - Shear Stress	44888 psi	40ksi max 4340 33-38 Rc
Pitch Dia - Shear Stress	25216 psi	40ksi max 4340 33-38 Rc
Compressive Stress on Teeth Sides	114 psi	12ksi max 4340 33-38 Rc
INTERNAL SPLINE		
Centrifugal Tensile Stress	1722 psi	
Beam Loading Tensile Stress	18200 psi	
Radial Tensile - Internal Spline	16449 psi	
Bursting Tensile Stress Total	53696 psi	45ksi max 4340 33-38 Rc

K_f (Fatigue Factor)		
Cycles	Reversing	Unidirectional
1.00E+03	1.8	1.8
1.00E+04	1	1
1.00E+05	0.4	0.5
1.00E+06	0.3	0.4
1.00E+07	0.2	0.3
K_a (Application Factor)		
Load		
Uniform		1
Light Shock		1.2
Intermittent Shock		1.5
Heavy Shock		1.8
K_m (Misalignment Factor)		
Misalign	Face Width	
in/in	0.5	1
0.001	1	1
0.002	1	1
0.004	1	1.5
0.008	1.5	2
K_w (Wear Factor)		
Rev of Spline	K_w	
1.00E+04		4
1.00E+05		2.8
1.00E+06		2
1.00E+07		1.4
1.00E+08		1
1.00E+09		0.7
1.00E+10		0.5

**BSi RADIAL INFLOW TURBINE DESIGNED FOR 280 PSIG / 220F AIR
6 LBm/SEC. AT 100 PSIG / 75F**

EXHIBIT 11

T Meter Spline 2 Data		
Maximum Operating Speed	30000 rpm	
D_{re} (Minor Dia)	0.719 in	
D_{ri} (Major Dia)	0.872 in	
D (Pitch Dia)	0.8 in	
N (Number of Teeth)	16	
Length	0.5 in	
D_h (Shaft ID - External)	0 in	
T_w (Radial Wall Thickness Internal Spline)	0.061 in	
T (Actual Tooth Thickness)	0.0812 in	
Pressure Angle	30 degrees	
K_f (Fatigue Factor)	1	
K_a (Application Factor)	1.5	
K_m (Misalignment Factor)	1	
K_w (Wear Factor)	0.5	
Application Data		
Torque	1008 in-lb	
Stress Results		
External Spline - Root - Shear Stress	20717 psi	40ksi max 4340 33-38 Rc
Pitch Dia - Shear Stress	11638 psi	40ksi max 4340 33-38 Rc
Compressive Stress on Teeth Sides	53 psi	12ksi max 4340 33-38 Rc
INTERNAL SPLINE		
Centrifugal Tensile Stress	1722 psi	
Beam Loading Tensile Stress	8400 psi	
Radial Tensile - Internal Spline	7592 psi	
Bursting Tensile Stress Total	25710 psi	45ksi max 4340 33-38 Rc

K_f (Fatigue Factor)		
Cycles	Reversing	Unidirection
1.00E+03	1.8	1.8
1.00E+04	1	1
1.00E+05	0.4	0.5
1.00E+06	0.3	0.4
1.00E+07	0.2	0.3
K_a (Application Factor)		
Load		
	Uniform	1
	Light Shock	1.2
	Intermittent Shock	1.5
	Heavy Shock	1.8
K_m (Misalignment Factor)		
Misalign in/in	Face Width 0.5	
0.001	1	1
0.002	1	1
0.004	1	1.5
0.008	1.5	2
K_w (Wear Factor)		
Rev of Spline	K_w	
1.00E+04		4
1.00E+05		2.8
1.00E+06		2
1.00E+07		1.4
1.00E+08		1
1.00E+09		0.7
1.00E+10		0.5

EXHIBIT 12

ROTOR DYNAMIC ANALYSIS

Barbour Stockwell, Inc.

Report No. 10207

PREPARED FOR:

Ramgen Power Systems

Bellevue, WA

February 25, 2003

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1. Introduction

BSi is currently manufacturing an air turbine drive system for Ramgen Power Systems for the purpose of testing a new compressor. In addition to this effort BSi has proposed, at the request of Ramgen, the performance of the following analytical work in support of the design of the compressor and the complete test rig.

- Task 1 – Rotor / Bearing System Review
- Task 2 – Dynamic Analysis

This document is a report of the Task 2 results. It includes a detailed dynamic review of the compressor rotor bearing system by itself and a review of the compressor assembly coupled to the torque meter and the drive turbine.

Deliverables include

- Whirl Speed Analysis
  - Compressor by itself
  - Compressor with torque meter and drive turbine
- Unbalance Response
  - Balance sensitivity for compressor rotor
  - Bearing loads
  - Expected bearing life
- Time Transient
  - Identify whirl speed modes
  - System reaction at identified speeds.

2. Compressor Rotor / Bearing System Whirl Speed Analysis

Ramgen has provided to BSi the compressor geometry to be tested. From this data a model was created for the Dyrobes Rotordynamic Software package that was used to predict damped critical speeds given the characteristics of the bearings employed. Whirl speed analysis is used to map the damped vibration frequencies at varying operating speeds. The model configuration from Dyrobes is shown in Figure 1. The model used NHBB ME-1108 ball bearings with 150 lbf preload as was recommended in Task 1 Analysis.

Figure #1
Dyrobes Damped Compressor Model

The resultant whirl speed mode map is shown in Figure #2. All modes mapped cross the synchronous line at angles obtuse enough to be of little concern other than operation exactly at those crossing points.

Figure #2
Whirl Speed Map - Compressor Model

3. Compressor, Torque Meter, and Drive Turbine Assembly Whirl Speed Analysis

The next analysis performed was a model that included the Compressor, the Torque Meter, the Jack Shaft, and the Drive Turbine. SensorData provided the data for the Torque Meter. They provided a cross-section of their shaft with dimensions and bearing locations. The Jack Shaft designed by BSi during Task 1 was used between the Torque Meter and the Drive Turbine. The Dyrobes model is shown in Figure 3.

Figure 3
Dyrobes Model - Assembly

When reviewing the whirl speed map, several modes are present in addition to the modes that were seen in the compressor only analysis. Similar to the compressor analysis, the crossing angles between the whirl speed mode lines and the operational speed line are relatively obtuse. There are several modes crossing the operation speed line in the speed range that Ramgen will operate.

Mode 1, 8100 RPM, is a rigid body mode for the drive turbine. The log decrement for this mode is 3.2, which means it is critically damped and not a concern.

Mode 2 and Mode 3 are also related to the drive turbine. Both of these modes are critically damped with log decrement values of 8.5 and 8.0 respectively.

All modes through the analysis range, 0 to 75000 RPM, show critical damping. This is not to say that loads are low when operating at an identified whirl speed mode. Loads are dependant on the forcing function, in this case, the unbalance of the various components. Critical damping only indicates that the vibration levels will not continually increase without end.

Figure 4
Whirl Speed Map - Assembly

4. Unbalance Response of the Compressor

The unbalance response was analyzed for several unbalance values. The response criteria were calculated as bearing load. Several curves are provided showing the response at different unbalance values. Figure 5, 20μ in MCD (Mass Center Displacement), has been extended to show the bearing loads at whirl speed mode 5. While the bearings are critically damped, the damping is sufficiently low to cause bearing failure at this point.

Figure 5
20 μ inch MCD – Bearing Loads

Since the maximum bearing load occurs at the maximum operating speed, the remainder of the sensitivity unbalance analysis was done at 20,000 RPM. As seen in figure 6, the bearing load at 20,000 rpm is 21 lbs with 20 μ inch MCD. Because the system is critically damped, the bearing load increased linearly with an increase in Mass Center Displacement. For example, 100 μ inch MCD results in a bearing load of 105 lbs and 200 μ inch results in a bearing load of 210lbs. The shape of the curve does not change.

Figure 6
20 μ inch MCD – Bearing Loads

The predicted bearing life is obviously dependent on bearing load, i.e. bearing life is dependent on compressor unbalance level. The table below lists the predicted bearing life values. The predicted life considered the bearing preload, the compressor weight, and the unbalance. It was assumed that lubrication is a suitable oil jet feed with good lubrication properties. The bearing life predictions did not include any aerodynamic loading.

Unbalance μ inch	L ₁₀ Life Hours
20	7.1E5
100	4.7E4
200	9.8E3
400	1.6E3

5. Time Transient Analysis

Two speeds were analyzed for a time transient analysis: 1) 19600 RPM and 2) 29800 RPM. Unbalance response predicts the system response based on an energy balance, while the time transient calculates the system response by stepping time. Each time increment uses the vibration data from the previous time increment and adds the available energy for the current time increment to determine the system reaction.

The bearing and joint reactions were reviewed for the entire system with a 50 μ inch unbalance level in the compressor and drive turbine. At 19,600 RPM all bearing reactions were similar to those seen in the unbalance response models and all spline joint reactions were below 3 lbs.

The bearing and joint reactions are much higher at the 29800 RPM. As Figure 7 indicates, the bearing loads will fail the compressor bearings. If the bearings were to survive, the loads delivered to the torque meter would cause the torque meter bearings and spline joints to fail.

Figure 7
Compressor Bearing Loads - 50 μ inch MCD

6. Operation Recommendations

The system analysis does not predict any damaging vibrations or loads in any of the system components in the operating range. If operation near 29000 RPM becomes necessary, additional damping will be required in the compressor bearing system.

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Retainer Ring Design

- Operating Conditions: 20,000 RPM

Bolted Configuration

- Operating Conditions: 30,500 rpm.

Temp. Distribution

Equivalent Stress

NODAL SOLUTION
STEP=2
SUB =20
TIME=2
SEQV (AVG)
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.009718
SMN =583.643
SMX =63830
583.643
7611
14638
21666
28693
35720
42748
49775
56802
63830

Radial Displacement

```
file
NODAL SOLUTION
STEP=2
SUB =20
TIME=2
UX      (AVG)
RSYS=11
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.009718
SMX =.009494
0
.001055
.00211
.003165
.004219
.005274
.006329
.007384
.008439
.009494
```

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2d Dove Tail FE Model

2d_dovetail
ELEMENTS
TEMPERATURES
TMIN=150
TMAX=262.065
150
162.452
174.903
187.355
199.807
212.258
224.71
237.162
249.613
262.065

Temp. Distribution

Equivalent Stress

SEQV (AVG)

PowerGraphics

EFACET=1

AVRES=Mat

DMX =.012782

SMN =84.057

SMX =299483

62 ksi

77 ksi

1.2782E-01

SEQV (AVG)

PowerGraphics

EFACET=1

AVRES=Mat

DMX =.012782

SMN =84.057

SMX =299483

Radial Displacement

Bearing Stress

S3 (AVG)
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.012782
SMN =-111771
-111771
-99352
-86933
-74514
-62095
-49676
-37257
-24838
-12419
0

RotorDynamics / Overspeed issues

- Compared to the monolithic Rotor, the weight decreased by 16% and the I_p decreased by 27%.
- Critical speed of the rotor expected to increase from 29,860 rpm to 33,000 ~ 35,000 rpm. BS needs to run analysis for exact number.
- Rampressor 1 - If the aero load is removed, then the drive can reach 23,000 rpm in .5 sec. Segmented design : Need to check for failure/safety issues at overspeed conditions. Need to re-evaluate this issue based on Aero. Data.

Mechanical Risk Issues

- The first manufacturing operation is EDM.
- EDM leaves a recast layer. Recast layer can be removed by ultrasonic polishers, rotary polishers, abrasive machining, heat treatment and chemical removal.
- Removing recast layer is expensive.
- Recast layer thickness and the HAZ thickness depends upon the heat sinking ability of the material and the power used. The recast layer is a very hard brittle layer and is prone to micro-cracks. Recast layer thickness can be 0.0002 in ~ 0.005 in.
- Thickness of recast layer has an effect on the structural integrity of this design. Need data from vendor. Data will dictate which faces will be machined.

**Pages 2A-223 through 2A-225 intentionally left blank under the provisions
of the Energy Policy Act of 1992**

Operating Speeds

- **Maximum Permissible – 29,000 RPM**
  - Maximum speed for any period of time
  - Determined by rotordynamics
  - Cannot exceed due to drive over-speed
- **Maximum Continuous – 27,931 RPM**
  - Maximum anticipated to be asked for by Aero:
    - Speed/Relative Mach # Margin
  - Mechanical Design Speed
    - Hot-to-Cold
    - Strake Tip Grinding
- **100% Physical Speed w/New PSN Design – 27,250 RPM**
  - Desired 100% Corrected Speed:
    - Anticipated Aero Design Point
    - PSN Designed At This Speed
    - 27,481 RPM 100% Corrected Speed (28,168 RPM 102.5% N_c)

All Physical RPM Values Based On 50 deg. F Air Supply (Total) Temperature

Speed Determinations

BSI Overspeed Calculations

- **BSI Estimates for 1.6S @ 30K RPM**
  - 355 HP load instantly removed from drive
  - 370 lbm-in² rotor inertia
  - 0.5 sec valve closure (nothing happens for ½ sec, then instant close)
  - 32,500 RPM is maximum system could get to => 8.3% speed increase
- **More Realistic Analysis**
  - **Non-Instant Removal of Aero Torque**
    - Rotor unstart or inlet starting (not back pressured) does not produce significant torque changes (based on CFD & analysis)
    - Only way feasible for quick aero unloading is if Boeing air supply was quickly cut off:
      - >100 ft. of 8" pipe + tank to blow down
      - Air control valve is slow (~1 sec to close)
  - **27.25K RPM HP < 355 HP used for calculations (at ~4 lbm/s physical flow)**
    - This HP requirement may increase depending on the final physical flow rate we end up at
  - **BSI Calculations Very Conservative**
    - BSI Drive Has Faster Than 0.5s Valve Response
      - More like 300 – 400 msec
    - Drive would not provide full torque during sensing and valve closure transient
    - BSI Can Do More Detailed Estimates/Modeling for \$\$
  - **Team Chose 4% Overspeed Value To Work With**
    - Could Instantly Lose Up To ½ of 355 HP and Not Exceed 29K RPM
    - Still Conservative

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- Compared to the monolithic Rotor, the weight decreased by 16% and the I_p decreased by 27%.
- BS recommends not to operate the compressor above 29000 RPM in its current configuration. Recommends using elastomeric or squeeze film dampers for speeds beyond 29,000 rpm.

Bolt Selection

Supplier: SPS Technologies

Each rotor needs 18 bolts

Bolt Part # LWB 22-4-42 H-11 or LWB 922-4-42

- **0.25 – 0.28 in. UNJF Bolt, Nominal Dia 0.25 in, Alloy Steel AMS 6485, length 2.5 in., 0.6875 in. grip. 220 Ksi minimum tensile strength, Temp limit 450F**

Nut Part # FN 22-428

- **220 ksi tensile strength, Temp limit 450F**

Bolt Torque

- **Yield Strength = 189 ksi. Torque = $0.20 \times 0.25 \times (0.80 \times 8338) / 12 = 28 \text{ ft-lb}$**

Bolted Configuration

- Operating Conditions: 30,500 rpm.

Temp. Distribution

Equivalent Stress (at 30,500 rpm)

NODAL SOLUTION
STEP=2
SUB =20
TIME=2
SEQV (AVG)
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.009718
SMN =583.643
SMX =63830
583.643
7611
14638
21666
28693
35720
42748
49775
56802
63830

- Peak Stress at 27,931 rpm ~ 53 ksi

Radial Displacement (at 30,500 rpm)

```
file
NODAL SOLUTION
STEP=2
SUB =20
TIME=2
UX      (AVG)
RSYS=11
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.009718
SMX =.009494
0
.001055
.00211
.003165
.004219
.005274
.006329
.007384
.008439
.009494
```

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Assembly Process

- Aluminum / Plexi-glass plate with protruding rollers / guide pins. The inner row of rollers/pins can be spring mounted.
- Lip seal strips to be inserted on one side of all segments.
- Segments to be “jiggled” into place.
- Disk to be installed to the top.
- Cover plate installed on disk.
- Insert bolts and lightly torque the nuts.
- Reverse assembly.
- Remove nuts and insert other cover plate.
- Torque assembly to full load.

[D]

Tip Grind and Balancing

- **Weigh-balance each segment “triplet”.**
- **Balance disk alone. Grinding can be done on top of disk posts or on disk sides.**
- **Assemble rotor.**
- **High Speed tip grinding at 21,000 rpm or 28,000 rpm or at both speeds ? Strake tips to be ground to a radius of 5.711 in. for 21,000 rpm grind, 5.710 in. for 28,000 rpm grind.**
- **Low speed balance assembly. Material can be removed from cover plates. Balance requirement = 0.005 oz-in.**

Conclusions

Recommendations

- Technically feasible and recommend going ahead with manufacturing.

Key Tasks / Issues

- Aero confirmation that bleed holes are o.k.
- Complete Seg 8 and Seg. 10 design.
- The solid models are ready for 'hot-to-cold' conversion. Lip seal slots are being modeled.
- Transition to completing and releasing drawings.
- Vendor selection – manufacturing time line is not known.

CONCEPTS NREC

TECHNICAL MEMORANDUM NO. 1006
Prepared for Ramgen Power Systems
Prepared by Nicholas C. Baines
Concepts NREC Project No. 6419
Letter of Agreement dated January 13, 2003
Project Director Nicholas C. D'Orsi
Project Manager Nicholas C. Baines
May 27, 2003

PRESWIRL VANE DESIGN

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PRESWIRL VANE DESIGN

1.0 INTRODUCTION

Ramgen Power Systems has a novel compressor designed to be capable of very high efficiency when running behind a highly swirling flow and is planning a rig test of the apparatus. To provide the required swirl, Ramgen envisions a preswirl device to be installed upstream of the compressor rotor. The preswirl device takes the form of an axial turbine stator blade. The key feature of the preswirl device is that it must provide a very carefully controlled incidence at the compressor, which must be maintained over the complete span of the compressor blade. The design specification is listed in Table 1.

Table 1. Preswirl Vane Design Specification

Inner radius of duct (in.)	5.036
Outer radius of duct (in.)	5.714
Mass flow rate (lbm/s)	3.184
Total temperature (°R)	510.0
Incidence angle (°)	0.0 at 20482.7 rpm on a 7.62° (hub to tip) rotor
Exit absolute Mach number	0.77
Exit absolute total pressure (psia)	23.5

2.0 DESIGN PROCEDURE

Initially the outline of the preswirl vane was developed using AXIAL^{TM1}, a meanline design and analysis code developed by Concepts NREC. The principal dimensions, including the inlet and exit blade angles, the true and axial chords, and the blade number, were obtained from this analysis. Because the analysis was only one-dimensional, attention was focused on the mid section of the vane, without regard at this stage for the spanwise variation of flow properties.

Based on this mid section, the full blade geometry was then developed using the AXCAD^{TM2} code of Concepts NREC. Eleven, equispaced, sections of constant radius were used to define the blade initially. Later it was found that closer control of the blade geometry was

¹ AXIAL is a trademark of Concepts ETI, Inc.

² AXCAD is a trademark of Concepts ETI, Inc.

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required near the end walls, and so three additional sections were added at 5%, 85%, and 95% span. Initial analyses were performed by the through flow analysis incorporated in AXCAD. In view of the importance in achieving the correct compressor blade incidence, however, most of the blade geometry development was done using Pushbutton CFD®³. This is a three dimensional, viscous analysis. Its particular advantage in this application is that it is optimized for analyzing blade passage flow fields, and is closely integrated with AXCAD. A large number of design iterations can therefore be run in a short space of time.

The domain of the CFD analysis was extended upstream of the vane leading edge and downstream of the location of the compressor blade leading edge in order to limit the influence of the boundary conditions. Adjustments to the section geometries and the blade number were made as necessary to achieve the required flow distribution at the compressor blade leading edge location. The fillets between the preswirl vanes and the end walls are not included in the CFD analysis. Experience suggests that these have very little aerodynamic influence, but the vane throat area was adjusted to allow for manufacturing fillet radii of 0.031 in.

3.0 RESULTS

The flow conditions at the location of the compressor blade leading edge, as computed by the CFD analysis, were circumferentially mass-averaged, and are plotted as functions of span in Figures 1-3. The variation of incidence angle is shown in Figure 1. Over the center 90% of the span, the incidence angle is about $+0.3^\circ$ to $+0.5^\circ$, which was considered to be acceptable for the compressor blade design proposed. The incidence angle is considerably higher than this close to the walls, and this is a result of underturning in the end wall boundary layers of the preswirl vanes.

Figure 1

The distribution of total pressure in Figure 2 shows two features that result from the flow turning in the preswirl vanes. One is the large loss of total pressure in the end wall boundary layers, and the other is the passage secondary flow vortices that are apparent at about 10% and 90% span. These features are also evident in the Mach number distribution in Figure 3.

³ Pushbutton CFD is a registered trademark of Concepts ETI, Inc.

Figures 2 and 3 indicate that the targets for total pressure and Mach number have been met over the majority of the blade span. As far as possible, the effects of the end wall boundary layers and secondary flows have been compensated by changes to the relevant blade sections, but there are limits to this process. For example, it proved impossible to eliminate the underturning close to the walls by increasing the stagger angle of the vane sections in these regions. Such increases only cause the deviation angle to increase, with little or no net effect on the flow angle. Furthermore, such restaggering increases the difficulty and cost of manufacture of the preswirl vanes. A better approach to this problem would be to remove the boundary layer by suction between the preswirl vanes and the compressor blades.

The preswirl vane geometry has been separately transmitted to Ramgen in electronic form as a solid model CAD file.

Figure 2

Figure 3

4.0 CONCLUSIONS

The preswirl vane design has been developed to meet the target incidence angle, total pressure, and Mach number at the location of the compressor blade leading edge, within practical limits set by aerodynamic and manufacturing constraints.

CHAPTER 3: COMPLETE DESIGN AND BUILD OF THE 2ND RAMPRESSOR RIG (TASK 18)

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3 COMPLETE DESIGN AND BUILD OF THE 2ND RAMPRESSOR RIG

3.1 Rampressor-2 Overview

3.1.1 Goals and Objectives

The Rampressor-2 program has three specific technical program goals:

- Demonstrate rotor flow path aerodynamics traceable to a ~10:1 Rampressor.
- Demonstrate a rotor mechanical design traceable to a ~10:1 Rampressor.
- Build a test rig capable of retrofit for operation at lower pressure ratio, specifically in the range of 4.5:1 to 7:1.

Achieving the first two goals would build the foundation for a ~10:1 Rampressor product. Achieving the third goal would allow the use of the Rampressor-2 test rig in the development of lower pressure ratio products.

The specific technical objectives which must be met to achieve these goals are:

- Gather data necessary to anchor the mean line design and the CFD design tool.
- Measure the rotor only performance at the design point and off design conditions.
- Demonstrate the starting process.
- Demonstrate self aspirating capability.
- Demonstrate a rotor mechanical design capable of operation with tip speeds of ~2000 ft/sec. This includes the development of segment attachment schemes, web designs, and cross hole geometries the meet acceptable stress levels. In addition, the development of design and machining expertise such that the rotor has the desired shape at the design point is required.
- Demonstrate a mechanical design traceable to low tip leakage.
- Develop a plan that details the retrofit required for operation at lower pressure ratios.

3.2 Static Test Rig Design

3.2.1 Components

3.2.1.1 Rig Overview

The Rampressor 2 test platform was developed as an extension of the Rampressor 1 rig architecture. In addition to its design for the higher pressure, temperature, load and speed conditions, produced by the increased pressure ratio disc, the rig incorporates alignment, flowpath serviceability and tip clearance control improvements. Functionality and mechanical design elements will be detailed in this section. Aerodynamic design will be covered in Section 3.3. The core rig, stationary hardware will be discussed within the following divisions:

Skid and Support Frames

Rig Frames

Disc Cartridge

Primary Flowpath

Bleed Flowpaths

Wheelspace Flowpaths

Tip Clearance Control

Drive and Coupling

A side view of the Rampressor-2 Test Rig extracted from the CAD model is presented in Figure 3.2.1-1

3.2.1.1.1 Skid and Support Frame

The skid and support frames provide the attachment points and base alignment for all components in the power train. The skid is a steel weldment which is passively attached to the facility floor (see Figure 3.2.1-2). Under all predicted operating and failure loads, the skid possesses sufficient stiffness to maintain alignment independent of the facility floor's surface condition (see Figure 3.2.1-3). All facility service attachments to components on the skid are flexible. Machined surfaces on the skid provide mounting points for the support frames, the inflow diffuser and the drive turbine. Steel V-tracks allow for translation of the flow conditioner.

The Rampressor rotor is contained within two structural rig frames. The rig frames are mounted on three rails suspended between two support frames (see Figure 3.2.1-4). The rig frames can be independently translated along the three rails between the support frames. During operation the frames are bolted together and bolted to the aft support frame. Independent translation of the rig frames allows the best access to the two components of primary investigation: the rotor and the tip ring.

3.2.1.1.2 Rig Frames

Two rig frames, fwd and aft, contain the Rampressor rotor. Their primary function is structural support and containment. They also serve as attachment points for flowpath shrouds, fluid services, and instrumentation. These structures and their attachments to the support frames are designed to withstand the radial loads generated by a single disc post failure at 110% speed (mass removal of one disc post and two segments) combined with the torsional loads generated by a deep rub of the disc strakes (blades) into the tip shroud. The radial, disc post load was applied in 45 deg increments to the structure to determine the maximum resultant stresses (see Figure 3.2.1-5 for model and boundary conditions).

3.2.1.1.3 Disc Cartridge

The disc cartridge; consisting of the rotor (disc, segments and coverplates), the bearings, the bearing lock nuts and the forward wheel space cover; can be installed as a unit into the rig frames (see Figure 3.2.1-6). The modular nature of this unit improves inspection and service of the flowpath by allowing quick access to the hardware.

The rotor is simply supported on two rolling element bearings. In the Rampressor 1 test rig, two opposed 108-size, hybrid, angular contact bearings were used. The bearing pair was preloaded in the aft direction by a spring. During operation, conditions were encountered which suggest that thermal expansion of the bearing races prevented the small degree of axial travel required to maintain the bearing preload under the varying thermal and mechanical loads. To address this concern, two alternate configurations were investigated for the Rampressor 2 rig: a simply-supported deep groove ball and roller arrangement, and an overhung deep groove ball and roller arrangement. Despite their limited life, rolling element bearing configurations were selected over hydrodynamic bearings for their superior strake tip position control. The simply supported rotor was selected (see Figure 3.2.1-7). This configuration demonstrated better rotordynamic stability over the operating speed range. The deep groove ball bearing was selected for the forward shaft end and a roller bearing selected for the aft shaft end. The deep groove bearing carries thrust loads which preload the bearings and which are aerodynamically generated during operation. The roller bearing allows for differential thermal component growth. This system requires an external preload in the forward direction which is discussed in Section 3.2.2.1.6. Due to the high surface speeds encountered in the design, commercial bearings were not available for this application. The hybrid bearing geometry used is a result of a coupled bearing system design and heat transfer analysis performed using SKF's Shaberth code. In addition to strict geometric tolerances this bearing system requires targeted oil introduction. The forward bearing requires that oil jet lubrication be used in conjunction to through-rotor lubrication to effectively cool the inner and outer race contact points. Oil jet cooling alone is sufficient for the roller bearing (see Figure 3.2.1-8 and Figure 3.2.1-9). To avoid oil pooling, which would promote roller skidding, on the reflected oil side of the bearing, oil drains are provided on both sides of the bearings.

The rotor cartridge contains all features necessary for rotor retention, sealing, and lubrication (see Figure 3.2.1-10). Aerodynamic loads generated on the rotor are transferred through the deep groove bearing. The inner bearing race is secured with a nut and lock washer to the disc shaft. The outer bearing race is captured between the forward frame components and the forward wheelspace cover. The forward wheelspace cover is the only component of the rotor cartridge which does not rotate during operation. Under normal operation neither the

bearing nut nor the forward wheelspace cover experience additional loading. In the event of a thrust reversal, the deep groove bearing, outer race, contact point shifts from the forward to the aft side. When this occurs, the bearing nut and the wheelspace cover carry the structural loads. Features on the bearing nut and the disc shaft convey oil to the underside of the bearing inner race. Additional features on the shaft provide sealing between pressurized cavities. Four tooth labyrinth seals on the forward and aft, disc shafts produce a seal between the oil drains and the pressurized wheelspace cavities. The wheelspace cavities are held at a pressure 5 psi or greater than ambient. This natural pressure gradient between the wheel space cavity and the oil cavity prevents the oil mist generated by the rotating bearings from entering the wheelspace cavities. As this seal represents a parasitic system loss, the tooth profile and the tip clearance of the labyrinth seal is designed to maximize pressure loss across the seal, thus reducing the buffer air flow requirement. A brass insert in the wheelspace covers makes up the stationary portion of the labyrinth seal. The insert also has the closest axial proximity to the rotor. In the event of a bearing failure, this sacrificial part protects other stationary components from contact and promotes in plane rotation. A second labyrinth seal is incorporated into the roller bearing lock nut. This seal prevents oil mist from leaving the rig along the drive shaft.

3.2.1.1.4 Primary Flowpath

The primary flowpath consists of inflow fluid conditioning, a pre-swirl nozzle stage, a compressor stage, discharge collection, and throttling. Detailed discussion of the static flowpath's aerodynamic design will appear in Section 3.3.1. The following section will discuss the functionality of the components.

Inflow air can be introduced into the rig from a facility supply or through natural aspiration. The facility air supply is conditioned in a 24 in. pipe diffuser prior to introduction into the Rampressor. A 24 in. service section is attached to the facility diffuser. The service manifold and facility diffuser reside adjacent to an inflow bellmouth. The fiberglass bellmouth is contoured per ASME flowmeter specifications. The bellmouth seals against the flange of the service section with an inflatable seal. The inflatable seal prevents leakage and minimizes the alignment requirements between the rig and the facility hardware (see Figure 3.2.1-11). While the Rampressor is intended to operate with simulated, ambient conditions; the inflow structure is designed to handle pressures up to 50 psia for pre-swirl nozzle characterization tests.

Air flows through an annular passage, along the service centerbody, and through a set of struts in the forward frame, before entering the pre-swirl nozzles (PSN). The six frame struts have a NACA 63-A015 low loss profile. All forward services leave the rig through the service centerbody. The centerbody is cantilevered from the forward frame (see Figure 3.2.1-11). Similarly the PSN shrouds are attached to the forward frame. The PSN stage is fabricated as a collection of individual blades. Each blade has integral hub and tip platforms. All blades are captured axially at the tip platform. The hub platform is positioned within a channel in the hub shroud, but the platform does not contact the channel. The PSN blade remains stationary while the channel thermally shifts. Only the seals contact the blades directly and prevent axial air flow under the blades. The blade's tip platform is only a half platform. The trailing edge half of the tip platform has been removed to allow for actuation of the movable rotor tip shroud. The outer PSN shroud contains a thin wall which acts in place of the downstream half of the PSN blade platform (see Figure 3.2.1-12). The exposed blade section is sealed to the outer PSN shroud after assembly to prevent flow between the pressure and suction side of the blades. The slash angle, the cut between the adjacent blades of the PSN stage, is defined to separate blade near the mid-passage and to minimize the trailing edge corner stresses. The profile of the blade trailing edge has corner, boundary layer control which creates an overhung section. This section is susceptible to shock loading from the passing rotor leading edge. The material selection and slash angle design are used to minimize the stresses resulting from a 3 atmosphere pressure load on the pressure surface (see Figure 3.2.1-13). Downstream of the PSN stage, the outer PSN shroud produces a 0.034 inch aft facing step in the flowpath upstream of the rotor.

Downstream of the rotor the compressed air is collected in a square cross-section manifold. The entrance area is sized for 2.5 times the expected flow to prevent throttling. The collector has two discharge paths. The primary discharge carries a wafer valve for throttling the flow. The secondary discharge has a binary valve for starting. The binary valve can be opened during starting to minimize the collector back-pressure; then closed for throttling with the primary valve. The exhaust collector also houses a high temperature sliding seal for the moveable rotor tip shroud. The tip shroud seal uses a combination of braided and laminated Graphoil packing rings to create a functional seal throughout its 700 degree Fahrenheit operating range. The packing is preloaded by a backing ring and 8 belleville washer stacks (see Figure 3.2.1-14).

3.2.1.1.5 Bleed Flowpaths

This section intentionally left blank under the provisions of The Energy Policy Act of 1992.

3.2.1.1.6 Wheelspace Flowpaths

A cavity is created on both sides of the rotor between the face of the disc and the stationary structure. These wheel spaces cavities serve three purposes: sealing, disc cooling and thrust balance. The wheelspace cavities act as air reservoirs for the labyrinth seals. These seals ensure that bearing lubrication is prevented from entering the flowpath. These seals are discussed in section 3.2.2.1.3. Disc cooling is performed by creating a continuous, outward, radial flow through the wheelspace cavity along the disc face. This flow reduces heating of the air and the disc due to air churning. The cavity width and other aspects of the cooling design will be discussed in section 3.3.1.2. The forward and aft flow circuits can be seen in Figure 3.2.1-15 and Figure 3.2.1-16. The third function of the wheelspace cavities is thrust balance. The differential static pressure in the forward and aft wheelspace cavities enables the disc to act as a thrust piston. The thrust force created between the cavities is used to preload the bearing system and to counteract the rotor's aerodynamic forces. Thrust is measured by a load cell behind the forward oil manifold. A proximity probe detects rotor position to ensure proper preloading of the bearing system (see Figure 3.2.1-17).

3.2.1.1.7 Tip Clearance Control

The tip shroud makes up the fourth wall of the Rampressor flowpath. An abradable coating is applied to this stationary wall to minimize the effects of contact with the rotor. Discussion of the coating and its selection process can be found in Section 3.3.1.3. The shroud has a shallow conical angle which thermally expands to match the rotor cone angle at the design operating point. To allow for off-design operation and to ensure minimum clearance can be achieved, the shroud is able to translate axially. This action varies to gap between the shroud and the rotor from 0.030 in. clearance to 0.020 in. interference.

The tip shroud is locally water cooled. The Rampressor 1 rig utilized a thermally controlled tip clearance system; wherein external heating and cooling was used to thermally vary the diameter of the shroud relative to the rotor. This type of system has thermal input from the tip shroud to the flowpath gas. Heat addition to the flowpath artificially alters the measured rotor performance by varying the rotor exhaust gas temperature. Accounting for this heat input requires knowledge of the split between energy added to the flowpath gas and energy added to the static structure. To eliminate the performance error associated in this calculation, the Rampressor 2 clearance system is designed as near adiabatic. The rig does not use an external source to vary the tip shroud size. The tip shroud utilizes heat addition from the flowpath to develop its conical match with the rotor. Clearance is varied by translation of the shroud. A small portion of the tip shroud is cooled. Water cooling through two annular passages upstream of the rotor is used to prevent misalignment caused by heat conduction into the static structure (see Figure 3.2.1-18). A secondary member, the inner tip shroud guide, forms portion of the water channel, routes water through the part and allows easy removal of the tip shroud from the rig. Since the flowpath air travels through the cooled region of at its highest Mach number and it lowest temperature, this clearance system is treated as adiabatic.

The moving tip shroud has a sliding seal on each end to prevent air escape. The forward seal is a conventional elastomer o-ring. The water cooling prevents local metal temperatures from exceeding the limits of this simple seal. The seal is located at the extreme upstream end of the tip shroud. This prevents the seal from bearing on the thin portion of the pre-swirl nozzle shroud and produces a small region of trapped air, at static flowpath conditions, to provide additional thermal isolation of the core air flow from the cooled section of the tip shroud. On the aft end of the tip shroud, the 700 degree Fahrenheit gas temperature prevents the use of elastomer o-rings. A four element graphite seal is used. The seal fits into a packing box integrated into the exhaust collector and is preloaded though a secondary ring. The assembly and preloading required for the graphite seal make it impractical to disassemble with each rig teardown. To allow the stationary and sliding portions of the seal to remain engaged during disassembly the tip shroud was split into two sections: abradable and sliding seal (see Figure 3.2.1-18). A metal o-ring is used the seal the static joint between the two shroud sections.

The tip ring is moved by two actuators which are mounted to the compressor skid. The tip shroud and the inner tip shroud guide are removed as a unit from an outer tip shroud guide. This allows the use of multiple tip shrouds; preventing delays during reapplication of the tip shroud's abradable coating. The outer tip shroud guide is attached to the forward rig frame through three linear guides. The outer tip shroud guide remains attached to the rig during disassembly to preserve the tip ring's alignment with the rotor centerline. The tip shroud actuators are attached to

the outer tip shroud guide 180 degrees apart, on a horizontal line through the rotational centerline (see Figure 3.2.1-19). Spherical bearings are used at the interface to prevent non-axial load transfer to the tip clearance system. The bearings handle the minor misalignment generated by actuator mount racking under varying seal loads and by differential thermal growth of the rig and supports. The actuators are stepper motor driven ball screw type. Actuators are controlled as master and slave to ensure coordinated movement. The actuators have position holding capability which prevents shifting under variable load.

3.2.1.1.8 Drive and Coupling

A radial inflow drive turbine is used to power the Rampressor 2 rig. The drive is capable of 1100 HP at 50,000 rpm. Control valves, hardware and software are provided by the manufacturer. The drive is attached to the skid with three trunion mounts. These circumferential pins are attached to the drive case and inserted into spherical bearings in trunion blocks bolted to the Rampressor skid (see Figure 3.2.1-20). Two horizontal pins provide primary load carrying and the third, bottom dead center, pin provides pitch control and lateral positioning. The cylindrical fits allow the case to expand and contract thermally without affecting alignment.

The Rampressor rotor is connected to the drive with a flexible element coupling. The original Rampressor rig used a quill shaft. The quill shaft design has high load carrying capability, but limited design life as a result of its reliance on sliding surface contact to handle misalignment. The Rampressor 2 rig uses a flexible element coupling (see Figure 3.2.1-21). This style of coupling handles misalignment by deforming a stack of thin metal discs. The mean stress in the disc is design for infinite life. The coupling design allows 0.040 in. axial and 0.25 degree, or 0.070 in., parallel misalignment between the rig and the drive. The coupling is attached to the drive and rotor shafts with keyed taper fits. Split lines in the coupling allow the part to be separated in way consistent with the sliding frame disassembly method of the Rampressor 2 rig.

3.2.1.2 Instrumentation

Instrumentation is a critical part of the Rampressor 2 rig, performance validation and safe operation. Instrumentation and instrumentation routing account for a significant portion of the component complexity and rig design effort. More than 450 measurements locations are integrated into the test rig. These measurements fall, broadly, into one of three classifications: control measurements, rig health monitoring measurements and performance measurements. Control measurements are those measurements associated with actively controlled, system valves or with parameters indicating system valve operation. Rig health measurements monitor time varying parameters which cannot be actively controlled, but have predetermined safe operating limits. Performance measurements are measurements used to evaluate efficiency, power, pressure ratio or other aerodynamic or mechanical performance metrics. Sampling rates for the instrumentation are a function of the associated data and control systems. Information on sampling rate can be found in the Control section, 3.2.2.3.

3.2.1.2.1 Control Instrumentation

Control of the Rampressor rig is exercised through inflow pressure control of the turbine speed and through a series of throttle valves used to vary pressure within the Rampressor. Control instrumentation consists of drive speed and flow condition measurements; rig valve positions, flow pressures and flow temperatures; rig inflow conditions; lube system, pump and flow conditions; and tip clearance actuator position.

The primary measurement for control of the Rampressor 2 is drive turbine speed. Since the Rampressor rig relies on the drive turbine to maintain its speed and relies on the relative Mach number for performance, accurate speed measurement is crucial to proper operation. Speed of the drive turbine is determined by a magnetic pick-up viewing a 6 tooth gear on the aft side of the drive turbine shaft. The signal is processed with a API-7580G frequency to DC signal isolator. 12 bit sampling yield a speed error of +/-12 rpm, or <0.05%. Redundant measurement of the rotational speed can be extracted from the tip clearance system. A hall-effect sensor, Honeywell DZH460, reads a slot in the coupling hub, yielding a 5Vone-per-revolution signal. The signal is sampled with a National Instruments PCI-6110E data acquisition board or a Tektronics 420A oscilloscope at 1MHz or better sampling rate, yielding <0.1% error. A similar calculation can be performed to determine the rotor speed using the tip clearance probes to detect stake tip passing. The same electronics are used for the Hall-Effect probe and the tip clearance probes.

Control of the drive speed is performed by the manufacturer's control system. Control is exercised through precision regulation of the inflow pressure. Facility instruments monitor the header pressure to the turbine control

valves and the inflow and exhaust air temperatures. The air temperatures are used to vary the inflow temperature so that the exhaust temperature does not drop below the dew point. Facility pressure measurement is performed with 0.25% stand-alone pressure transducers and K-type thermocouples.

Six throttling valves control the flowpath back-pressure and the wheelspace cavity pressures inside rig. Open loop control of the valves necessitates only static pressure measurements in the cavities. All cavity pressure measurements taken for control purposes are also used in the rotor thrust calculation, pressure times area force summation for the loads on the rotor surfaces. Since the pressures are involved in this calculation, measurements are taken at the closest practical proximity to the rotor. See Figure 3.2.1-22. All measurements are taken with 0.25% stand-alone static pressure transducers. Three forward and three aft wheelspace cavity measurements are sampled. The other control pressures are sampled at one location. Redundancy for all these pressure measurements is captured in the performance measurements.

While most testing is expected to be performed with 50° F inflow air temperature, provisions have been made in the design of the rig to operate with inflow air temperatures as low as -50° F. At the same rotational speed, this lower temperature air will have a higher relative Mach number. When chilling the air with liquid nitrogen, the inflow temperature is used to control the LN₂ mixing valve position. A platinum RTD is used to measure this temperature.

Pressure control is used for the lubrication system. As a fixed geometry, oil jet lubrication system; only inflow reservoir pressure monitoring is required for system control. Two pressure measurements are taken on each the ball and the roller bearing supply lines to verify pump operation. Measurements are taken downstream of any flexible supply lines, in rigid fittings, just before oil enters the frames. 0.25%, intrinsically safe, stand-alone transducers are used. Additional health measurements provide redundancy in the performance evaluation of this system.

In the Rampressor rig, the rotor performance is strongly dependant on tip clearance. Repeatability of the tip shroud position is important for repeating test conditions. Control of the tip shroud actuator is contained within the actuator control system. Comparison of stepper motor count and LVDT output allows the actuator to repeat axial position within .003 in. Additional tip clearance measurement methods are discussed in the Performance Instrumentation section.

3.2.1.2.2 Rig Health Monitoring Instrumentation

Rig health monitoring is performed on several Rampressor systems to ensure that the rig is operating within design parameters. The parameters associated with these measurements cannot be directly affected during the test. Operation outside of the specified limits indicates a problem with the associated system requiring hardware repair, rework or redesign. Rig health parameters are primarily associated with proper operation of the bearing system and with maintaining alignment.

The lubrication system's control parameter is oil supply pressure. Health measurements include: oil flow rate and oil temperature rise. The oil flow rate to the roller and the ball bearing is measured on the supply side of the flow circuit. 0.1-1 gpm Hedland H600B-001 flow meters are used to measure the 0.3 gpm flows. Predictions for oil temperature rise have been provided with the bearing system design. Higher temperatures are indicative of roller skidding; a sign of larger than predicted operating clearance, light roller loading or poor oil system drainage. Inflow temperature is measured for each bearing, at the rig frame entrance, with a, platinum RTD. The oil outflow temperature is measured inside the rig. A pool is created in the interior, rotor side, oil drain cavity. This pool minimizes oil cooling by the purge air which is also exhausted through the oil drains. K-type thermocouples are used because of the difficult interior routing. These values are also used to calculate the parasitic loss of the bearings.

In addition to bearing temperature extrapolation through oil temperature rise, direct measurement of the bearing outer race temperature is made. Bare-bead, K-type thermocouples are potted into radial holes in the frame's bearing bores and the bead is then ground flush with the bore surface. See Figure 3.2.1-23. When the bearings are installed and have thermally expanded, the races make contact with the thermocouple bead. Three thermocouples are installed for each bearing race.

Proper operation of the bearings is dependant on maintaining the bearing thrust load between prescribed limits. A pressure times area calculation on the rotor surfaces is the control method for thrust load determination. Two additional measurements provide redundancy for the control calculation. A load cell is installed to measure thrust loads transmitted through the ball bearing's outer race as shown in Figure 3.2.1-24. A positive reading on the load

cell provides an indication of proper bearing installation, i.e. full engagement of the bearing in the housing and proper seating of the forward wheelspace shroud. This indication is important because the installation of the ball bearing is blind and the race width can vary between bearings. A small gap behind the bearing could result in a rapid axial bearing movement during operation. Direct thrust measurement from the load cell is unreliable for determining bearing thrust loads. In addition to the preload on the thrust measurement resulting from compression during bearing installation, thermal growth of the bearing race and of the forward oil manifold appear as thrust increases. Additionally, some of the applied thrust is reacted by the press fit between the outer bearing race and the forward frame. This press fit and its associated friction vary with operating speed and flow conditions. Even with its inherent uncertainty, the load cell can provide useful data. The load cell's measurement is important reinforcement of thrust trending and can be indicative of coupling applied, axial loads or other mechanical loads which are not captured in the pressure times area measurement. The load cell is a 0-5000 lbf, semi-custom fabrication based on Honeywell's Model D platform. A third indication of thrust is provided by a thrust proximity probe, viewing the forward face of the disc shaft. A Metrix 1000 series, 9 meter, eddy current probe is used. This probe measures the position of the shaft relative to the forward frame in order to detect a thrust reversal. In a thrust reversal the contact switches from the forward to the aft face of the ball bearing outer race. Operation in a reversed thrust state is undesirable, because of the loss of radial rotor control during the contact face transition.

Proper operation, of the drive train, is monitored by a pair of proximity sensors. Metrix 1000 series, 9 meter, eddy current probes are used. The probes are mounted 90 degrees apart into a bracket connected to the drive face. The probes are positioned to detect variation in the centerline position of the flex coupling's center spool. By comparing the centerline position shift relative to the drive centerline, the angular misalignment of the coupling can be held within manufacturer's limits. In this way, the sensors monitor thermal drift and operational static deflection between the rig and the drive rotational centerlines.

Velocimeters are used to monitor the severity of rotational imbalance in the drive and the rig. Velocimeters have been selected over accelerometers in accordance with best practice as discussed in ISO 10816-1:1995 "Mechanical Vibration-Evaluation of machine vibration by measurements on non-rotating parts". Wilcoxon PC420VP-10, 0-1 inch/sec, velocity transmitters are used for this purpose. Three sensors are mounted in a tri-axial arrangement on the horizontal, vertical and axial surfaces of each the forward and aft rig frames and the drive turbine, as shown in Figure 3.2.1-25. Operational vibration is held within limits prescribed in the standard for the machine power class. An additional velocimeter is mounted radially on the movable tip ring in order to detect rotor contact.

Water cooling systems for the tip ring and aft frame operate on RTDs installed into the static components. The temperature of the component is maintained by adjusting the supply water temperature. Platinum RTDs are used to monitor the rig supply and discharge water temperatures. Used in conjunction with the cooler's pump flowrate, the heat extraction can be calculated.

3.2.1.2.3 Performance Instrumentation

The primary purpose of the Rampressor 2 rig is to measure flow path performance. The type, location and redundancy of total and static pressure and temperature measurements have been designed around this theme. All performance related pressures are collected using the facility, Scanivalve Hyscan system. This modular wind tunnel pressure system is calibrated prior to each test and offers 0.25% FS accuracy. A list of instrument channels and their associated module pressure range can be found in the Chapter 3 Appendix. All wall statics associated with the flowpath are manufactured with thick wall geometry. 0.020 in. diameter, sharp edge holes penetrate the flowpath. Larger diameter pressure lines are installed into counter-bored holes on the exterior side of the hardware for egress.

Inflow conditions are measured in the 24 in. pipe section upstream of the inflow bellmouth. The local Mach number in this region is 0.012. With the low Mach number, static temperature and pressure measurements are used to establish total pressure and temperature conditions for the rig entrance. Temperature is measured with 4, platinum RTDs. Static pressure is measured with 4, United Sensor, PTC-6 probes. All measurements are collected 4 in. from the outer walls in the same plane. Similar measurements have 90 deg spacing. Temperature and pressure measurements are separated by 45 deg.

Characterization of the Pre-Swirl Nozzle (PSN) is important for establishing the rotor entrance conditions. To avoid the need to survey upstream of the rotor, PSN exit conditions in operation are correlated with data collected from static flow surveys. The operational surveys are risky because, the rotor vibrations induced by a strong once-per-revolution excitation and its harmonics can damage the rotor. And the three-per-revolution excitation can

damage the probe. Flow angle, total pressure and static pressure are collected at the PSN exit. The PSN Omega Bar, $((P_{t\_inlet} - P_{t\_exit}) / (P_{t\_exit} - P_{s\_exit}))$, is calculated for comparison to the inlet corrected mass flow conditions. A look-up table is generated from the results. Using the table, rotor entrance total pressure can be derived from mass flow, inlet plenum total pressure and PSN exit static pressure. The cobra probe, used to determine flow angle and PSN exit total pressure, is a 0.040 in. diameter tube, custom fabrication. The probe has been calibrated in the test cell for Mach numbers from 0.1 to 0.9 and for flow angles of 120 to -120 degrees. Survey plane, tip and hub, static pressure measurements are built into the survey hardware. PSN exit, hub static pressures are collected with the facility Hyscan system through 4, thick wall tube installations with 90 deg spacing. PSN exit, tip statics are only present for PSN characterization testing, because of the moveable tip shroud. Additionally, PSN entrance hub and tip static pressure measurements; rotor hub entrance static pressures measurements; and PSN exit, tip pitch static pressure measurements are available for PSN troubleshooting.

Development of an instrumentation suite to sense conditions on the rotor proved impractical. High speed, case mounted sensors and telemetry systems were considered, but each required development outside the scope of this project. Previous attempts to capture rotor relative, shock location data resulted in detection of strong probe-boundary layer interaction which masked primary flowpath conditions. Flowpath performance is derived from the rotor entrance and exhaust conditions.

Rotor exhaust conditions are measured with stationary rakes and a performance probe survey. Three, three station, dual element rakes are positioned one duct height downstream of the rotor exit plane. Circumferentially, the rakes are oriented 90 degrees from one another and from the survey position. The three rakes measure exit total pressure and total temperature from a stationary position, downstream of the rotor as shown in Figure 3.2.1-26. The stationary position is important for baselining the performance probe data. The performance probe is mounted to the moving tip ring and therefore can vary its rotor-relative position from one to three duct heights. The stationary rakes allow for monitoring of the circumferential variation of the flow and for comparison of the performance survey data to fixed position data. The performance probe, used for the survey, is a dual-element, Kiel head probe with 45 degree side channels. The side channels allow the probe to collect flow angle data like a cobra probe. Primarily the angle data is used to speed nulling of the probe during transition events between started and unstarted inlet operation. The Kiel head design makes the measurement insensitive to the minor, flow angle changes which occur during throttling. The Performance probe was calibrated in the facility laboratory for flows from 0.1 to 0.9 Mach and flow direction from -120 to 120 deg. The thermocouple is .020 in. diameter, ungrounded, K-type.

Small flow interactions, inherent to the rotating flowpath seals, require mass closure on all system flows for accurate performance tracking. System inputs are: primary flowpath inflow, forward wheel space cavity supply, aft wheel space cavity supply, aft shaft seal supply and drive inflow. Exhausted air includes: primary exhaust, forward rotor bleed, aft rotor bleed, fwd wheel space exhaust, aft wheel space exhaust, and lube system entrained air. Aft shaft seal losses require estimation based on supply pressure and geometry. System, air, inflow and outflow rates are collected with fixed orifice flow meters. Critical flow devices are used on all rig inflows. The rig's, ASME contoured, inflow bellmouth is used as a redundant inflow measurement. For normally aspirated testing the inflow bellmouth acts as the primary inflow measurement. For normal aspiration, the ASME, delta pressure, mass flow equations are reinforced by comparison to measurements from the critical flow venturi used in baseline testing. Exhaust flows are measured with unchoked ASME bellmouths or venturis to minimize total pressure drop. The aft shaft seal loss is estimated from its geometry and supply pressure.

Tip shroud position is determined by axial position and rotor relative location. Because of the strong function of performance on tip clearance, determination of average tip gap and its deviation is crucial. The nominal tip clearance can be calculated from the initial geometry, the tip ring position as determined by actuator's stepper motor count, and by the temperature distribution in the tip ring. Exact matching of the rotor and tip ring, taper angles requires that the components are at design point temperature. A linear array of four thermocouples measures the tip ring temperature over the rotor. The array is replicated at 90 degrees to inspect for circumferential variation. A second method is used to directly measure the shroud to rotor gap. Eight capacitance probes are mounted in the tip shroud at a known distance from the uncontacted abradable surface. The probes are arranged in two planes of 4 probes. In-plane, the probes are separated by 90 degrees. In addition to clearance, this arrangement allows redundant detection of static and rotating, rotor eccentricity. The probes detect the change in capacitance associated with tip passing. The capacitance system uses Capacitec 6100 amplifiers and probes. Both clearance methods determine only the gap between the shroud and the rotor. Determination of the gap between the contacted abradable surface and the rotor requires careful tracking of the test conditions run.

Overall rotor performance is a strong function of the bleed flow conditions. Forward and aft bleed flow data is collected in the stationary reference frame. Three static temperatures and six static pressures are collected on the hub surface of the bleed flowpath. Measurements are taken in three planes. Each plane has 120 degree symmetry for redundancy and detection of circumferential distribution. Static pressure is measured in close proximity to the rotor exit plane and several duct heights downstream. Temperature is taken with ungrounded, K-type thermocouples.

Drive power is determined from temperature drop across the impeller. Four temperature measurements are collected in the inflow and exit planes. . K-type, ungrounded thermocouples with 90 degree circumferential spacing are used. One thermocouple in each plane is shrouded to ensure total temperature conditions are captured. Mass flow is determined upstream in a calibrated flow tube.

Humidity is monitored for normally aspirated tests. For primary testing using facility, the air is considered dry. Facility air is provided with a -40 degree Fahrenheit or better dew point. For normal aspiration tests, inflow humidity is determined from local airport conditions and from a humidity device located in the rig's settling chamber.

3.2.1.3 Control and Data Acquisition

The Rampressor 2 control and data acquisition systems are comprised of several independent systems. To facilitate rapid integration of the Ramgen and the facility control systems, minimal interaction between the systems is required. Facility systems are used to control flows into the rig and to collect performance related data. Operational controls are managed by Ramgen systems. Three systems are utilized for operational control: Ramgen PLC, drive turbine controller and tip shroud controller. The use of independent systems results in redundant operational visibility in the event of control system lock-up. Either system contains enough information to safely stop the system in the event of the loss of the other system. Ramgen and facility systems are connected using two, binary, shutdown signals. In fully automatic operation, an interruption in normal operation of either control system initiates shutdown procedures in both systems.

3.2.1.3.1 Boeing PLC

Facility control and data acquisition is accomplished through the facility's PLC. The facility PLC manages the control valve operation and mass flow calculation for eight facility air legs, four supply legs and four vacuum legs, in addition to two heat exchangers. The facility PLC also coordinates data acquisition for integrated pressure acquisition and temperature devices; as well as, stand alone, voltage input devices. Independent devices include the facility's 256 channel Hyscan pressure system and the 300 channel temperature acquisition system. The PLC accommodates information transfer of 100 facility parameter channels at 0.25 Hz to Ramgen's data reduction computer. The facility PLC also manages steady-state data collection of all channels during set-point acquisition. Set-point data is manually initiated during test. 1000 scans of each data channel are averaged to produce a data set. The 0.25 Hz data is received by a Ramgen PC during test and the set-point data sets are processed after test. The software generates Rampressor performance calculations based on both data sets.

3.2.1.3.2 Ramgen PLC

Ramgen's PLC provides operational control for the Rampressor 2 rig. The Allen Bradley PLC4 manages disc thrust controls and rotor throttling; as well as, lubrication system operation. Additionally, the PLC controls data acquisition for all parameters discussed in section 3.2.1.2.2, Rig Health Monitoring Instrumentation. Ramgen's PLC acquires data at 1 Hz.

3.2.1.3.3 Drive Turbine Controller

The Barber Stockwell turbine, used to drive the Rampressor 2 rotor, is controlled by the manufacture's control system. The system allows operation by valve position or speed control. The system uses pressure control to manage system speed and acceleration rates. The system is operated independently. Secondary control is provided by the facility through operation of a fast-acting, shut-off valve on the drive supply. The control system speed signal is parsed to the Ramgen and facility PLCs for condition monitoring. The drive system updates at 20 HZ and stores data on speed, valve position and user input, control parameters.

3.2.1.3.4 Tip Shroud Controller

The tip shroud is operated using two independent systems. Parker Hannifin actuator controllers are used for positioning and Capacitec proximity sensors are used for clearance monitoring. Due to the proprietary nature of the Parker Hannifin, actuator controller the system is not integrated into Ramgen's PLC. Independent operation of this system is deemed acceptable since the system operation has only a secondary impact on operational safety, through its impact on the thrust load. To ensure safe operation the travel rate is limited in the software to prevent damaging incursion rates of the rotor into the abradable. Measurement of the clearance gap is recorded with set-point data capture of the Capacitec system data. Data acquisition is performed using a Tektronic, 4 channel oscilloscope. User controlled, acquisition rates up to 5 MHz are possible.

3.2.1.4 Facility

The Boeing Nozzle Test Facility was selected for Rampressor 2 testing. Candidate facilities were evaluated on capability, cost and availability. The Rampressor test required:

- Compressed Air 1: 8 lbm/sec continuous, air flow at ambient temperature and 50 psia.
- Compressed Air 2: 10 lbm/sec continuous, dry air flow at 350 degrees Fahrenheit and 250 psig OR a rotor drive capable of 500 Hp at 40,000 RPM.
- Compressed Air 3: 0.25 lbm/sec continuous, air at ambient temperature and 120 psig.
- Vacuum Source: 1.5 lbm/sec air flow at 600 degrees Fahrenheit and 5 psia.

Facility instrumentation and build support were also considered. Eleven facilities received second tier scrutiny. Rental of space, equipment and instrumentation was evaluated. The Nozzle test facility was ultimately selected because of its superior extensive flow capability and its instrumentation suite.

Air resources at Boeing's Nozzle Test Facility fulfilled the Rampressor 2 test requirements. Within the Nozzle Test Facility, Boeing operates two test cells. Test Cell #1 contains 4 independently controlled, compressed air lines; 4 independently controlled, vacuum lines; and a central shop air circuit. These systems were used to satisfy the Rampressor flow requirements. Each compressed air leg can flow air to 300 psig. 2 of the 4 legs are connected to a boost circuit, and are capable of 1000 psig. Total system capacity is approximately 75 lbm/sec at 300 psig. This resource is shared by several cells in the Nozzle Test area. To avoid supply pressure variation and test conflicts with other users on the 1000 psig system, the high pressure legs were not incorporated in the test design. Primary air for the Rampressor 2 is provided on the A leg of the 300 psig system. Supply air for the drive is drawn from the B leg of the 300 psig system. The flow is split and passed through two steam heat exchangers. The heat exchangers are capable of elevating the temperature of 8 lbm/sec of air to 350 degrees Fahrenheit. Downstream of the heat exchangers the flows are recombined. The combined flow passed through a 0.5 second safety valve before entering the drive turbine control valves. The close proximity of the safety valve eliminates excess compressed air storage in the system. The minimal system size adds redundancy to the drive turbine shut-off procedure. Three of the four available vacuum legs are utilized to support the Rampressor 2 rig bleeds. All vacuum legs are four inch pipe and are supplied by a common manifold. The forward, rotor bleed is transferred, metered and throttled on a single leg. The aft rotor bleed is flowed through two vacuum legs. The dual leg arrangement allows accurate metering of the starting and started bleed flows. Secondary rig flows, wheel-space and sealing, and control valve air flows are drawn from the shop air system. This system is pressure regulated, but not flow controlled. Shop air remains charged in the event of a facility shut-down. This phenomena is utilized in the shutdown procedure. In the event of an unexpected event in either the Ramgen or the facility controls, facility compressed air valves close. This closure removes motive force from the drive and starves the compressor, but the thrust balance system, which draws air from the shop air source, remain active to maintain disc position during spool down.

Instrumentation resources at Boeing's Nozzle Test Facility support the Rampressor 2 testing. All performance measurements, for the Rampressor flow path, are collected on facility instrumentation. 180 pressure and 120 temperature measurements are collected. The primary benefit is not the channel count; rather, the calibration procedures. Pressure sensors undergo a rigorous pretest calibration. All Hyscan system pressure channels are system calibrated to NIST standards twice daily and intermediate system pressure drift is monitored against dead weight reference channels during test. This procedure offers high confidence result. In addition, the facility set-point data collection procedure utilizes a thousand scan average to define a data point for each data channel.

Boeing Nozzle Test offers additional technical support beyond the requirements of the test. The test cell has 220 and 480 three phase power to support larger electrical equipment. A wide range of forklifts and cranes are available. A large graveyard of test equipment is available for use in subsystem check-out. A surveying group can offer short notice, equipment and expertise in machinery alignment. And an on-site calibration lab has a permanent small scale nozzle test apparatus for flow calibration of rakes and probes. All of these resources were leveraged in the installation of the Rampressor 2 hardware.

Figure 3.2.1-1: Overview of Rampressor Test Rig

Figure 3.2.1-2: Rampressor-2 Test Skid

Skid Structural Deflection in the lateral direction due to a disk post failure as well as maximum applied load due to a strake tip rub at full speed

Figure 3.2.1-3: Structural Analysis of Rampresor-2 Test Skid

Static Structure FDR – Rig Assembly

Initial Assembly

1. Install spline nuts into frame halves.

3. Align and attach the other support frame.

2. Align and attach one support frame.

4. Attach support frames to the skid.
Rough align the support frames.
Separate frames for component installation.

Figure 3.2.1-4: Rampresor-2 Core Rig Support Frames

Figure 3.2.1-5: Structural Analysis Model of Rampressor-2 Core Rig

Figure 3.2.1-6: Rampressor-2 Rotor Assembly mounted in Core Rig

Figure 3.2.1-7: Simply supported Rampressor-2 Rotor

Figure 3.2.1-8: Fwd and Aft Rotor Bearing Lubrication

Figure 3.2.1-9: Forward Rotor bearing lubrication details

Figure 3.2.1-10: Section through Rampressor-2 Rotor showing Air and Oil Seals

Figure 3.2.1-11: Connection between Boeing Facility and Rampressor-2 Rig

Figure 3.2.1-12: Section through Rampressor-2 Rig showing Pre-Swirl Nozzle

45 Psi applied to aerofoil pressure surface

Figure 3.2.1-13: Peak Stresses in Pre-Swirl Nozzle

Figure 3.2.1-14: Section through Rig showing Exhaust Collector

Forward Wheel Space Purge System

Figure 3.2.1-15: Detailed view of Forward Wheel Space Purge System

Aft Wheel Space Purge System

Figure 3.2.1-16: Detailed view of Aft Wheel Space Purge System

Figure 3.2.1-17: Section through Rig showing Rotor Axial Proximity Probe

Figure 3.2.1-18: Section through Rig showing Tip Ring Assembly

Figure 3.2.1-19: Tip Ring Assembly Actuation System

Figure 3.2.1-20: Drive Turbine Mounting System

Figure 3.2.1-21: Flex Coupling used to connect Rampressor-2 Rotor to Drive Turbine

Figure 3.2.1-22: Wheelspace Pressure Measurement locations

Figure 3.2.1-23: Main Shaft Bearing Thermocouple Locations

Figure 3.2.1-24: Cross-section through Rampressor-2 rig showing Thrust Sensor

Figure 3.2.1-25: Location of Frame Mounted Velocimeters

Figure 3.2.1-26: Exhaust Duct Rakes

3.3 Flowpath Design

3.3.1 Static Flowpath

3.3.1.1 PSN

The Rampressor rotor is preceded by a set of inlet guide vanes that set up the incoming velocity vectors for an absolute Mach number of 0.7-0.9 and flow angle that, with the rotor speed, aligns the relative flow with the strakes. These IGVs are referred to at Ramgen as Pre-swirl nozzles, or PSNs. Preliminary design of the PSN blades was accomplished using Concepts NREC's AXCAD software. Additional blade modification and fine-grid CFD simulation was accomplished using Numeca's AutoBlade, and FINE/Turbo codes respectively.

3.3.1.2 Secondary Flows

Figure 3.3.1.2-1 shows the all the secondary flow circuits in the rig. The different colors indicate the various flow streams. The wheel space air that comes in (indicated in green color) not only purges the disk cavity, but also cools the disk and exits out in a different pipe circuit (indicated in blue color). There is a small amount of leakage of that wheel space air that is used for purging the bearing cavity (near the hub seals – indicated by the yellow lines) as well as the a small amount leaking through the labyrinth seals and into the main bleed flow (indicated by the red lines). The requirements for these secondary flows are

- To ensure that the flow allotted for disk cooling is appropriate enough to establish a reasonable thermal gradients as well as a reasonable maximum temperature on the rotor
- To ensure that there is no excessive heat input into the bearing system from the surrounding static structure

Based on the above requirements, using the ANSYS FEM model, the procedure adopted for calculating the appropriate secondary flows is

- (a) Assign a certain amount of disk cooling flow in the wheel space at the an initial inlet condition
- (b) Perform an integrated thermal analysis of disk and the adjacent fluid
- (c) If the requirement for the fluid temperature and/or the metal temperature is not met, reiterate with an updated disk cooling flow.

Figure 3.3.1.2-1 Secondary flow circuit in the Rampressor 2 Rig

3.3.1.3 Tip Clearance

Tip clearance sensitivity CFD simulations were conducted following the technique described in Section 3.3.2.2. These showed the rotor to yield its maximum performance at tip clearances of 0.001 and 0.002 inches, while the rotor performance deteriorated sharply at a tip clearance of 0.004 in (see Table 3.3.1.3-1 below). At such tip clearances, leakage flow over the strake tip occupies a significant enough portion of the rotor passage that it blocks the passage flow and deteriorates the ramp shock system, such that rotor unstart is arrived at lower rotor pressure ratios.

Table 3.3.1.3.1 This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

3.3.1.4 Abradable Testing

This section describes tests undertaken to determine the appropriate abradable material for the Rampressor -2 shroud as well as an appropriate coating for the Rampressor-2 strake tips.

3.3.1.4.1 Introduction

The Rampressor-2 unique geometry and flow path design require a precise tip gap tolerance. One way to obtain the required tight tip gap is to actively rub into the abradable in a precise and controllable manner that leaves a smooth clean surface. Given that the strakes will be intentionally rubbed into the abradable, the soft titanium strakes will require a protective coating to prevent damage or transfer of the strake material. Because of the high speed and the unique angle of the Rampressor-2 strakes there was no existing data to aid in our choice of materials, therefore a course of testing was required to better understand the interaction between the abradable and strake coating.

These tests were undertaken at the Sulzer-Innotec facility in Winterthur, Switzerland

3.3.1.4.2 Goals

The goals of this series of tests were two fold. The first goal was to find an abradable that would wear in a smooth manner. The second goal was to find a strake tip treatment that would withstand the force of the interaction with the abradable and maintain the integrity of the strake tips.

3.3.1.4.3 Test apparatus

The tests were conducted using the Sulzer-Innotec non standard 23.62 inch diameter test rig at a temperature of 698°F produced by a high velocity burner and at a tip speed of 1476 Ft/sec - the maximum speed of their test rig. The blades were mounted on the rotor at an angle of 82 degrees to the rotor rotation which will replicate the strake angle in the Rampressor-2 test rig. The test samples of abradable were mounted on a stepper motor driven specimen stage

3.3.1.4.4 Test parameters

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3.3.1.4.5 Test matrix

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3.3.1.4.6 Conclusion

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3.3.1.5 Exhaust Collector Mechanical Design

The exhaust collector gathers the air flowing out of the compressor main flowpath and directs it to the backpressure valves and mass flow meter. The exhaust collector is an annular duct with a narrow radial slot which allows air to escape with minimal distortion to the upstream flow. The radial slot leads to a manifold connecting to the two valves. One valve is intended for precise control of backpressure and the second is intended to provide very low back pressure for starting. Operating temperatures is up to 600F and the exhaust collector is designed to

accommodate thermal expansion and safely contain pressure with high metal temperatures. The exhaust collector incorporates a sliding seal to the tip ring and static seals to the aft frame all of which must accommodate thermal expansion while maintaining concentricity with the tip ring to avoid applying loads to the tip ring. The exhaust collector is fabricated from welded 304 stainless steel, which provides a factor of safety of over 3:1 compared to yield strength at 600F. The factor of safety is over 10:1 compared to the ultimate strength for the worst case pressure differential of 9 atmospheres.

3.3.1.6 Heat Transfer Analysis

The thermal analysis was conducted using ANSYS. The boundary conditions were generated using the aero mean line predictions in conjunction with the results from the CFD analysis. The requirements for performing the thermal analysis are

- A steady state analysis was needed to predict the static structure displacements as well as the rotating components displacements at various design conditions. This was important in order to set the requisite tip clearance levels.
- Ensure that the bearing loads in the rail system are tolerable. This requirement meant that the temperatures in the core static structure near the linear bearing in the rails needed to be less than 75 F.
- Ensure that the misalignment caused by differential thermal growths between the forward and the aft frame are within the flex coupler design limits. In terms of the temperature, the temperature difference between the two frames near the rotor bearing surface needed to be within 40 F
- Analysis to ensure that the static structure temperatures near the rotor bearings were below 250 F.
- The temperature predictions needed to be in sufficient detail to perform the required structural analysis to predict the life cycle of the rotating components at the running temperatures.

In order to meet the abovementioned requirements for the static structure, cooling channels were required both in the tip ring and in the region where the aft frame was in contact with the rail system. The model used for thermal analysis is shown in figure 3.3.1.6-1. The thermal model comprised of 661,689 elements and 910,443 nodes. The boundary conditions applied to the model are shown in figures 3.3.1.6-2 through 3.3.1.6-4. The boundary conditions are generated using standard convective heat transfer correlations using either flat plate or pipe flow, as appropriate to the region. The predicted full load steady state temperature distribution is shown in figure 3.3.1.6-5. The predicted temperatures at design point for the core static structure as well as the tip ring are shown in figures 3.3.1.6-6 and 3.3.1.6-7. Based on the thermal analysis, the following conclusions can be made

- the temperature distribution on the frames create an acceptable amount of load into the bearings in the rail system
- the predicted temperature difference between the aft and the forward frames is around 40F
- the predicted temperatures near the rotor bearings are below 250 F
- In order to maintain reasonable temperatures on the tip ring, 1.5gpm of water at 70 psia and 60F needs to be circulated in the cooling channels. This results in a heat pick of 10F in the water and requires a chiller with a capacity of 6500 Btu/hr.
- In order to maintain frames at reasonable temperatures so that it does not overload the linear bearings in the rails, around 6gpm to 8gpm of water at 70 psia and 60F is required. This results in a heat pick of 10 F in the water and requires a chiller with a capacity of 35000 Btu/hr.

The rig structural analysis uses the results from the rig thermal analysis to predict the displacements as well as loads in different components of the rig. The requirements for performing the structural analysis are

- To predict the radial and axial displacements of the various components, which will assist in creating the component drawings
- To ensure that the total load transferred into the linear rail bearings is below 13000 lbf.
- To ensure that the axial and radial displacements of various components are not unduly constrained. This results in reasonable stresses in the various components

- To ensure that the difference in radial displacements between the rotor bearing bores due to thermal loading is within 0.0009 inches so that the flex coupler can tolerate the misalignment induced by the differences in thermal behavior.

The model used for structural analysis is shown in 3.3.1.6-8. The structural model comprised of 227,864 elements and 379,335 nodes. The boundary conditions applied to the model is shown in figure 3.3.1.6-9. The predicted design point axial and radial deflections are shown in 3.3.1.6-10 & 3.3.1.6-11. The predicted axial displacements at design point for the core static structure is shown in figures 3.3.1.6-12. This figure indicates that the forward portion of the rig moves aft and the aft portion of the rig moves forward at design point thus closing the separation distance between the two wheel space cavities. The radial displacement plots for the rig is shown in Figure 3.3.1.6-12 and this indicates that the radial growth of the rig is pretty axisymmetric even though the structure itself, by construction, is non-axisymmetric. This predicted axisymmetric deflection shape is expected to be very important in maintaining tight running tip clearances. A good measure of the axisymmetric nature of the radial displacements is noticed using figure 3.3.1.6-13. As can be seen, the growth differs in the 0.000x inches and this the maximum predicted misalignment due to thermals is expected to be 0.00065 inches, which is lower than the 0.0009 inch requirement.

Figure 3.3.1.6-14 indicates that the radial growths for other components are axisymmetric in nature, as well.

Based on the above analysis, the following can be concluded

- The radial and axial displacements of different components of the rig are predicted and is detailed enough to perform hot to cold conversions and thus aid in the drawing creation. The deflections indicate that the displacements are symmetric/axisymmetric and the maximum concentricity issues are about 0.001 inches.
- The total force transmitted to the linear bearings in the rails is about 1000 lbf which is well below the 13000 lbf design limit for these bearings
- The predicted misalignment in radial displacements between the forward and the aft bearing bores is less than the limit of 0.00095 inches (which is $\frac{1}{2}$ of the overall misalignment that the bearing system and the flex coupler can withstand).

The FE thermal model is a 180 degree model that has adiabatic boundary conditions at the plane of symmetry

Figure 3.3.1.6-1. Ansys Thermal Model

Figure 3.3.1.6-2. Thermal Model Boundary Conditions

Figure 3.3.1.6-3. Thermal Model Boundary Conditions

Figure 3.3.1.6-4. Thermal Model Boundary Conditions

Figure 3.3.1.6-5. Design Point Temperature Distribution

Figure 3.3.1.6-6. Core Rig Design Point Temperature Distribution

Figure 3.3.1.6-7. Tip Ring Design Point Temperature Distribution

Figure 3.3.1.6-8 Core Rig Ansys Structural Model

Figure 3.3.1.6-9. Core Rig Structural Model Boundary Conditions

Figure 3.3.1.6-10. Core Rig Axial Deflections (in inches)

Figure 3.3.1.6-11. Core Rig Radial Deflections (in inches)

Figure 3.3.1.6-12. Core Rig Axial Deflections (in inches)

Figure 3.3.1.6-13. Radial Deflections (in inches) near the rotor bearing surfaces

Figure 3.3.1.6-14. Radial Deflections (in inches) of the exhaust shell and bleed liners

3.3.1.7 Containment

The segment containment analysis was done using empirical constants and formulae derived from tests done by Westinghouse. Two different conditions were tested namely case perforation and case burst. The case perforation method is based on the fragment kinetic energy exceeding the shear carrying capability of the case (in our case, the tip ring). The case burst method is based on the fragment kinetic energy exceeding the tensile carrying capability of

the case (tip ring) material. As a result, the case (tip ring) bursts. The relevant figure for this is figure 3.3.1.7-1 and the corresponding equations are

$$\text{Kinetic Energy of the segment} = \frac{1}{2} m r^2 (2\pi \text{rpm}/60)^2 (1 - m / (m+m_2))$$

$$\text{Dynamic shear capability of containment structure} = K\zeta_d P T^2 + A T \epsilon \sigma_d$$

$$\text{Tensile carrying capability of containment structure} = Q \epsilon \sigma_d$$

where,

m = mass of components released during failure

m_2 = effective case mass

ζ_d = dynamic shear strength

σ_d = average dynamic plastic flow

ϵ = plastic strain

r = radial cg of the components

b_x = axial length of the segment

T = case thickness

P = perimeter of the impact

D = Inner diameter of case

K = experimentally determined constant (got from literature)

Q = volume of containment structure

Using the above equations, the total kinetic energy of the fragment (which is defined as one disk post plus two segments) is 41,249 lbf-in and the total dynamic shear carrying capability of a ½ inch thick tip ring is 63,651 lbf-in and the total dynamic tensile carrying capability of a ½ inch thick tip ring is 474,374 lbf-in. Since, kinetic energy of the fragment is less than the dynamic shear and tensile carrying capacity of the tip ring, a disk post failure will be captured by the surrounding tip ring.

In addition to the fragment containment analysis, an additional analysis was done using ANSYS to ensure that the maximum stress in the core rig due to the disk post failure is below the ultimate tensile strength (UTS) of the core rig material. The model used for this analysis is shown in figure 3.3.1.7-2. This analysis was done by calculating the radial load arising due to a disk post failure event at full speed and applying that to the static structure at the bearing retention location. A sensitivity study (10 cases in all) was done assuming different bearing loads at different theta locations to calculate the most stressful condition for static structure. As figure 3.3.1.7-2 indicates, the loads are applied at several different angular locations namely 0 degrees (top dead center), 45 degrees, 90 degrees, 135 degrees and 180 degrees (bottom dead center). Based on the sensitivity study, the worst case stresses were generated in the weakest portion of the core rig, which are the struts, when a 40000 lbf load was applied directly in line with the struts (figure 3.3.1.7-3). The peak stresses here were 108ksi which is below the UTS and even the yield strength (140 ksi) for this material. This indicates that the rig can handle loads generated due to a disk post failure event.

A full report of this particular work is provided in the Chapter 3 Appendix

Figure 3.3.1.7-1. Fragment containment critical criteria

Figure 3.3.1.7-2. Ansys model for analyzing the effect of disk post failure on the static structure

Figure 3.3.1.7-3. Maximum predicted stress in the core rig static structure due to a disk post failure

3.3.2 Rotor Flowpath Aerodynamics

3.3.2.1 Mean Line Design

The preliminary design variables such as flow-path area ratios and shaft speed were generated with Ramgen Power Systems' performance prediction system named CADRE. This one-dimensional mean-line code is used for the design and analysis of Rampressor-based equipment. CADRE conserves mass, momentum, and energy at various stations through a Rampressor starting at the inlet and ending after a diffuser/de-swirl component. Fluid properties are calculated based on the NIST REFPROP 7 (Reference Fluid Thermodynamic and Transport Properties Database, NIST Standard Reference Database 23). No ideal gas assumptions are used in the calculations, making the code suitable for gases such as CO₂ and various refrigerants.

The code models a Rampressor as four distinct components: 1) booster stage, 2) pre-swirl nozzle (PSN), 3) rotor, and 4) diffuser/de-swirl (DDS). The booster stage model takes a pressure ratio and an efficiency to calculate modified inlet conditions for the PSN, but is rarely used as most designs are normally aspirated. The PSN uses a constant θ model to compute total pressure loss as a function of PSN exit Mach number. The model subdivides the rotor into a number of sections and employs generalized one-dimensional compressible flow equations (e.g., Shapiro [7]). These equations account for area change, total pressure loss, and mass addition and removal. In the rotating reference frame, rothalpy (rotating enthalpy) is conserved along with mass.

To achieve the stated performance goals, an appropriate shock system must be set up in the ramp section of the rotor. The pattern of shock waves shown in Figure 3.3.2.1-2 (lower) is comparable to the pattern of shock waves that might be found in a supersonic intake of a missile or supersonic aircraft. A schematic illustration of a typical planar 2D supersonic intake as would be applied on an aircraft or missile is shown in Figure 3.3.2.1-2 (upper) along with the associated oblique and normal shock waves. For inlet relative Mach numbers ranging from 1.5 to 3.5, it is common practice to design the intake so as to establish a series of oblique shock waves that decelerate the supersonic flow entering the inflow plane from the inlet relative Mach number to a pre-normal shock Mach number of ~1.3 (Mahoney [8]; Seddon et al. [9]).

Total pressure loss through the oblique shock system is determined with curve fits of available Rampressor compression ramp geometries for a variety of inlet Mach numbers. Pressure loss terms in the various sections of the rotor are entered to account for viscous effects, some shock interactions with the inlet, and tip leakage. Ideally,

these loss factors would utilize empirical formulae derived from an extensive database of both component and full rig test results. However, while Ramgen is in the process of developing such a database, one is not currently available - so it is necessary to use available literature and engineering judgment in determining these loss factors. The terminating shock wave located after the throat is modeled from first principles. This normal shock, whose loss scales with pre-normal shock Mach number M_1 , as $(M_1-1)^3$ (e.g., Freeman & Cumpsty [10]), is usually constrained in the design phase to have an incoming Mach number less than 1.3 to minimize total pressure loss in the flow. The strong relationship between pre-normal shock Mach number and the total pressure loss makes this design parameter of primary importance. While the effect of the bow shock in front of the strake leading edge is not explicitly accounted for with a unique loss term, its strength is minimized through the design of the leading edge (see CFD Analysis of the Rampressor Rotor section for further detail). The design code calculates required rotor geometry according to user inputs for mass addition or removal and mean-line Mach number at the throat. The DDS is modeled using a constant θ , like the PSN.

Some key results of the meanline model are found in Table 3.3.2.1-1. Figure 3.3.2.1-1 shows the rotor in profile and may be useful for determining the locations of the various stations. Note that only the first and last rows are in the stationary reference frame and that all other values are for the rotating reference frame. Thus the total pressure listed are relative total pressures and not absolute. Also, the areas are for a single flowpath, and thus don't apply in the stationary reference frame. The capture area of flowpath is listed in the rotor entrance row. These areas are idealized and do not represent the exact helical geometry which is more complex as the distance between the strakes is a function of the height of the strake—this is the nature of a helix. Also, the compression from the side wall change in thickness is taken into account.

3.3.2.2 Compression Ramp Design

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Table 3.3.2.1-1: Summary of Mean Line Design

Location	Total Pressure (psia)	Static Pressure (psia)	Total Temperature (°F)	Static Temperature (°F)	Mach (—)	Flowpath Area (in ²)
Exit from PSN (static frame)	13.76	9.97	49.98	5.08	0.694	—
Rotor Entrance (rotating frame)	146.03	9.97	534.64	5.08	2.40	0.875
Throat (rotating frame)	143.86	59.37	554.65	332.58	1.20	0.354
Pre Normal Shock (rotating frame)	140.66	50.78	554.01	302.73	1.30	0.368
Diffuser Exit (rotating frame)	129.45	105.94	549.16	493.68	0.55	0.467
Rotor Exit (static frame)	148.67	105.94	587.22	493.68	0.72	—

Figure 3.3.2.1-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992

Figure 3.3.2.1-2: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992

3.3.2.3 3D CFD Performance Analysis

3.3.2.3.1 3D Viscous CFD

Solution of the compressible form of the Navier-Stokes equations for the simulations presented herein is conducted using a finite-volume, density based scheme that has been modeled after the explicit 4-stage Runge-Kutta time integration method of Jameson, Schmidt and Turkel [13]. Implementation of this numerical algorithm is conducted in the FINE/Turbo code of Numeca, described by Hakimi [14]. Multigrid acceleration is employed to damp varying error wavelengths efficiently. A cell-centered second-order central discretization of space is conducted, employing scalar artificial dissipation through a blend of second and fourth differences to avoid odd/even decoupling and oscillations in the vicinity of high pressure gradients (see eqn's. 1.1-1.5). The particular treatment of dissipation is that of Martinelli and Jameson [15]. Equation 1.1 presents the integral form of the solution and flux vectors of the continuity, momentum, and energy equations, along with the source term vector (e.g., turbulence source terms) and artificial dissipation terms. It is not desired to provide full details of the code employed, but it is necessary to discuss a few key features to aid following discussions. The second and fourth differences are laid out in equations 1.2 – 1.5, where Δ and ∇ are forward and backward difference operators. The spectral radii of the flux Jacobian matrices, denoted by λ , are scaling parameters in the dissipation terms and v is a pressure gradient sensor employed in the dissipation logic. Within the calculation of first- and third-order adaptive dissipation coefficients, $\varepsilon^{(2)}$ and $\varepsilon^{(4)}$ respectively, two constants, $\kappa^{(2)}$ and $\kappa^{(4)}$, are available that can be controlled by the user to increase or decrease the amount of dissipation injected into a solution. A delicate balance must be maintained when setting these values to enable reasonable convergence rate and level while maximizing solution accuracy. Generally, higher values of these constants can be employed when finer computational grid density is used - and it is instructive to determine sensitivity to these values when testing the dependence of a solution to grid resolution.

$$\int_V \left(\underbrace{\underline{Q}_t}_{\text{vector of conserved variables}} + \underbrace{E_x + F_y + G_z}_{\text{flux vectors}} \right) dV - \int_V \underbrace{\underline{S}}_{\text{source term vector}} dV - \underbrace{DQ}_{\text{artificial dissipation terms}} = 0 \quad (1.1)$$

$$D^{(2)}Q = \nabla \left((\lambda)_{i+\frac{1}{2},j,k} (\varepsilon^{(2)})_{i+\frac{1}{2},j,k} \right) \Delta Q_{i,j,k} \quad (1.2)$$

$$D^{(4)}Q = \nabla \left((\lambda)_{i+\frac{1}{2},j,k} (\varepsilon^{(4)})_{i+\frac{1}{2},j,k} \right) \Delta \nabla \Delta Q_{i,j,k} \quad (1.3)$$

$$\varepsilon_{i+\frac{1}{2},j,k}^{(2)} = \kappa^{(2)} (\lambda)_{i+\frac{1}{2},j,k} \max(v_{i+2,j,k}, v_{i+1,j,k}, v_{i,j,k}, v_{i-1,j,k}) \quad (1.4)$$

$$\varepsilon_{i+\frac{1}{2},j,k}^{(4)} = -\max(0, \kappa^{(4)} (\lambda)_{i+\frac{1}{2},j,k} - \varepsilon_{i+\frac{1}{2},j,k}^{(2)}) \quad (1.5)$$

Turbulence is modeled using the one equation closure of Spalart and Allmaras – *S-A* – [16]. The model has been chosen based on its demonstrated accuracy of predicting boundary layer development (Bardina et. al. [17]), separation in diffusing flows (Yaras & Grosvenor [18]), turbomachinery flows (Heidegger et. al. [19]) and shock wave / boundary layer interaction (Mohler [20]). The model was particularly chosen over other two equation models that offer similar prediction accuracy - or higher order closures that would require dramatically higher grid resolution and/or CPU time to achieve convergence - due to its high numerical robustness, lower sensitivity to grid resolution and computational economy (e.g., Hellsten [21] & Pope [22]).

Validation of CFD Scheme

While development of a Rampressor rotor is initiated with mean-line analysis and method of characteristics

calculations, the design is heavily dependent on design-cycle CFD. This unique rotor incorporates several positive features of both conventional turbomachinery and aircraft engine intakes, as well as several of the related aerodynamic complexities. In order to trust CFD predictions that guided design modifications, it was necessary to understand how well the employed numerical scheme and turbulence model could capture the physics of boundary layers, diffusing flows and separation, shock wave / boundary layer interaction, and features specific to compressor aerodynamics. Three suitable experiments were chosen from the literature, and results of the test case predictions are presented herein.

Transonic separating diffuser

The first test case run was the transonic diffuser of Bogar, Sajben and Kroutil [23]. It was chosen to test the ability of the employed numerical scheme and turbulence model to predict flow separation in presence of the strong shock wave / boundary layer interaction produced in this diffuser. The entrance to throat area ratio was 1.4, and the exit to throat area ratio was 1.5. Reasonable two-dimensionality of the flow was produced with a sidewall spacing of approximately four throat heights, and suction slots placed on the side walls of the constant area sections upstream and downstream of the diffuser and along the top corners of the diffuser. The throat height was 44mm (1.7in) and an exit static pressure to inflow total pressure ratio of 0.72 was applied.

Figure 3.3.2.2-1 displays a schematic of the two-dimensional diffuser geometry, the flow field, and the computational grid employed. Two grids were tested employing 89 nodes in the streamwise direction and 61 in the radial, with inlet y_1^+ values of 1 and 5. Total pressure, total temperature, and 1% turbulence intensity were specified at the inflow boundary based on experimental measurements. The outflow static pressure was set to an average (less reflective) condition, and all other flow properties were extrapolated from the interior of the computational domain.

Comparisons between prediction and experiment are shown in Figures 3.3.2.2-2 and 3.3.2.2-3. Flow separation due to the shock wave / boundary layer interaction, and subsequent pressure recovery characteristics are shown to be predicted quite well. However, decreasing the 3rd-order artificial dissipation coefficient constant, $\kappa^{(4)}$, produced slightly better boundary layer displacement due to separation and subsequent local reduction in pressure recovery. Note that variation of the 1st-order artificial dissipation coefficient constant, $\kappa^{(2)}$, was also tested due to its importance in the prediction of strong pressure gradient flows. Values between 0.25 and 0.5 were run, but no appreciable difference in results was observed. It was generally concluded that boundary layer development and flow separation due to a strong shock wave / boundary layer interaction could be adequately captured by the employed numerical scheme and turbulence model.

4:1 pressure ratio centrifugal compressor impeller

The next validation test case run was the centrifugal compressor impeller described by McKain and Holbrook [24]. It was chosen to test the ability of the employed numerical scheme and turbulence model to predict performance of a conventional compressor rotor at on- and off-design conditions. Although the Rampressor rotor employs an unconventional technique to achieve compression, its performance is still dependent on such items as tip clearance and passage aspect ratio (blade span/chord length). This compressor produces a reasonably high stage total pressure ratio, adequate information is given on the rotor tip clearance, and the quality of data is quite good. Details of the experimental measurements are given by Skoch et al. [25]. The impeller contains 15 main blades with 15 splitter blades, and employs 50 degrees of backsweep from radial at discharge. The splitter blade leading edge - located at 30% of main blade chord - is offset slightly toward the main blade suction surface in order to produce an even flow split. The inlet tip diameter is 210 mm (8.26 in) and the inlet blade height is 64 mm (2.52 in). The exit diameter is 431 mm (16.97 in), the exit blade height is 17 mm (0.67 in) and the exit tip clearance is 0.203 mm (0.008 in). The stage (impeller with vaned diffuser) was designed to produce a pressure ratio of 4:1 at the design mass flow. The standard day corrected speed for the design flow of 4.54 kg/s (10 Lbm/s) was 21,789 rpm. The inlet relative Mach numbers on the suction surface ranged from 0.45 at the hub to 0.9 at the tip.

Figure 3.3.2.2-4 displays four views of the three-dimensional impeller computational grid employed. Note that a single blade and splitter channel was modeled, and periodic boundary conditions were employed to reproduce the 360° impeller. Two grids were tested employing approximately 3.8×10^5 and 3 million nodes, with y_1^+ values ranging from 1-5 and 0.1-0.5 in the passage respectively. A complex multi-block structure was employed, but the node distributions were approximately 177×49×45 in the streamwise, pitchwise and spanwise directions for the coarser grid, and 353×97×89 for the finer. Total pressure, total temperature and 1% turbulence intensity were specified at the inflow boundary based on the experimental measurements. The exit mass flow was set to a range of values to produce a constant speed line, and all other flow properties were extrapolated from the interior of the

computational domain. A real-gas definition of air was used for the working fluid.

Comparisons between a predicted constant speed line and experiment are shown in Figures 3.3.2.2-5 and 3.3.2.2-6. Pressure ratio and efficiency at the maximum efficiency point are shown to be predicted quite well by the fine grid, with the predicted accuracy reducing somewhat at off-design. The coarse grid also compared well, when the 3rd-order artificial dissipation coefficient constant, $\kappa^{(4)}$, was reduced to 0.001. Interestingly, at low massflow the results of both coarse grid simulations matched regardless of the 3rd-order artificial dissipation coefficient constant, $\kappa^{(4)}$. This difference in behavior exists because at this point on the constant speed line, the impeller is near surge, and the lower grid resolution is too coarse to fully capture the boundary layer and tip clearance flow in this condition. The results of the coarse grid with minimized dissipation showed a slight dip in characteristic that could indicate approach of surge. However, convergence was stable at these points and the magnitude of difference between these predicted operating points with the fine grid results, and experiment, were small. The numerical scheme and turbulence model employed were deemed adequate for the purposes of predicting compressor rotor performance at design and off-design – particularly when comparing with other published predictions (e.g., Larosiliere et al. [26]).

1.6:1 pressure ratio transonic fan rotor

The last validation test case run was the transonic fan rotor, commonly known as Rotor 67, described by Strazisar et al. [27] and later by Wood et al. [28]. It was chosen to test the ability of the employed numerical scheme and turbulence model to predict performance of a conventional axial transonic compressor rotor. Although generated by different means, both the Rampressor rotor and Rotor 67 contain a leading edge and passage shock system that must be predicted correctly. The rotor contains 22 blades. The tip radius at leading edge is 25.7 cm (10.1 in), and the hub-tip ratio ranges from 0.375 to 0.478. The tip clearance is 0.61 mm (0.024 in). The standard day corrected speed for the design flow of 33.25 kg/s (73.3 Lbm/s) is 16,043 rpm, and the tip LE relative Mach number is 1.38.

Figure 3.3.2.2-7 displays three views of the three-dimensional rotor computational grid employed. Note that a single blade passage was modeled, and periodic boundary conditions were employed to reproduce the 360° row. Based on the previous results, one grid was generated employing approximately 2.3×10^5 nodes, with y_1^+ values ranging from 1-5 in the passage respectively and $\kappa^{(4)}$ was set to 0.001. A complex multi-block structure was employed, but the node distributions were approximately $89 \times 41 \times 61$ in the streamwise, pitchwise, and spanwise directions respectively. Profiles of total pressure, total temperature, and 1% turbulence intensity were specified at the inflow boundary based on the experimental measurements. The exit mass flow was set to a range of values to produce a constant speed line, and all other flow properties were extrapolated from the interior of the computational domain. A real-gas definition of air was used for the working fluid.

Comparisons between Mach number distributions plotted at 10% span from tip at the maximum efficiency operating point are displayed in Figure 3.3.2.2-8. Consistent LE and passage shock structure, and matching spot values of relative Mach number in key areas are shown. The numerical scheme and turbulence model employed were found to adequately capture the leading edge and passage shock structure of this transonic axial fan rotor.

CFD ANALYSIS OF THE RAMPRESSOR ROTOR

As was discussed earlier, the potential for 1D prediction of losses and ultimate performance of the Rampressor is limited due to the absence of the types of well established correlations and design practices available for conventional turbomachinery design. In fact it is generally accepted that the same is true for any supersonic compressor concept, and that CFD must be relied upon to predict losses in such designs (e.g., Cumpsty & Freeman [29]).

The numerical scheme and turbulence model described earlier has been used to verify and refine the Rampressor design described in the previous Design Methodology section. Successful prediction of Rampressor rotor performance at design-point and off-design requires accurate capture of boundary layer development, diffusing flows and separation, shock wave / boundary layer interaction, and a variety of compressor-related phenomena such as tip clearance flow and passage secondary flow vorticity. Validation of the CFD scheme and the Spalart-Allmaras turbulence model presented in the previous section combined with the cited previous validation literature provided reasonable evidence that these features could be adequately captured, and the observations provided direction in what grid resolution should be employed to achieve successful prediction. Using this guidance led to a grid resolution of approximately 7 million cells in the Rampressor rotor passage.

Reduced flow-path model

Using the initial geometry definition techniques described in the previous sections, a number of candidate designs were developed that were intended to capture inflow at an approximate Mach number of 2.4 and use a series of oblique shock waves to compress and decelerate the flow to a low supersonic Mach number at the flow-path minimum area or throat. At design pressure ratios, a normal shock would be stabilized downstream of the throat intended to decelerate the flow to subsonic Mach numbers. The flow would then experience additional subsonic diffusion until the desired rotor discharge Mach number was achieved.

In order to compare designs and assess their performance relatively quickly, a reduced model with fewer grid points than the full rotor model was developed that encompassed only the section of flow-path between rotor leading edge, and a short distance aft of the trailing edge. Figure 3.3.2.2-9 identifies components of the rotor included in the model, and the location of boundaries. This model was used primarily to evaluate shock structures developed by different flow-paths, rather than other phenomena listed above, such as tip clearance and secondary flows. The flow-field is based on a geometry where the strake surface used to define the upstream boundary of the flow-path has a sharp leading edge and the wall itself has no thickness. This simplification eliminates the generation of a leading edge bow shock. Slip surfaces were applied at the rim, shroud, and strakes. Velocity vectors, static temperature, and low turbulence intensity (~1%) were specified at the inflow boundary to produce Mach 2.4 flow with zero incidence to the strakes, and a radial-equilibrium distribution of static pressure was set at the outflow boundary. A real-gas definition of air was used for the working fluid.

The general grid structure used for these reduced model simulations is shown in Figure 3.3.2.2-9. A large number of design variations were considered using the baseline grid resolution displayed in the figure, of approximately 3×10^5 nodes (233 streamwise, 29 spanwise and 49 pitchwise). However, to obtain a more detailed picture of the compression ramp shock structure, a limited number of simulations were run at a much higher resolution of approximately 11×10^6 nodes (889 streamwise, 113 spanwise and 113 pitchwise). In this case, the grid distribution was nearly uniform in the pitchwise and spanwise directions – with a bias of streamwise resolution toward the ramp section.

The remainder of this Section intentionally left blank under the provisions of The Energy Policy Act of 1992.

Viscous rotor passage model

The next step of design after reaching an optimum ramp configuration using the reduced flow-path was to test the design in presence of boundary layers, leading edge, tip clearance and other compressor related features. A viscous rotor passage model was constructed. Figure 3.3.2.2-13 displays the computational domain for the rotor flow-path and related boundaries. Total pressure, total temperature, flow angle, and low turbulence intensity (~1%) were specified at the inflow boundary to produce Mach 2.4 flow with zero incidence to the strakes, and an average (less reflective) static pressure condition was set at the outflow boundary. All solid surfaces were set as no-slip, with strake and the axial region of rim representative of the rotor width set to design rotation speed. A real-gas definition of air was used for the working fluid. Flow control techniques are applied in this design to combat shock wave boundary layer interaction. These devices were included in the present computational model, but they are not described due to their proprietary nature.

The computational grid used for the Rampressor rotor defocused flow-path simulation is displayed in Figure 3.3.2.2-14, showing rim and strake surfaces. The employed grid resolution was chosen to capture the variety of aerodynamic phenomena discussed earlier, such as passage shock system, secondary and tip clearance flows. In the rotor passage in between the strakes, $937 \times 65 \times 89$ grid nodes were specified in the streamwise, pitchwise and spanwise directions respectively. In front of the strakes, $737 \times 33 \times 65$ grid nodes were employed and $297 \times 33 \times 65$ grid nodes were set aft of the strakes. The resulting rotor passage grid contained approximately 7.6×10^6 nodes.

Clustering of the grid points near solid surfaces in the pitchwise and spanwise directions was prescribed to yield y_1^+ values near to 1. In reality, y_1^+ ranged between 0.1 and 5 in various parts of the passage due to the nature of local acceleration and deceleration of the flow, but it was verified that the grid density and clustering were in line with the findings of the earlier validation studies, such that predictions would not suffer from significant artificial diffusion, and the location of the first node from the surface was verified to be in the range needed for the Spalart-Allmaras turbulence model everywhere. In the streamwise direction, clustering was employed to capture the strake leading and trailing edges, as well as the throat curvature.

Note that a non-conformal multi-block grid structure was employed in a few key areas. For instance, Grid

quality was maximized in the vicinity of the swept leading edge, by employing small blocks upstream of the LE that were fit to its geometry, and interfacing them with the rest of the passage grid that employed radial grid lines in the spanwise direction (as indicated in Fig. 3.3.2.2-14). Likewise, the grid structure employed adjacent to each periodic boundary was set up independently of one another, and then communication from one to the other was set through non-conformal interfaces. In the case of such interfaces in FINE/Turbo, the connecting surface is decomposed into polygonal elements based on the resolution of mating grid surfaces. The solver calculates fluxes through these elements, and transfers them to corresponding cells on both sides of the connection. Ghost cell values at the interface are then calculated through a weighted averaging procedure to preserve mass conservation (Demeulenaere [34]).

Viscous rotor passage models of the focused and defocused designs are compared qualitatively in Figure 3.3.2.2-15 in terms of relative Mach number distributions at mid-pitch. Centerline relative Mach number and absolute total pressure distributions are displayed in Figures 3.3.2.2-16 and 3.3.2.2-17. Compared to the reduced flow-path model, these simulations incorporated such effects as boundary layers, leading edge shocks, and tip clearance flow. The presence of boundary layers highlighted the importance of minimizing Mach number upstream of the terminating shock to avoid significant flow separation. The LE shock was found to be of lower importance for reasons discussed below. Tip clearance, as one would expect in any kind of rotor with such a high pressure ratio, had a strong influence on performance and produced high levels of blockage at the shroud. Clearances tested ranged from 0.0254 mm (0.001in) to 0.254 mm (0.01in). The rotor test rig that has been built employs a translating abradable shroud such that a wide range of clearances can be tested. Tip clearance was identified as a major issue for Rampressor performance at the beginning of the first design program, and ongoing efforts in rotor design and the implementation of flow control techniques are being conducted to minimize its impact, as well as that of other factors such as inflow distortion.

The results of several viscous CFD studies showed some intriguing trends regarding boundary layer development in the rotor passage. In contrast to conventional transonic and supersonic rotor designs, the very high hub-tip ratio inherent in the Rampressor design results in essentially no untwist of the strakes due to centrifugal loading – which leads to higher flexibility in strake design. A drawback of the high hub-tip ratio is the resultant close proximity of endwall boundary layers to one another, and a subsequent potential for such a low aspect ratio blade row to suffer from strong secondary flow losses. However, viscous rotor passage CFD studies have shown that the combination of very high stagger and high tip speed results in a shroud that rotates closely aligned to the opposite direction of boundary layer development, and therefore acts directly to thin the shroud boundary layer. This effect in combination with certain flow control techniques employed in the design tend to compensate for endwall boundary layer development that would otherwise dominate the flow-field.

The remainder of this Section intentionally left blank under the provisions of The Energy Policy Act of 1992.

Some comments on off-design performance

The design presented herein has been developed to prove the Rampressor concept for high pressure ratios. This is the second such major rotor design that has been carried out, and a series of rotating rig tests based on this design will be completed in 2006. This design has concentrated on peak performance, and it is expected that off-design performance will be given higher priority in following design studies. However, a few points can be made on the subject here. It is widely known that with increasing inflow Mach number axial compressors suffer from significant increase in losses and furthermore, that sensitivity to incidence is substantially increased.

Typical supersonic compressor designs employ LE thicknesses of 1.5-3% staggered gap and maximum blade thickness of 6-8% staggered gap (Freeman & Cumpsty [10]).

The Rampressor design minimizes LE shock loss by employing an LE thickness of approximately 0.4% staggered gap with significant sweep and 4% strake thickness in the section of the passage containing the shock system. It can be seen clearly from the viscous rotor passage CFD results presented in Figure 3.3.2.2-18 that only an extremely weak shock wave is generated at the leading edge on the upstream side of the strake. The LE shock wave propagating inside the passage (dashed line) is similar in strength to those seen in conventional designs (e.g., compare change in Mach number to that of the transonic fan rotor in Fig. 3.3.2.2-8), but it's interaction with the oblique shock system generated by the ramp (dash-dot line) is benign. Furthermore, the leading edge shock does not move or vary in strength as a function of rotor back-pressure, as it does in conventional designs. Instead, the terminating normal shock wave indicated in Figure 3.3.2.2-15 moves inside the diffuser section, while the oblique shock system remains stationary. Cumpsty [35] describes the operation of a number of supersonic compressor

designs that have been tested. These designs employed a combination of negative camber and relatively high blade count to produce a series of oblique shocks and minimize relative Mach number upstream of the terminating passage shock. This is in general the same process used in the Rampressor design method, where the ramp is designed to produce a series of weak oblique shocks, and a contraction ratio is selected to minimize the relative Mach number upstream of the normal shock. The difference is in the significantly higher degree of freedom of profile and contraction ratio that a rim of the type in the Rampressor affords versus the cambering of thin blades. Conventional designs have been observed to be highly sensitive to off-design conditions because the passage shock system would move significantly with moderate changes in incidence - resulting in acceleration where diffusion should occur for example. Some techniques, particularly suitable to the Rampressor rotor design, have been identified that would enable the terminating shock to move in a section of the passage with minimal impact on passage aerodynamics (hence change in loss) and these are being studied presently.

Figure 3.3.2.2-1. Schematic of separating transonic diffuser flow and computational grid.

Figure 3.3.2.2-2. Predicted static pressure for separating transonic diffuser

Figure 3.3.2.2-3. Predicted velocity profiles for separating transonic diffuser

Figure 3.3.2.2-4. Computational grid used for 4:1 centrifugal compressor simulations (3 million cell version displayed) a) trailing edge at hub, b) leading edge at hub, c) tip clearance at leading edge

Figure 3.3.2.2-5. Predicted pressure ratio for the 4:1 centrifugal impeller at 100% speed.

Figure 3.3.2.2-6. Predicted efficiency for the 4:1 centrifugal impeller at 100% speed.

Figure 3.3.2.2-7. Computational grid used for 1.6:1 transonic fan rotor. a) leading edge at hub, b) tip at leading edge

Figure 3.3.2.2-8. Comparison of a) Expt., reproduced from Strazisar et al. [26], b) Predicted Mrel for 1.6:1 transonic fan rotor, 10% span from tip, operating point near peak efficiency.

Figure 3.3.2.2-9. Computational grid used for reduced flow-path model. Note: all solid surfaces set as slip surfaces.

Figure 3.3.2.2-13. Computational domain used for viscous Rampressor rotor simulations.

Figure 3.3.2.2-14. Computational grid used for Rampressor rotor simulations.

Figure 3.3.2.2-18. View of Rampressor leading edge region at 10% span from tip

Loss Mechanisms

There are two dominant loss mechanisms in the Rampressor rotor. The first is a significant zone of flow separation that exists in the rotating diffuser section of the rotor passage. This flow separation exists due to: 1) a throat Mach number that is higher than ideal, 2) non-ideal geometrical design of the diffuser section of the rotor passage, 3) flow blockage that exists at the throat. Correction of such losses would involve the design of ramp and diffuser sections that optimized the contraction ratio and diffuser loading distribution respectively, along with the

implementation of specific flow control to augment passage boundary layers. The second major loss mechanism is the tip clearance flow. Having such a high pressure ratio across a single axial rotor makes such tip clearance inevitable. Reduction of such losses requires the implementation of sophisticated tip clearance mitigation techniques.

Final Flowpath Performance Predictions

Maximum Rampressor 2 rotor performance predicted using CFD is given in the table below. These results were calculated using the closest configuration simulated to that constructed in the test rig (designated 'as-built'). A 0.001 in tip clearance was employed in these calculations. It was found that adding four more bleed slots to the strake in the region ranging from half way up the compression ramp, through the throat, and entering the downstream rotating diffuser improved both the rotor maximum performance, as well as its starting characteristics.

Table 3.3.2.2-1 Comparison of aero performance between two rotor configurations at 0.001" tip clearance

	As-built rotor, 0.001 in Tip Clearance	As-built rotor, 0.001 in Tip Clearance, 4 additional strake slots
Max PR	8.2:1	8.9:1
Max effy	69.9	72.3

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3.3.2.4 3D CFD Starting Analysis

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3.3.3 Rotor Mechanical Design and Analysis

This section of the report covers the Mechanical Design of the Rampressor-2 Rotor. This work was completed during the period mid 2004 to mid 2005. During this period various configurations were investigated in an effort to satisfy the aerodynamic requirements as well as the Mechanical requirements. Discussion herein will be constrained to the final design configuration only.

The following sections cover the Design Requirements and the Methodology used in the design and analysis of the various components that comprise the Rampressor-2 Rotor. A view of the Rotor assembly with key components identified is presented in Figure 3.3.3-1

Figure 3.3.3-1: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

3.3.3.1 Design Requirements

This section of the report covers the Mechanical Design requirements for the Rampressor-2 Rotor. Generally applicable requirements include the following:

Aerodynamic Design Point (ADP)	41,837 RPM <1>
Maximum Rotor Speed	46,021 RPM (110% ADP)
Maximum Operational Speed	44,766 RPM (107% ADP)
Burst Requirement	125% of Maximum Rotor Speed
Cyclic Life Requirement	> 5,000 cycles to Maximum Operational Speed
Total Life Requirement	> 500 hours at Maximum Rotor Speed

<1> Refinement of the Aerodynamic model led to a slight change in the Aerodynamic Design Point, such that the final value became 41,439 rpm. Virtually all rotor mechanical design was complete by this stage, and the small change is considered inconsequential. It should be noted, however, that the segment detailed design took account of the refined final figures.

A 'best guess' at the rig duty cycle is presented in Figure 3.3.3-2 based on the test cycles that were run during the Rampressor-1 testing. Because of the exploratory nature of this type of testing, it is impossible to predict the cycles that will be run. Therefore an upper bound is established and hardware designed to meet that requirement.

Figure 3.3.3-2: Duty Cycle Assumed for Rampressor-2 Rotor Mechanical Design.

3.3.3.1.1 Rotordynamic Requirements

The rotor was required to have acceptable rotordynamic behavior throughout the operating range. Specifically to ensure that the principal modes of the system are at least 10% away from the highest operating speeds. The usual industry criteria is for critical speeds to be 20% away from the operating speeds for a ball and roller bearing system using 100% of the predicted design point stiffness. The 10% criteria is based on Ramgen's philosophy of using 50% ball bearing stiffness and 100% roller bearing stiffness predicted by the bearing design tool because of the uniqueness of Rampressor 2 static structure

3.3.3.1.2 Life Requirements

Life requirements for the Rampressor 2 rotor were as follows:

LCF: A minimum life requirement of 5,000 cycles was required. Although this is substantially higher than the number of duty cycles that the Rampressor-2 rig was estimated to experience – approx 1,000 – the accuracy of fatigue life predictions at such a low number of cycles was not considered to be reliable. LCF assessments were made for the Disc and coverplates. Segment lifing was performed using a fracture mechanics approach.

Fracture Life: Using a Linear Elastic Fracture Mechanics approach, an upper bound on the allowable stresses in the Rampressor-2 segments was established such that the life of the segments from an assumed initial flaw size to critical crack size was > 5,000 duty cycles. Non-destructive inspection of every segment was then performed to ensure that any flaws present were smaller than the assumed initial flaw size.

3.3.3.1.3 Burst Margin Requirements

The traditional method for calculating the burst speed margin of a turbine rotor has been the use of an equation relating the component average tangential stress to the material ultimate strength.

$$BM = (MUF * UTS / AWMHS)^{1/2}$$

Where,

BM = Burst margin

MUF = Material Utilization Factor. Typically used value is 0.90 for the Disc material used.

UTS = Minimum UTS at the component maximum average bulk temp.

ATS = Area Weighted Mean Hoop Stress at design speed.

The design intent is to have a 20 percent burst margin over the operating speed to provide a safe margin on hub burst. The requirement for adequate burst margin stems from the safety hazard involved if an uncontained failure occurs.

3.3.3.1.4 Secondary Flows

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3.3.3.2 Rotor Assembly Design and Analysis

This section discusses the Mechanical Design of the Rampressor-2 Rotor Assembly. The Rampressor-2 rotor comprises the center section known as the disc with shafts friction welded to either side of the disc. Segments (compressor blades) are inserted into individual disc slots and retained axially by coverplates fore and aft. The rotor is then supported on bearings on either shaft. Views of the Rampressor-2 rotor are provided in Figure 3.3.3-3 and Figure 3.3.3-4

Figure 3.3.3-3: Cross Section through Rampressor-2 Rotor.

Figure 3.3.3-4: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

3.3.3.2.1 Disc & Coverplate Design and Analysis

Extensive effort was undertaken during the Rampressor-1 design to select a suitable material for both the monolithic and segmented (Rampressor 1.6s) rotors. Because of the similarity between Rampressor-1 and Rampressor-2 and the fact that long lead forged parts were already in-house as unused spares, it was concluded that Ti 6-2-4-2 was the candidate material of choice for the disc.

Preliminary disc sizing was driven by the burst margin requirements coupled with the knowledge of the approximate rim width and height that was developed from the aerodynamic design of the flowpath. Sizing of the disc posts was driven by the number of segments, though limited by the load carrying capability of the disc.

Final detailed design and analysis of the disc was performed at MTU-AENA. MTU Aero Engines has pooled its activities in the U.S. under the umbrella of MTU AENA (Aero Engines North America), which operates two locations near East Hartford, Connecticut: a manufacturing subsidiary in Newington makes rotating components (disks, rings and shafts) and a design center in Rocky Hill develops components for joint MTU/Pratt & Whitney engine programs.

Reports covering the Mechanical Design and analysis of the Disc and Coverplates are included in the Chapter 3 Appendix. A section through the rotor showing the coverplate detail is provided in Figure 3.3.3-5

Figure 3.3.3-5: This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

3.3.3.2.2 Segment Design and Analysis

Many considerations went into the Rampressor-2 segment design. Along with all of the requirements previously discussed in the aerodynamic and mechanical sections of this document, the design also had to allow for secondary flow capacities, maintain adequate sealing between the multiple components on the flowpath, while at the same time minimizing the part count on the rotor.

In late 2003 a conceptual study was performed by Optimization Analysis Associates that investigated various flowpath segment configurations for the nascent Rampressor-2 using strake aspect ratio of 2.2. This study determined that the configuration most likely to prove fruitful was one that employed a configuration similar to that shown in Figure 3.3.3-6. Some modifications were made to this concept as the Rampressor-2 design progressed but the fundamental concept remained unchanged. Throughout the design of the Rampressor-2 flowpath, the geometric envelope did not vary significantly between preliminary and final designs.

Following the conceptual analysis, further work was performed on the aerodynamic design of the flowpath and the Mechanical Design Requirements became more defined (Operating speed, disc rim width etc). This allowed the Mechanical Design to progress from the conceptual phase to the preliminary design phase. In a similar manner to the conceptual study, The preliminary Mechanical Design of the Rampressor-2 segments commenced with a two dimensional (2D) study that incorporated the initial flowpath model conducted to optimize the baseline rim definition and underlying dovetail features of the segment design. The results from this study were then used as the starting point for the three dimensional (3D) analysis of the segments.

For the 2D study, a scripting process was employed through the finite element analysis (FEA) package ANSYS. These scripts utilized text based inputs to manipulate the segment geometry in an iterative process. Nearly every feature of the 2D segment was optimized by analyzing an array of logical inputs for each different feature. This allowed for rapid selection of candidate segment geometries with the physical boundaries imposed by the aerodynamic requirements. The down-selected geometries were then modeled in simple 3D sub-models in order to more accurately characterize the mechanical behavior of the segment. Figure 3.3.3-7 shows a typical 2D model of the segment end-cap region and Figure 3.3.3-8 shows a typical 3D FEA sub-model. This very simple, easy to manipulate model was used to simulate hundreds of different segment attributes and made processing the results very manageable.

Results from the 2D/3D Sub-model study contributed to the down select of two segment designs for the Rampressor-2 rotor; a 72 segment design and a 105 segment design. Since there are three flowpaths in the aerodynamic definition of the Rampressor-2 rotor, the 72 and 105 segment designs would require 24 and 35

individual segments for their respective segment designs.

Using the results from the 2D study as a baseline, the 3D segment model was created in SolidWorks. The aerodynamic flowpath used for CFD simulations was used as the foundation of the 3D segment model. As shown in

Figure 3.3.3-9, this allowed for perfect representation of the flowpath in the segment model. The 3D model shown in Figure 3.3.3-10 provided segment geometries for conducting structural analyses on every segment designed for the Rampressor-2 rotor. Figure 3.3.3-11 shows a typical flowpath segment, with some of the key features called out.

In an effort to minimize the total part count and thus cost and lead time, the 72 segment design was pursued. Results from multiple structural analyses conducted on the most highly loaded segments indicated the parts would fail under these extreme conditions. However, stresses on the 105 segment design were determined to be more manageable. In addition, the smaller segment width engendered by the 105-count segment design had the added benefit of a lower strake tip deflection. This was an important consideration as differential radial growth across the strake tip contributes directly to strake tip gap which in turn is a prime mover in the starting capability of the Rampressor. For these reasons, the 105-count segment design was chosen for the Rampressor-2 rotor. Detailed structural analyses were performed on each of the 35 unique segments in order to characterize the mechanical behavior. A typical detailed FE mesh is shown in Figure 3.3.3-12. Stress and displacement plots are shown in Figure 3.3.3-13 and Figure 3.3.3-14.

Figure 3.3.3-6: Rotor Configuration selected after proof of concept study

Figure 3.3.3-7: 2D FE Model of Segment End Cap Region

Figure 3.3.3-8: 3D Sub-model of segment strake region employed in preliminary design phase

- Figure 3.3.3-9:** This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.
- Figure 3.3.3-10:** This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.
- Figure 3.3.3-11:** This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.
- Figure 3.3.3-12:** This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.
- Figure 3.3.3-13:** This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.
- Figure 3.3.3-14:** This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

3.3.3.2.3 Intersegment Sealing

As discussed in the sections detailing the segment design and the secondary flow system design, avoiding cross-communication between the various components of the rotor airflow and avoiding leakage of the rotor flow into unwanted areas is crucial to the performance of the Rampressor-2 rotor.

The segment design thus evolved into one that required use of inter-segment seals to maintain the flow separation discussed above. Since this was of such critical importance to the performance of the rotor, an in-house test program to evaluate various sealing concepts and materials was instituted. The following describes the test program.

Abstract

A primary objective of this series of tests was to determine the ability of thin film Kapton or titanium strips or aluminum strips to seal the intersegment gaps in the Ram 2 test rig. The secondary objective was to test various adhesives and RTVs that could be used to bind the strips in place.

The investigation was carried out in three phases. The first was an examination of the adhesion characteristics of various RTVs at elevated temperatures. None of the samples showed adhesive properties that were suitable for use of RTV alone in the Rampressor two test rig.

The second phase was a pressure vessel test of various seal candidates. The primary objective was to understand how well thin film seal strips (Kapton® & thin Ti or aluminum) reduce leakage and maintain pressures. It was found that the corona etched Kapton® met our requirements of 100 hours and 25 thermal cycles.

The third phase of the investigation was a body force pull test of the Kapton® film that had been bonded into 0.010" x 0.033" slots.

The primary objective of this series of tests was to determine how well the seal-adhesive system would withstand the centrifugal forces and temperatures that it would encounter in an operational Ram2 test rig. There were three different adhesives tested –Cytec BR 680-3, P²SI HTA-52P and P²SI HTA -55P. Body force pull or parallel force pull on the Kapton® strip was selected because it most closely replicated the force that our analysis indicated that the strips would encounter. Although the match was not exact it permitted the system to be evaluated. It was found that the Cytec product –BR 680-3 produced the best still retaining sufficient strength after 200 heat cycles.

Test Series 1: RTV Testing

The purpose of this series of tests is to ascertain the adhesion characteristics of various RTVs at elevated temperatures. The products used were Loctite Red, 59630, Loctite copper 5920, PRC-Desoto, PR 1995B and Cotronics 1531.

Steel blanks were prepared that were one inch by .25 inch by .125 inch thick. After cleaning the blanks, a 0.0005" film of RTV was installed between the steel blanks. (See Figure 3.3.3.2.3-1) The article was then allowed to cure for 72 hours before being placed into an oven at 600 degree F. After 90 minutes the articles were removed and

subjected to a shear force measurement.

After removal of the test articles from the oven, the majority failed in handling, showing poor strength under elevated heating conditions. Of the samples tested only one - PRC-Desoto, showed minimal strength of ~0.5 LbF shear strength.

None of the samples achieved acceptable shear strength at operating temperatures.

Test Series 2: Pressure Testing

A direct comparison of the leakage rate with and without the seal in place does provided at least a relative measure of the effectiveness of the seal. Both seal slot and flap configurations were tested to determine differences in effectiveness of the two sealing methods. By applying pressure across the seal at high temperature, the temperature compatibility of the seal was evaluated. In this configuration, the material strength, plasticity & toughness can be evaluated but not measured. This testing did not provide definitive proof that the evaluated seal configurations would work in the Rampressor-2 test article because test conditions did not replicate the full environment. (In Rampressor-2, there will be significant differences between static and total flow conditions on both sides of the seal, as well as a high g-field caused by rotation. Neither of these effects were simulated.) The testing did provide relative data on seal effectiveness and temperature compatibility at realistic temperatures.

A pressure vessel was constructed using two 2-inch 150lb. flanges. (See Figure 3.3.3.2.3-2) The topmost flange was fitted with a .25-inch compression fitting for tubing connection as well as a compression fitting for a free stream air thermocouple. The bottom flange had a .010 slot machine into the base (See Figure 3.3.3.2.3-3). This was connected to 20 feet of coiled .25 inch stainless steel tubing and a flow meter. (See Figure 3.3.3.2.3-4).

The samples of Kapton and Ti strips were placed into the base flange (Figure 3.3.3.2.3-5) and held in place by bolted clips. The test strips used were .874 long by .250-wide in different thicknesses depending upon the material tested. The material used was Kapton® in 0.003", 0.004" and 0.005" thickness, Titanium strips in 0.004" thickness and Aluminum strips of 0.005" thickness. The two clips were configured so that one clamped the test piece while the second clip had a machined relief to provide both a pressure path as well as an insertion stop for the test strips to ensure consistent installation. The two flanges were then bolted together and connected to the .25 stainless supply line. The unit was then placed into the oven and the temperature was brought to 650°F and held while pressure simulations were run. Pressure was then randomly applied while the flows were recorded.

As can be seen in Figure 3.3.3.2.3-6, the 0.005" thick Kapton polyimide film produced the least amount of leakage of the samples tested.

Test Series 3: Pull tests

The purpose of this series of tests was to determine if the combination of the seals and the adhesive would withstand the possible G loads in the Rampressor 2 test rig. As a first step an ANSYS Finite Element analysis was completed that showed that the minimum was determined to be 3lbf.

Six Titanium bars were prepared by milling 18 slots in one face to simulate Rampressor 2 slots. (Figure 3.3.3.2.3-7) Three of the bars were cleaned with Turco 5578 and stored in nitrogen packs. Three of the bars were heated to a temperature of 700 deg.F for a period of three hours in order to simulate the temperature required to remove polyimide adhesives. The adhesives used were Cytec BR 680-3, PSI 55 and 52.

Each of the test adhesives were used on one bar that had been heated and one bar that was cleaned. Corona etched Kapton strips were used for all tests. The bars, with the strips installed were then brought to 300 deg. F for 3 hours to cure the adhesives. After the cure process was completed clips were attached to the exposed strips to permit the full body pull tests.

The test blocks were placed in the oven and brought to 650 deg.F for four hours and were then removed and allowed to come to 90 deg.F Every other cycle a digital meter was used to apply force to one of the strips in each block. The cycling and tests were continued until 200 hours was reached. The results from the pull-tests are presented in Figure 3.3.3.2.3-8

After 200 hours the corona etched Kapton with the Cytec BR 680-3 adhesive retained enough adhesion that use in the Rampressor-2 rig was judged acceptable.

Figure 3.3.3.2.3-1: Schematic of RTV Test Article

Figure 3.3.3.2.3-2: Pressure Test Vessel – Upper Flange

Figure 3.3.3.2.3-3: Pressure Test Vessel – Lower Flange

Figure 3.3.3.2.3-4: Pressure Test Setup

Figure 3.3.3.2.3-5: Detailed view of Test Strip Test Article

Figure 3.3.3.2.3-6: Results from Seal Strip Pressure Tests

Figure 3.3.3.2.3-7: Pull-Test Article

Figure 3.3.3.2.3-8: Segment Seal Bond Strength from Seal Strip Pull-Tests

3.3.3.3 Heat Transfer

The thermal analysis on the rotating components is done in order to help predict

- accurate thermal growth of the rotating components, which ultimately aids in determining the tip clearances, and creation of cold condition drawings.

- accurate material stresses which helps determine the life of the components.

The rotating components for which the thermal analyses have been performed are the segment, disk, coverplate and the snap ring. The procedure for performing the thermal analysis is as follows

- all analyses have been performed at full load steady state condition (100% speed).
- the meanline aero conditions and CFD, where appropriate, were used for generating the appropriate boundary conditions
- the convective heat transfer coefficients were generated using flat plate correlations, along with disk pumping for the radial surfaces and for the lab seals and hub seals, appropriate seal correlations were used.
- Since the flow is supersonic in most regions, where such flow occurs, suitable recovery factor were used to calculate the near wall fluid temperatures. Figure 3.3.4.3-1 defines the recovery factor used and shows the temperatures generated due to the recovery factors.

The segment thermal model along with the types of correlation used is shown in figure 3.3.4.3-2. Some typical boundary conditions calculated using the appropriate correlations for the segments is shown in figure 3.3.4.3-3. The temperature distribution for the different segments is shown in figure 3.3.4.3-4. All the pre-inlet segments have similar temperature distribution whereas each of the compression and diffuser ramp segments have unique temperature distribution due to the uniqueness of the applied boundary conditions.

The conclusions from the segment thermal analyses are

- Based on the thermal analysis prediction for 21 different segments, the pre-inlet segments (segments 19 through 35) have similar radial, axial and circumferential temperature distribution
- Each of the compression ramp and diffuser segments have unique temperature distribution (segments 1 through 18)
- The temperature predictions are detailed enough to allow for structural deflection as well as stress predictions.

The thermal analysis of the disk, cover plate and the snap ring were done together and the thermal model along with the appropriate boundary conditions applied at 100% speed is shown in figure 3.3.4.3-5. The requirements for performing the thermal analysis on these components are to ensure the following

- The disk is cooled appropriately so that material stresses are at a manageable level
- The outlet temperature of the disk cooling/purge flow is below 150 F. This is done in order to ensure that the static structure is not heated up by the purge flow coming out of the wheel space cavities
- The temperature predictions are done in enough details to help perform the structural analysis to help predict the radial growth and disk burst margins.

The analysis procedure adopted was

- use a 2D axisymmetric model for the three components
- Assume that a good heat conduction path exists from the disk to the cover plate and the snap ring
- Heat input from the segment is imposed as a boundary condition rather than including the segment in this analysis
- The disk cooling is modeled by using fluid elements in Ansys rather than as a convective boundary conditions because this allows for a better prediction of the heat pick up.

For the secondary flow temperatures, aero's radial equilibrium estimates (see figure 3.3.4.3-6) were used rather than the CFD because at the time the analysis was performed, the CFD results were not available. The predicted design point (100% speed) temperatures for the above mentioned components are shown in figure 3.3.4.3-7. The results indicate that the temperature gradient results in providing benefits such as structurally better burst margins and lower thermal growth.

The conclusions from this analysis are

- The disk temperature are below 500F and the thermal gradients are reasonable which should allow for manageable material stress levels
- The predicted outlet temperature of the disk cooling/purge flow is around 125 F, which should allow for the static structure to be cooled by the purge flow rather than the purge flow heating up the static structure
- The temperature predictions are done in enough details to help perform the structural analysis to help predict the radial growth and disk burst margins.

-To take into account the irreversibilities in the boundary-layer flow system in high speed ($M > 1$) flow, a recovery factor is defined by:

$$-r = (T_{aw} - T_{\infty}) / (T_{te} - T_{\infty})$$

- T_{aw} = Adiabatic Wall Temperature

- T_{∞} = Free Stream Static Temperature

- T_{te} = Free Stream Total Temperature

Figure 3.3.4.3-1 Effect of Recovery factor in supersonic flow for the Rampressor flow path

Figure 3.3.4.3-2 This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992

Figure 3.3.4.3-3 This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992

Figure 3.3.4.3-4 This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992

Note:

- Used Plane 55 Axisymmetric elements for capturing the conduction effects
- Used Fluid 116 and Surf 151s for capturing the heat pickup in the disk cooling/purge flows

Figure 3.3.4.3-5 Applied boundary conditions for rotating components other than segments

Figure 3.3.4.3-6 This Figure intentionally left blank under the provisions of The Energy Policy Act of 1992.

Figure 3.3.4.3-7 Predicted rotating component temperature distribution at 100% design speed at full load steady condition

3.3.3.4 Bearings

The characteristics of the Rampressor-2 bearings were a fundamental input to the Rotordynamics analysis and thus a considerable number of iterations were required before a suitable bearing design could be settled upon.

Requirements imposed on the bearing design included the following:

- Tolerate a steady state thrust load of ~2500 lbf with the ability to operate continuously at ~1500 lbf
- Operate at ~46,200 rpm (110% design speed)

Bearing stiffness commensurate with the rotordynamics requirement that the system critical speeds are at least 10% away from the operating speeds. This is for the case when the predicted ball bearing design stiffness is reduced by 50% and the roller bearing stiffness is at 100% as previously discussed in the requirements section

- Maximum radial load capability of 150 lbf at 46,200 rpm
- A minimum life of 100 hours at maximum load conditions
- Ability to maintain 0.001 inch clearance at the stroke tips
- Bearing system design takes up all the axial thrust with minimal adjustment of the wheel space cavity pressures

The bearing design was carried out by Alouette Technologies of San Diego, CA and the detailed design and analysis report is included in the Chapter 3 Appendix. Some highlights from the report include the following:

- Predicted bearing life is greater than 100 hours
- Bearings require about 50% more oil flow than that in Rampressor 1 rig due to additional heat generated at higher speeds
- Bearing Configuration – Fwd Side
- 208 Size Split Inner Ring hybrid ball bearing 40mm x 80 mm x 15mm
- Contact Angle of 16°
- Bearing Balls are made of Silicon Nitride material
- Bearing races are made of Cronidur 30
- Bearing cages are thermoplastic cages
- At a thrust load of 2500 lbf, bearing stiffness is 2.75E6 lbf/in
- Bearing Configuration - Aft Side
- 108 Size Hybrid Roller bearing 40mm x 68mm x 15mm
- At a radial load of 100 lbf, the bearing stiffness is 3E6 lbf/in

3.3.3.5 Rotordynamics

As discussed above in the bearing design section, the rotordynamics analysis was closely allied to the bearing design due to the sensitivity of the rotordynamics to the bearing stiffness values. The rotordynamics analysis of the Rampressor-2 was performed by Barbour Stockwell Incorporated of Woburn, MA and the analysis report is included in the Chapter 3 Appendix

Figure 3.3.3-9 and Figure 3.3.3-10 show the primary components of the rotordynamics model. Some highlights from the rotordynamics analysis include:

- The critical speeds are 10.1% away from the 110% operating speed, (45,583 rpm) for the 50% bearing stiffness reduction case
- Radial loads in all the cases above are at or below 200 lbf for 110% speed case for both bearings
- The bearing stiffness are directly proportional to the axial thrust load. The bearing stiffness values have been designed for axial thrust loads between 1300 lbf and 2500 lbf.

Figure 3.3.3-9: Primary Components of Rotordynamics model.

Figure 3.3.3-10: DyRoBeS Analytical model of Rampressor-2.

3.4 Testing

Testing of the Rampressor-2 rig began in mid-December, 2005. Initial tests focused on characterization of the pre-swirl nozzle exit conditions to ensure conformance with predictions. After these tests were successfully concluded, spin tests of the segmented rotor were begun. Mechanical failure of the flex coupler on February 15,

2006 brought spin testing to a halt while the root cause was pursued and corrected. Budget Period 4 ended shortly after the flex coupling failure so a subsequent report will contain a detailed description of the root cause investigation and findings. Testing has resumed as of the writing of this report and initial results indicate the rotordynamic cause of the flex coupling failure has been found and corrected.

Section 3.4.1 describes the pre-swirl nozzle characterization tests while Section 3.4.2 shows a post-test report from the flex coupling failure and discusses some of the root cause findings and fixes. Section 3.4.3 describes the rotordynamic analyses and fixes implemented after the conclusion of BP4.

3.4.1 PSN Characterization

The first test conducted of the Rampressor-2 rig was a survey of the PSN exit flow. This test was conducted for two reasons. The first reason was to measure the performance of the PSN. When testing the rotor it was only possible to measure the combined performance of the PSN and the rotor. By measuring the performance of the PSN it was possible to separate the PSN performance from the combined PSN and rotor performance, and, hence, determine the performance of the rotor. The second reason was to compare the actual PSN flow measurements with the predicted flow. This allowed the flow prediction tools to be tuned to provide better predictions for future designs.

Since it was not possible to install instrumentation between the PSN and the rotor, the rotor was removed for the PSN testing. A hub ring and a tip ring designed for the PSN test extended the flow path downstream of the PSN and accommodated the required instrumentation. Figure 3.4.1-1 is a sketch of the Rampressor-2 flow path which shows the close proximity of the PSN and the rotor. Figure 3.4.1-2 shows the flow path used during the PSN characterization tests in which the rotor was removed. In Figure 3.4.1-3 a sketch of the PSN vanes is shown. Figure 3.4.1-4 shows a photograph of the Rampressor-2 configuration used during the PSN characterization tests.

The measurements for the PSN characterization tests consisted of the air mass flow rate, the pressure and temperature in the inlet plenum, a wide range of static pressure measurements in the vicinity of the PSN, and total pressure measurements using the cobra probe. Eight static pressure measurements were made midway between vanes at the leading edge of the vanes: four on the hub and four on the shroud. These eight measurements were also recorded during rotor testing. Four static pressure measurements were made on the PSN hub midway between vanes at the trailing edge of the vanes. Sixteen static pressure measurements were made on the PSN tip ring at the trailing edge of the vanes. These measurements were divided into four groups of four. Each group of four had the same orientation relative to the vanes for the purpose of determining symmetry of the PSN exit flow. An additional four static pressure measurements were made on the tip ring slightly downstream of the group of sixteen.

The cobra probe could be mounted at two locations on the tip ring to determine PSN exit flow symmetry. The position closest to top dead center is called position 0 while the other position is called position 1. The two locations can be seen in Figure 3.4.1-4. In addition, the tip ring could be rotated between four different positions, referred to as positions C, D, E, and F, to move the rake to different locations relative to the trailing edge of the two PSN vanes. The cobra probe could be in eight different positions within the flow field. These positions were labeled by combining the tip ring position with the cobra probe position (e.g. C1, D0, etc.).

The tip ring position indicator is shown in Figure 3.4.1-5 and Figure 3.4.1-6. Figure 3.4.1-7 shows the locations of four of the static pressure ports relative to the PSN vanes for tip ring positions C, D, E, and F. The reference point on the vane is the downstream location where the vane stops touching the outer wall. Figure 3.4.1-8 shows the locations of the cobra probe relative to the PSN vanes for the four tip ring positions when the cobra probe is mounted in position 0. This figure uses the same PSN vane reference location. Table 3.4.1.2-1 shows the location of the cobra probe inlet for each of the eight measurement positions. The location assumes that the cobra probe is halfway between the flow path walls and is rotated 66° to align with the flow. Changing the radius or the angle of the cobra probe will result in a slight change in the measurement location.

The PSN characterization test plan required operating the PSN at specific Mach number conditions, so the first task was to determine what flow rates corresponded to those conditions. This was accomplished by increasing the mass flow rate through the PSN until the corrected mass flow rate stopped increasing. Once the flow choked, the corrected mass flow rate remains constant although the actual flow rate increases. The corrected mass flow rate, $\dot{m}_{corr}$, is defined as

$$\dot{m}_{corr} = \dot{m} \sqrt{\frac{T_{plenum}}{518.4} \frac{14.696}{P_{plenum}}}$$

where $\dot{m}$ is the measured mass flow rate, T_{plenum} is the measured temperature, in degrees Rankine, in the inlet plenum, and P_{plenum} is the measured pressure, in psia, in the inlet plenum. Once the choked corrected mass flow rate, $\dot{m}_{corrM=1}$, was measured the corrected mass flow rate at other Mach numbers was calculated according to the following equation.

$$\frac{\dot{m}_{corr}}{\dot{m}_{corrM=1}} = \frac{M \left(1 + \frac{\gamma-1}{2} M^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}}{\left(1 + \frac{\gamma-1}{2} M^2 \right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}}$$

The results from this test are shown in Table 3.4.1.2-2.

Table 3.4.1.2-2 To characterize the PSN the corrected mass flow rate was first set to that which corresponded to the desired Mach number. The cobra probe was moved to the desired radial location and its angle was remotely adjusted until the difference between the left and right channels was zero. A complete set of data was then taken using the Boeing data acquisition system. This data included the angle and total pressure from the cobra probe. The cobra probe was moved to the next radial location and the process was repeated until data had been gathered across the flow path height. A list of the nominal radial locations at which measurements were taken is shown in

Table 3.4.1.2-3.

Table 3.4.1.2-4 shows all of the PSN surveys conducted by PSN Mach number, tip ring configuration, and cobra probe position.

3.4.1.1 Results

Figure 3.4.1-9 shows the total pressure measurements from the center channel of the cobra probe when located at position C0 at different radii over a range of corrected mass flow rates. As one would expect, the total pressure increases as the corrected mass flow rate increases. For the lower corrected mass flow rates, the total pressure is nearly constant but at the higher corrected mass flow rates the total pressure is lower near the inner wall (lower radius). Figure 3.4.1-10 shows the total pressure measurements from the center channel of the cobra probe at different cobra probe locations with a corrected mass flow rate of 2.62 lbm/s. The measurements show variation in the total pressure with measurement position at any given radial position.

Using the total pressure measurements and static pressure measurements, the local Mach number of the flow can be calculated. Figure 3.4.1-11 shows the local Mach number when the cobra probe is located at position C0 at different radii for a range of corrected mass flow rates. As one would expect, each local Mach number radial profile is similar to the corresponding total pressure radial profile seen in Figure 3.4.1-9. Figure 3.4.1-12 shows the local Mach number with measurement position at different cobra probe locations with a corrected mass flow rate of 2.62 lbm/s. The measurements show variation in the local Mach number at any given radial position as expected based upon the wide variation seen in the total pressure measurements shown in Figure 3.4.1-10.

Figure 3.4.1-13 shows the continuity corrected flow angle measurements from the center channel of the cobra probe when located at position C0 at different radii for a range of corrected mass flow rates. It is seen that the flow angle is typically about 69° in the center of the flow path and lower near the two walls presumably due to boundary layer effects. The flow angle profile varies only a few degrees with corrected mass flow rate. Figure 3.4.1-14 shows the flow angle measurements from the center channel of the cobra probe at different cobra probe locations with the corrected mass flow rate fixed at 2.62 lbm/s. The measurements show a decrease in the flow angle near the walls relative to the flow angle in the center of the flow path. The flow angle varies with cobra probe position and is seen to change by about 5° in the center of the flow path and by up to almost 20° near the walls.

Figure 3.4.1-15 shows several mass averaged flow properties at the different measurement conditions for a

corrected mass flow rate of approximately 2.62 lbm/s. The mass averaged Mach number, $M_{\dot{m}avg}$, is defined by

$$M_{\dot{m}avg} = \frac{\sum \dot{m}_{local} M_{local}}{\sum \dot{m}_{local}}$$

where $\dot{m}_{local}$ is the local mass flow rate and M_{local} is the local Mach number. The mass averaged flow angle is defined in a similar fashion. The mass averaged total pressure ratio, π , is defined by

$$\pi = \frac{P_{t,\dot{m}avg,out}}{P_{t,\dot{m}avg,in}} = \frac{\frac{\sum_{out} \dot{m}_{local} P_{t,local}}{\sum_{out} \dot{m}_{local}}}{\frac{\sum_{in} \dot{m}_{local} P_{t,local}}{\sum_{in} \dot{m}_{local}}} = \frac{\sum_{out} \dot{m}_{local} P_{t,local}}{\sum_{in} \dot{m}_{local} P_{t,local}}$$

where $P_{t,\dot{m}avg,out}$ is the mass averaged out flow total pressure, $P_{t,\dot{m}avg,in}$ is the mass averaged in flow total pressure, $P_{t,local}$ is the local total pressure, $\sum_{out}$ is a summation over the out flow from the PSN, and $\sum_{in}$ is a summation over the in flow to the PSN. For these tests, the inlet pressure was used as the mass averaged in flow total pressure. Since mass is conserved across the PSN

$$\sum_{out} \dot{m}_{local} = \sum_{in} \dot{m}_{local}.$$

The mass averaged total pressure loss coefficient, ϖ , is defined by

$$\varpi = \frac{P_{t,\dot{m}avg,in} - P_{t,\dot{m}avg,out}}{P_{t,\dot{m}avg,out} - P_{\dot{m}avg,out}}$$

where $P_{\dot{m}avg,out}$ is the mass averaged out flow static pressure.

As mentioned earlier, the purpose of these tests was to characterize the performance of the PSN so that the rotor performance could be separated from the combined performance of the PSN and rotor. In order to do this it is necessary to know the mass averaged total pressure ratio across the PSN and the PSN downstream Mach number as functions of corrected mass flow rate. Using the data curve fits for these two quantities were generated. The PSN mass averaged total pressure ratio data and curve fit are shown in Figure 3.4.1-15 while the mass averaged Mach number data and curve fit are shown in Figure 3.4.1-16. Since the PSN is expected to operate with a Mach number in the range of 0.6 to 0.8, the curve fit was biased to provide a good fit to the data in that range. No bias was applied to the total pressure ratio data since the error with the curve fit is small. The mass averaged flow angle was also calculated for each measurement with the results shown in Figure 3.4.1-17. It can be seen that the mass averaged flow angle at position C0 varies by less than one degree over the corrected mass flow range. The mass averaged flow angle varies more between different measurement positions but the total variation is just over 4 degrees. To complete the analysis, the mass averaged total pressure loss coefficient, ϖ , was calculated for each measurement with the results being shown in Figure 3.4.1-18. The total pressure loss coefficient is expected to be nearly constant with corrected mass flow rate and position, but, instead, is seen to vary significantly with both corrected mass flow rate and position.

3.4.1.2 Conclusions

The PSN was designed to provide a uniform Mach number and a uniform flow angle of approximately 66°. In general, the performance of the PSN meets the design goal. The variation of the mass averaged Mach number between different measurement positions was typically found to be less than 1% for the same nominal corrected mass flow rate. The measured mass averaged continuity corrected flow angle was found to be between 63° and 68°

depending upon the measurement position. Although the flow looks good when comparing the mass averaged values, the flow was not as uniform as expected. Due to the limited number of measurements, it is not known if the variation is a characteristic of the PSN flow field or an isolated nonuniformity. The variations are concerning and additional measurements should be taken to better characterize the PSN flow if time permits.

Table 3.4.1.2-1: Nominal location of cobra probe for each measurement position. This measurement assumes the cobra probe is at an angle of 66°, the nominal design flow angle, and at midheight in the flow path. The angular location will change slightly with radius.

Measurement Position	Measurement Distance Downstream from PSN Vane Reference Point	Measurement Angular Location From TDC Looking Upstream
C0	0.855 inches	31.41°
D0	0.855 inches	34.22°
E0	0.855 inches	37.03°
F0	0.855 inches	39.84°
C1	0.855 inches	59.53°
D1	0.855 inches	62.34°
E1	0.855 inches	65.16°
F1	0.855 inches	67.97°

Table 3.4.1.2-2: Table showing the relationship between the Mach number and the corrected mass flow rate for the Rampressor-2 PSN. The corrected mass flow rate at a Mach number of 1.0 was measured experimentally and used to calculate the other corrected mass flow rates.

PSN Mach Number	Corrected Mass Flow Rate [lbm/s]	% Choked Flow
0.0	0.000	0.0
0.1	0.466	17.3
0.2	0.916	33.5
0.3	1.334	49.2
0.4	1.707	62.9
0.5	2.026	74.4
0.6	2.285	84.0
0.7	2.481	91.4
0.8	2.615	96.2
0.9	2.691	99.0
1.0	2.715	100.0

Table 3.4.1.2-3: Nominal cobra probe radial locations used during the PSN survey.

Sample	Radius (in)	Comment
NA	4.782	Inner flow path radius
1	4.804	
2	4.835	
3	4.867	
4	4.898	
5	4.929	
6	4.960	
7	4.992	
8	5.023	
9	5.054	
10	5.085	
11	5.117	
12	5.148	
13	5.179	
14	5.210	
15	5.242	
16	5.273	
17	5.304	
18	5.335	
19	5.367	
20	5.398	
21	5.429	
22	5.461	
23	5.492	
24	5.523	
NA	5.452	Outer flow path radius

Table 3.4.1.2-4: Measurement configurations and flow conditions for PSN surveys.

Cobra Probe Location	0	0	0	0	1	1	1	1
Tip Ring Index	C	D	E	F	C	D	E	F
PSN Mach Number								
0.3	X							
0.4	X							
0.5	X							
0.6	X	X	X			X		
0.7	X	X	X		X	X	X	
0.8	X	X	X		X	X	X	
0.9	X	X	X		X	X		
1.0	X							

Table 3.4.1.2-5: PSN flow properties for a nominal corrected mass flow rate of 2.62 lbm/s

Measurement Position	Average Corrected Mass Flow Rate (lbm/s)	Mass Averaged Mach Number	Mass Averaged Flow Angle	Mass Averaged Total Pressure Ratio (π)	Mass Averaged Total Pressure Loss Coefficient (ϖ)
C0	2.616	0.8078	67.50°	0.9590	0.1223
D0	2.619	0.8054	63.63°	0.9452	0.1663
E0	2.619	0.8065	66.82°	0.9514	0.1464
C1	2.616	0.8034	66.45°	0.9551	0.1357
D1	2.620	0.8079	65.59°	0.9447	0.1674
E1	2.622	0.8295	67.63°	0.9603	0.1134
Average	2.619	0.8101	66.27°	0.9526	0.1419

Figure 3.4.1-1: Sketch of the flow path with the rotor installed. Flow is from left to right. A PSN vane can be seen on the left side of the image and just downstream of that are the two holes for cobra probe insertion during the PSN characterization tests. The rotor is seen downstream of the two holes.

Figure 3.4.1-2: Sketch of the flow path used during the PSN characterization tests. Flow is from left to right. A PSN vane can be seen on the left side of the image and just downstream of that are the two holes for cobra probe insertion.

Figure 3.4.1-3: Sketch of the PSN vanes. The PSN had a total of 36 vanes with equal angular spacing. At the outer radius of the PSN the spacing between vanes, as shown by the line, was 0.848 inches.

Figure 3.4.1-4: Photograph of Rampressor-2 configured for PSN testing. The PSN tip ring is the outer silver ring on the left while the PSN hub is the inner silver ring. The bellmouth is the dark gray object on the right. The cobra probe mounting hardware is mounted in position 1. Position 0 can also be seen.

Figure 3.4.1-5: Sketches of tip ring position indicator. In the left sketch the tip ring, the part with the four dowel pin slots, is in position C. Rotating the tip ring clockwise moves it to positions D, E, and F, respectively. The right sketch shows the angular spacing between the dowel pin slots.

Figure 3.4.1-6: Photograph of Rampressor-2 configured for PSN testing showing the indicator for positions C, D, E, and F. The indicator is in the center of the photograph and shows the tip ring to be in position D.

Figure 3.4.1-7: Sketches of the static port locations for the four different tip ring positions. The top left sketch is for tip ring position C, the top right is for position D, the lower left is for position E, and the lower right is for position F. The distance between the vane reference point and the leftmost static port is 0.284 inches for tip ring position C, as shown by the line, 0.024 inches for position D, 0.248 inches for position E, and 0.513 inches for position F.

Figure 3.4.1-8: Sketches of the cobra probe locations relative to the PSN vanes for the four different tip ring positions with the cobra probe mounted in position 0. The top left sketch is for tip ring position C, the top right is for position D, the lower left is for position E, and the lower right is for position F. The distance between the vane reference point and the centerline of the cobra probe mounting port is 0.131 inches for tip ring position C, as shown by the line, 0.135 inches for position D, 0.400 inches for position E, and 0.664 inches for position F.

Figure 3.4.1-9: Measured total pressure radial profiles at position C0 for a range of corrected mass flow rates.

Figure 3.4.1-10: Measured total pressure radial profiles at different cobra probe positions for a corrected mass flow rate of 2.62 lbm/s.

Figure 3.4.1-11: Local Mach number radial profiles at position C0 for a range of corrected mass flow rates.

Figure 3.4.1-12: Local Mach number radial profiles at different cobra probe positions for a corrected mass flow rate of 2.62 lbm/s.

Figure 3.4.1-13: Continuity corrected flow angle profiles at position C0 for a range corrected mass flow rates.

Figure 3.4.1-14: Continuity corrected flow angle radial profiles at different cobra probe positions for a corrected mass flow rate of 2.62 lbm/s.

Figure 3.4.1-15: Calculated mass averaged total pressure ratio for each measurement and curve fit to be used during rotor testing.

Figure 3.4.1-16: Calculated mass averaged Mach number for each measurement and curve fit to be used during rotor testing.

Figure 3.4.1-17: Calculated mass averaged flow angle for each measurement.

Figure 3.4.1-18: Calculated mass averaged $\bar{\omega}$ for each measurement.

3.4.2 Flex Coupling Failure Report

3.4.2.1 Test Conditions and Description

- First drive rotor test.
- Windmill under low plenum pressure.
- Set drive speeds and explore control at speeds below 20K rpm.

Rig bearing temps, vibration and drive control were demonstrated at speeds to 12K rpm. Ramp from 12-20k

rpm was initiated. High vibrations encountered at 18.7-19.8K rpm. Vibration along the axis of rotation showed the greatest amplitude. Slow down was requested. Video indicated hardware failure at the coupling. Emergency shutdown was conducted. Shutdown terminates rig and drive air supplies.

Figure 3.4.2-1: Speed and vibration for the entire run. Vibration is measured with Wilcoxon PC420VP-10 transducers. Output is 0-1 ips FS.

Figure 3.4.2-2: Vibration baseline at 12,000 rpm

Figure 3.4.2-3: Vibration trace for one minute surrounding the event and shutdown.

3.4.2.2 Post-Test Inspection

Physical and video evidence indicates failure of the drive end, element pack in the John Crane TSCW-0033-W533-H129 coupling.

Figure 3.4.2-4: Failed flexible coupling and bent turbine shaft

Observations associated with the failure teardown:

- Drive-side, flex coupling, shim pack was partially torn. Shim pack was not completely separated. All fasteners were in place at the drive, driven and center spool split line flanges. One fastener head (of 6) on the drive hub was sheared. All fasteners associated with the two flexible element packs were present. All gag bolts were removed prior to installation (See Figure 3.4.2-4 and Figure 3.4.2-5).
- Flex coupling cover was thrown from its normal position. Two bolts (5/16 UNC) were sheared from the non-instrumented half of the cover. Two bolts were sheared and one stripped on the instrumented half of the cover. Installed instruments were destroyed (two drive hub, 1/rev indicators and one proximity probe). 1/rev sensors are installed in close proximity to the drive hub flange. The proximity probe is installed on the driven side of the center spool, split line, flange. Sensors have approx. 0.080 in. running clearance on coupling fasteners. The proximity probe provided output until 14971 seconds test time.
- Two extended take-offs for the aft bleed manifold experienced deformation as a result of contact. All 18 retaining fasteners (1/4 UNC) were sheared at the aft frame interface. Aft bleed manifold gasket showed signs of wear (scrubbing). Four vacuum hoses were disconnected from the take-offs. Fasteners and washers were collected from the ground below the rig. Manifold was constrained by cooling plumbing and ID, wheelspace supply air, fittings. (See Figure 3.4.2-6 and Figure 3.4.2-7)
- Drive shaft bent relative to the drive face. Brass shaft cover heavily damaged. (See Figure 3.4.2-16)
- Drive shaft was free spinning, without axial play, at time of disassembly.
- Drive hub remained intact and attached. Residual bolt torque was present. Hub was removed using standard thermal methods and the jack screw. Keys appeared in good condition. Drive hub was warped.
- Metallic, non-magnetic shavings were found in the drive turbine exhaust. Quantity was small. Manufacturer suspects aluminum seals.
- Used, drive mist lubrication system, oil was collected. Oil appears brown (scorched) and contains small

metallic, magnetic shavings.

- Rig side, flex coupling, center spool; driven hub and Ramgen rotor are free spinning. Obvious bend in flex coupling. (See Figure 3.4.2-8)
- Driven, flex coupling hub; flex elements; and element bolts were intact at the beginning of the disassembly. Destructive disassembly was required for driven hub removal. Flex element bolts were drilled out to preserve the flexible element. Hub was sectioned at one keyway to allow removal. Damage to the hub center spool was produced during screw head grinding. Damage to the driven hub and the aft oil manifold resulted from hub grinding. (See Figure 3.4.2-5)
- The driven hub retention bolt had minimal preload, but was fully engaged. The taper fit had a reduction in internal clearance. Nominal clearance is 0.030 (distance between shaft end face and the threaded, bottom plate of the taper fit). Teardown clearance was 0.012 in.
- Flex coupling center spool shows appreciable deformation near frame interface. (Figure 3.4.2-8)
- Frame separation required slightly higher forces. Minor extraction tool binding encountered during disassembly.
- Rotor does not show visible signs of damage. Access prevented runout readings in the rig.
- Bearings do not show race galling. Ceramic balls and rollers are free of chipping. Cages are intact.
- The tip ring abrasible was not contacted. (See Figure 3.4.2-9)
- Aft shaft labyrinth seal shows wear. Grooves cut in brass seals ring. (See Figure 3.4.2-12)
- Aft coverplate and segment labyrinth seals have cut into the abrasible. Coverplate seals have cut in the deepest. Segment seal is barely contacting (approx. 0.010 in. overlap). (See Figure 3.4.2-14)
- Aft hub wheelspace labyrinth seal is preferentially worn into the brass. South side (6:00 to 11:00 ALF) shows grooving in the brass. (See Figure 3.4.2-10)
- Forward wheelspace labyrinth and coverplate seals show grooving from contact. Forward segment labyrinth seal has not contacted. (See Figure 3.4.2-13)
- Forward hub wheelspace labyrinth seals show grooving and indication of sliding. (See Figure 3.4.2-11)
- Aft bearing ID race locknut was hand tight after unpeening anti-rotation tabs. Forward bearing locknut had minimal preload after unpeening anti-rotation tabs.
- A small quantity of black dust was wiped from the thru-race ID oil passages during disassembly.

Figure 3.4.2-5: Driven flex coupling hub prior to destructive removal

Figure 3.4.2-6: Sheared aft bleed manifold bolts (only ID bolt holes shown)

Figure 3.4.2-7: Damaged aft bleed manifold teak-off

Figure 3.4.2-8: Flex coupling, center spool damage

Figure 3.4.2-9: Non-contacted tip ring abradable

Figure 3.4.2-10: Aft, hub, wheelspace, labyrinth seal (BDC oil scoop shown on right side of photo)

Figure 3.4.2-11: Fwd, hub, wheelspace, labyrinth seal (BDC oil scoop shown on right side of photo)

Figure 3.4.2-12: Aft shaft seal grooving

Figure 3.4.2-13: Forward coverplate and segment labyrinth seal grooving

Figure 3.4.2-14: Aft coverplate and segment labyrinth seal grooving

Figure 3.4.2-15: Failed drive end flexible element stack

Figure 3.4.2-16: Bent drive shaft with brass seal wear

3.4.2.3 Event Data

Rig health data is collected using Ramgen's PLC at 1 Hz rate. The following figures and comments are focused on a one minute period surrounding the failure event. Charts without strong time dependencies are shown with expanded timelines. "Times of interest" are denoted on several charts. While each indicated time appears as two converging lines, each of the three pairs represents a single focus area. The first "Time" represents the onset of excessive vibration, as indicated by the velocimeter output. The second "Time" denotes catastrophic coupling failure, as indicated by the loss of the drive speed indicator and the change in the drive vibration signature. The third "Time" indicates facility shutdown, as indicated by the supply plenum pressure decay to ambient.

Figure 3.4.2-17: Proximity probe data

Proximity probe data should be viewed with skepticism. Probes are factory calibrated but have not been verified using known gaps in their final installation. Trends are correct, but global values are not confirmed.

The RTRAFTPROXZ targets the forward end of the disc shaft. Large forward thrust is indicated at the second "Time of interest", consistent with heavy deflection of the forward load cell seen in RTRFWDTHRUST on Figure 3.4.2-18. SHAFTPROXY targets the horizontal position of the flex coupling center spool.

Figure 3.4.2-18: Rotor Thrust

The forward bearing of Ramgen's rotor is loaded using differential pressure in the disk cavities. Rotor thrust is indicated by PBALTHRUST and RTRFWDTHRUST. PBALTHRUST is the pressure times area summation of aero loads on the disk faces. RTRFWDTHRUST is a load cell on the forward side of the forward bearing. Positive thrust indicates forward thrust. Design thrust loads are 2500-1300 lbf. Target thrust limits for this test were 2000-1400 lbf. Wheelspace pressures are the dominant thrust inputs for this test. Wheelspace supply and throttling valve positions are displayed on the chart as WSAFTAIRP1, WSAFTAIRP2, WSAFTAIRP3, WSAFTBPOUT, WSAFTDBPOUT, WSAFTSPLYP, and EXITDUCTP. A thrust load correction occurs at 14955 seconds indicated by a reduction in aft wheelspace pressure. This correction is nearly coincident with the onset of high vibrations.

Figure 3.4.2-19: Bearing race temperatures

All three bearing temperatures do not read. Best forward and aft thermocouples are connected. The devices had not yet been migrated to isolated transmitters. Voltage bleed for thermocouples results in a few degrees of error. Scaling issues with the transmitter resulted in under recorded temperatures. Peak forward bearing temperature should read 160 degF rather than 98.6.

Figure 3.4.2-20: Ramgen rotor bearing oil temperatures

Figure 3.4.2-21: Ramgen rotor bearing oil pressures

3.4.2.4 Comments on flex coupling installation

A John Crane TSCW-0033-W533-H129 coupling was used to connect the drive turbine to the Ramgen rotor. Coupling design parameters are reflected on the manufacturer's part drawing as follows:

- Torque Rating (Lb*In) 4300
- Peak Torque (Lb*In) 8600
- Speed (RPM) 46,500
- Angular Stiffness (Lb*In/Degree) 130
- Transmitted Axial Force (Lb) 160
- (Max at max deflection)

Misalignment capacity (Running condition)

- Max. Axial (In) 0.040 (bi-directional)
- Max. Parallel (In) 0.070
- Equivalent Angular (Degree/End) 0.33

For installation the coupling Hub (drive hub as indicated by reference number 2 on the part drawing HCA2211-617999-1) was detached. The location of the six bolts was noted for reinstallation.

The taper fit on the drive hub was inspected (70%+ contact as noted using layout die). The coupler was heated (heat gun, warm to touch) and installed on the drive shaft. Base on a discussion with the designer, stainless steel bolts were selected for installation (recommended for Ti, but used on both coupling hubs with a removable thread locker) and a "tight" torque was applied. The puller plate was removed and a drop measurement verified the 0.030

in. design gap between the shaft end and the puller plate. The puller plate was reinstalled and the removal screw tested. The hub was reheated in a similar manner and reinstalled with thread locker. The gap to the puller plate was not reverified.

The remained of the coupling, without the drive hub, was measured for length with and without the gag bolts. The gag bolts were reinstalled. The driven hub was warmed and installed on the rotor end using the same bolts type, torque and thread locker as the drive hub.

Boeing surveying service performed the laser alignment of the drive train using Leica equipment with Axyz Coordinate Measuring System V1.4.1.1 software. The drive shaft center was interpolated for comparison to the flex coupling, middle spool center and the Ramgen rotor center. The deep groove forward bearing installed for the laser alignment did not have detectable axial play, so alignment was performed without additional thrust load. The drive face and the aft support frame, aft face were compared for assessing the angular misalignment. At the completion of alignment the following conditions were achieved:

- Rotor to drive centerline deviation: 0.003 ± 0.003 in.
- Drive to rig planar misalignment: $0.016 \text{ deg} \pm 0.005$ (Assumes rotational axis are
- perpendicular to measured surface, add 0.016 uncertainty (0.005 flatness)
- for each machined faces for 0.048 deg.)
- After four translations, centerline actuation shift (return error) $< .005$ in.(same
- order of magnitude as the measurement uncertainty).
- Flex coupling free length is $14.382 \pm .002$ in.
- Rig to drive spacing for the coupling is $14.383 \pm .007$ in.

The rotor and coupling were then removed. A second set of bearings was installed and the rotor reinstalled in preparation for spin tests. The gag bolts were reinstalled in the flex coupling. The coupling was reinstalled as before. The frame was rolled into running position and the coupling to drive hub gap was measured. The free to gagged coupling differential length was compared to the gap measurement. This installation resulted in a coupling free state with 0.005 in. compression. Gag bolts were removed and cosmoline was cleaned for final installation.

During operation, axial deflection of the frame was monitored with a dial indicator mounted on the drive face reading the maximum (TDC) support frame deflection. Correlation to centerline deflection is made using data from preswirl nozzle capacity tests, wherein TDC and centerline deflection data was collected. In operation the inflatable seal generates a centerline axial deflection of 0.002 in. A supply plenum pressure of 19 psia produces an axial deflection of 0.007 in. (includes 0.002 in. of seal pressure). MAINSUPPLY on Figure 3.4.2-18 shows plenum pressure for Run 54 was below 16 psia.

3.4.3 Overview of Rotordynamic Analysis and Fixes Implemented After BP4 Conclusion

Extensive rotordynamic analysis and substantial component redesign occurred after the end of Budget Period 4 to fix the problem uncovered by the flex coupling failure. The following presentation gives an overview of the root cause analysis and results, as well as the current design status. A detailed description will be included in a future period report.

Rotordynamics Overview

- What is it?
  - Finite-element modeling and prediction of turbomachinery component critical speeds and system vibration behavior
  - Based on rotor and shaft structure, bearings, aero loading, imbalance
  - Industry-standard approach to avoiding vibration problems
- What tool was used?
  - DyRoBeS (Dynamics of Rotor Bearing Systems) software by Dr. W. J. Chen is widely used and respected in industry
- What problems can be detected?
  - Excessive vibration and bearing loads, insufficient damping, and imbalance loads can be found and obviated prior to test

0900-01178 DOE

Rampressor-2 Rotordynamics Approach

- Rotordynamics analysis subcontracted to Barbour Stockwell (BSi)
- Contractor had built the 800 kW drive turbine
- Contractor pedigree included design of drive turbine and success building turbines for many spin pit applications
- Rotordynamics analysis during conceptual design stage showed vibration problem at 44,000 RPM

0800-01178 DOE

Rampressor-2 Rotordynamics Approach

- Compressor stiffness significantly increased to push this mode beyond our operating envelope
- Custom aggressive-design bearings with above-normal stiffness
- Rotor shaft stiffened by moving bearings closer together
- Final analyses showed second vibration mode apparently moved out of range
- First vibration mode around 30,000 RPM deemed 'stable' in contractor's report

0900-01178 DOE

Ram-2 Coupling Failure

- Failure occurred February 15, 2006 during initial speed run
- Flex coupling and turbine shaft failed from excessive vibration
- Vibration onset nearly instantaneous at 18,000 RPM
- No model had predicted this
- No instrumentation indicated a problem before vibration onset

0900-01178 DOE

Rotordynamics Audit - FTT

- Florida Turbine Technologies (FTT) hired to investigate failure
- Team of rotordynamics experts performed audit of DyRoBeS model
- Attempt to explain failure with rotordynamics
- Found model error which had artificially increased stiffness of the coupling joint
- When corrected, predicted vibration had worsened significantly
- Suspicion was strong that rotordynamic modeling was key to finding root cause
- Despite extensive modeling, FTT was unable to re-create failure conditions to our satisfaction
- Turbine portion not audited due to belief that manufacturer would have used appropriate modeling techniques

0800-01178 DOE

Mechanical Fixes

- Suspicion that flex coupling was too heavy (16 lbs) led to replacement design
- FTT-designed quill shaft reduced weight (10 lbs less)
- Static structure stiffness increased with additional bracing
- Extensive instrumentation added to monitor shaft motion in real-time
- Compressor bearing clearances modified to eliminate possible problem
- Oil changed from mineral to synthetic
- Several other changes implemented to address possible problem areas

0800-01178 DOE

Rotordynamics Audit – Dr. Chen

- Work continued in parallel to identify true root cause
- Dr. W.J. Chen, DyRoBeS author, renowned rotordynamics scientist, and experienced turbomachinery engineer hired to audit the model
- Major problems found in turbine model
- Turbine damping incorrect by over two orders of magnitude
- Highly non-linear bearing spring rate had been modeled as linear
- Turbine damper design allowed some motion but 'bottomed out' quickly, causing massive vibration
- Modeled correctly, the 'smoking gun' was clear - the rig had badly flawed rotordynamics
- Realization reached the weekend before test resumption – short, careful test with new instrumentation and quill shaft confirmed model results

0900-01178 DOE

Rampressor-2 Redesign With Damping

- Ramgen brought Dr. Chen to Seattle to train Ramgen personnel in DyRoBeS and oversee a rapid assessment of redesign options
- In three days Ramgen personnel were trained, evaluated redesign scenarios and selected a design solution
- Ramgen has been forced to assume responsibility for re-bearing turbine
- Squeeze film dampers added to turbine bearings
- Compressor roller bearing replaced with tilting-pad hydrodynamic bearing
- Minimal changes to existing components

0900-01178 DOE

Rampressor-2 Redesign With Damping

- Damped models show rotordynamics have been fixed
- Dampers and tilting-pad bearing absorb vibration energy, 'spread' peak vibration across RPM range and reduce amplitude
- Dr. Chen is satisfied with the redesign and new results
- Ramgen, Dr. Chen and other experts are confident we've solved the problem

Example of February Configuration – DyRoBeS Crashes when Turbine 'Bottoms Out'

Redesigned Rampressor-2 With Damping

0900-01178 DOE

Return to Test

- Tilting-pad hydrodynamic bearings are complete and in-house
- Turbine modifications are starting production
- With accelerated manufacturing testing resumes last week of August
- With committed manufacturing testing first week of September
- Testing will resume with 'bald rotor' to ensure rotordynamics have been fixed
- Segmented rotor will be installed once full-speed operation is demonstrated
- Focus will eagerly shift to aero demonstration

0800-01178 DOE

CHAPTER 3 APPENDIX

Meas-Instru Correlation_Instrum ent number	Boeing SYM	Instrument Tagg	Description	Instru Tagg description	A-D responsibility	Units	On design Mean value	On design Min value	On design Max value
	PSH			Drive air supply line static pressure (static plenum for secondary air line rake)	Boeing Hyscan Pres Sys				
50003	WP	AS101CW	Facility flow meter	Main air inflow total mass flowrate - Boeing calculated	Boeing PLC	lbm/sec	3.05	0	4.5
	HUMID	AS101MH		Test cell humidity	Boeing PLC	%	60	75	0
	PFM2P	AS101MPd		Core rig air MCV downstream pressure for the facility critical flow venturis	Boeing PLC	psia	25	0	50
	PFM1P	AS101MPf		Core rig air MCV upstream pressure for the facility critical flow venturis	Boeing PLC	psia	45	0	300
6937		AS101MPs	Pressure transducer, 30 inMercury to 60 psig, 4- 20 mA, general purpose, 1/4 NPT	Main air secondary flow meter upstream pressure	Ramgen AB PLC	psia	23	0	30
40012	P0409	AS101MPs	Scanivalve Hyscan 50 psid module, vacuum reference	Ramgen test rig plenum pressure inflow static pressure	Boeing Hyscan Pres Sys	psia	23	0	30
	TFMP	AS101MTf		Core rig air MCV upstream temperature for the facility critical flow venturis	Boeing TC(k) Sys	degF	70	0	120
40070	TRTD01	AS101MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Test diffuser plenum temperature	Boeing PLC	degF	50	30	100
40012	P0410	AS102MPs	Scanivalve Hyscan 50 psid module, vacuum reference	Ramgen test rig plenum pressure inflow static pressure	Boeing Hyscan Pres Sys	psia	23	0	30
40070		AS102MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Test diffuser plenum temperature	Ramgen AB PLC	degF	50	30	100
40012	P0411	AS103MPs	Scanivalve Hyscan 50 psid module, vacuum reference	Ramgen test rig plenum pressure inflow static pressure	Boeing Hyscan Pres Sys	psia	23	0	30
40070	TRTD02	AS103MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Test diffuser plenum temperature	Boeing PLC	degF	50	30	100
40012	P0412	AS104MPs	Scanivalve Hyscan 50 psid module, vacuum reference	Ramgen test rig plenum pressure inflow static pressure	Boeing Hyscan Pres Sys	psia	23	0	30
40070	TRTD03	AS104MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Test diffuser plenum temperature	Boeing PLC	degF	50	30	100
		AS111CW		Main air flowrate using secondary fluid metering	Ramgen AB PLC				
40012	P0403	AS111MPd	Scanivalve Hyscan 50 psid module, vacuum reference	Bellmouth downstream/throat pressure	Boeing Hyscan Pres Sys	psia	14.7	7014.4	15
40012	P0401	AS111MPs	Scanivalve Hyscan 50 psid module, vacuum reference	Boeing flow conditioner plenum total pressure upstream of rig	Boeing Hyscan Pres Sys	psia	23	0	30
40012	P0404	AS112MPd	Scanivalve Hyscan 50 psid module, vacuum reference	Bellmouth downstream/throat pressure	Boeing Hyscan Pres Sys	psia	14.7	7014.4	15
40012	P0402	AS112MPs	Scanivalve Hyscan 50 psid module, vacuum reference	Boeing flow conditioner plenum static pressure upstream of rig	Boeing Hyscan Pres Sys	psia	23	0	30
40012	P0405	AS113MPd	Scanivalve Hyscan 50 psid module, vacuum reference	Bellmouth downstream/throat pressure	Boeing Hyscan Pres Sys	psia	14.7	7014.4	15
40012	P0406	AS114MPd	Scanivalve Hyscan 50 psid module, vacuum reference	Bellmouth downstream/throat pressure	Boeing Hyscan Pres Sys	psia	14.7	7014.4	15
40012	P0407	AS115MPd	Scanivalve Hyscan 50 psid module, vacuum reference	Bellmouth downstream/throat pressure	Boeing Hyscan Pres Sys	psia	14.7	7014.4	15
40012	P0408	AS116MPd	Scanivalve Hyscan 50 psid module, vacuum reference	Bellmouth downstream/throat pressure	Boeing Hyscan Pres Sys	psia	14.7	7014.4	15
	DPAS12	AS121MPd		Main air secondary flow meter differential pressure	Boeing PLC	psid	1	0	5
	WDT	AS201CW		Secondary air system mass flow rate	Boeing PLC	lbm/sec			

	PFM1S	AS201MPf		Secondary air system fluid meter upstream pressure	Boeing PLC	psia	275		
	TFMS	AS201MTf		Secondary air system fluid meter temperature	Boeing TC(k) Sys	degR	50		
	PFM2S	AS211MPd		Secondary air system fluid meter downstream pressure	Boeing PLC	psia	270		
		BED01MDx		Axial position of structural component					
		BED01MDx		Structural component displacement with eddy current probe	Ramgen AB PLC	mils	45	20	80
		BED01MDy		Structural component displacement with eddy current probe	Ramgen AB PLC	mils	45	20	80
		BED02MDx		Axial position of structural component					
	WBLA01	BLA01CWo		Aft performance bleed mass flow rate	Boeing PLC	lbm/sec	1.5	0.104	1.5
40087	PV1	BLA01MPf	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	Aft performance bleed flow path upstream flow meter static pressure from pressure system leg#1	Boeing PLC	psia	20	8	150
40190	P0503	BLA01MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bleed flow path plenum static pressure at rotor exit location 1	Boeing Hyscan Pres Sys	psia	20	8	150
	TV1	BLA01MTf		Aft performance bleed flow path flow meter temperature taken with 1/16 stnl sheath K-type TC with standard low noise connector	Boeing TC(k) Sys	degF	TBD	50	<700
40574	T0501	BLA01MTs	Thermocouple, k-type, 0.125 dia., ungrounded junction, 304 stnl sheath, mini connector, 6 in length	Aft bleed collector temperature	Boeing TC(k) Sys	degF			<700
	WBLA02	BLA02CWo		Aft starting bleed mass flow rate	Boeing PLC	lbm/sec	1.5	0.104	1.5
40087	PV2	BLA02MPf	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	Aft starting bleed flow path upstream (throat) flow meter static pressure from pressure system leg#2	Boeing PLC	psia	20	8	150
40190	P0504	BLA02MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bleed flow path plenum static pressure at rotor exit location 2	Boeing Hyscan Pres Sys	psia	20	8	150
	TV2	BLA02MTf		Aft starting bleed flow path flow meter temperature taken with 1/16 stnl sheath K type TC with standard low noise connector	Boeing TC(k) Sys	degF	TBD	50	<700
40574	T0502	BLA02MTs	Thermocouple, k-type, 0.125 dia., ungrounded junction, 304 stnl sheath, mini connector, 6 in length	Aft bleed collector temperature	Boeing TC(k) Sys	degF			<700
40190	P0507	BLA03MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bleed flow path plenum static pressure at rotor exit location 3	Boeing Hyscan Pres Sys	psia	20	8	150
40574	T0503	BLA03MTs	Thermocouple, k-type, 0.125 dia., ungrounded junction, 304 stnl sheath, mini connector, 6 in length	Aft bleed collector temperature	Boeing TC(k) Sys	degF			<700
40574	T0504	BLA04MTs	Thermocouple, k-type, 0.125 dia., ungrounded junction, 304 stnl sheath, mini connector, 6 in length	Aft bleed collector temperature	Boeing TC(k) Sys	degF			<700
40087	P0501	BLA11MPd	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	Aft performance bleed flow path venturi downstream pressure	Boeing Hyscan Pres Sys	psia	20	8	150
	P1203	BLA11MPf		Aft performance bleed flowmeter total pressure	Boeing Hyscan Pres Sys	psia	20	8	150
6937		BLA11MPs	Pressure transducer, 30 inMercury to 60 psig, 4-20 mA, general purpose, 1/4 NPT	Aft bleed flow path plenum static pressure at mid-duct location 1	Ramgen AB PLC	psia	20	8	150
40871	T0505	BLA11MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft bleed static temperature near rotor exit location 1	Boeing TC(k) Sys	degF			<700
	P0502	BLA12MPd		Aft starting bleed flow path venturi downstream pressure	Boeing Hyscan Pres Sys	psia	20	8	150

	P1204	BLA12MPf		Aft starting bleed flowmeter total pressure	Boeing Hyscan Pres Sys	psia	20	8	150
40190	P0505	BLA12MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bleed flow path plenum static pressure at mid-duct location 2	Boeing Hyscan Pres Sys	psia	20	8	150
40871	T0506	BLA12MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft bleed static temperature near rotor exit location 2	Boeing TC(k) Sys	degF			<700
40190	P0506	BLA13MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bleed flow path plenum static pressure at mid-duct location 3	Boeing Hyscan Pres Sys	psia	20	8	150
40871	T0507	BLA13MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft bleed static temperature near rotor exit location 3	Boeing TC(k) Sys	degF			<700
	DPV2	BLA21MPd		Differential pressure aft performance bleed flow meter	Boeing PLC	psid	45	0	50
	DPV1	BLA22MPd		Differential pressure aft starting bleed flow meter	Boeing PLC	psid	45	0	50
	WBLF01	BLF01CWo		Forward bleed mass flow rate	Boeing PLC	lbm/sec	TBD	TBD	TBD(.25)
40087	PV4	BLF01MPf	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	Forward bleed flow path flow meter static pressure leg #3	Boeing PLC	psia	9	6	14.7
40190	P0511	BLF01MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Forward bleed flow path plenum static pressure at rotor exit location 2	Boeing Hyscan Pres Sys	psia	9	6	14.7
	TV4	BLF01MTf		Forward bleed flow path flow meter temperature 1/16 stnl sheath K-type Tc with standard low noise connector	Boeing TC(k) Sys	degF	TBD(150)	20	TBD(150)
40871	T0508	BLF01MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Fwd bleed passage static temperature location 1	Boeing TC(k) Sys	degF	650	50	700
40190	P0512	BLF02MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Forward bleed flow path plenum static pressure at rotor exit location 1	Boeing Hyscan Pres Sys	psia	9	6	14.7
40871	T0509	BLF02MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Fwd bleed passage static temperature location 2	Boeing TC(k) Sys	degF	650	50	700
40190	P0513	BLF03MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Forward bleed flow path plenum static pressure at rotor exit location 3	Boeing Hyscan Pres Sys	psia	9	6	14.7
40871	T0510	BLF03MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Fwd bleed passage static temperature location 3	Boeing TC(k) Sys	degF	650	50	700
40871	T0511	BLF04MTs	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Fwd bleed passage static temperature location 4	Boeing TC(k) Sys	degF	650	50	700
40087	P0508	BLF11MPd	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	Forward bleed flow path venturi downstream pressure leg #3	Boeing Hyscan Pres Sys	psia	9	6	14.7
6937		BLF11MPs	Pressure transducer, 30 inMercury to 60 psig, 4-20 mA, general purpose, 1/4 NPT	Forward bleed flow path plenum mid passage static pressure location 1	Ramgen AB PLC	psia	9	6	14.7
40190	P0509	BLF12MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Forward bleed flow path plenum mid passage static pressure location 3	Boeing Hyscan Pres Sys	psia	9	6	14.7
40190	P0510	BLF13MPs	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Forward bleed flow path plenum mid passage static pressure location 2	Boeing Hyscan Pres Sys	psia	9	6	14.7
	DPV4	BLF21MPd		Differential pressure forward bleed flow meter	Boeing PLC	psid	45	0	50
	WBRA01	BRA01CWi		Aft bearing shaft seal buffer supply flow rate	Boeing PLC	lbm/sec	TBD	TBD	TBD
40190	P1015	BRA01MPd	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bearing shaft seal buffer supply flow meter downstream/throat pressure	Boeing Hyscan Pres Sys	psia	25	14.7	45
40190	P1016	BRA01MPf	Reducer bushing, 1 in. NPT(M) to 1/4 in. NPT(F), steel	Aft bearing shaft seal buffer supply flow meter upstream pressure	Boeing Hyscan Pres Sys	psia	85	40	120

	T0101	BRA01MTf		Aft bearing shaft seal buffer supply flow meter supply air temperature	Boeing TC(k) Sys	degF	50	40	70
40871		BRA01MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft bearing journal temperature	Ramgen AB PLC	degF	180	50	250
40871		BRA02MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft bearing race temperature 90 deg.	Ramgen AB PLC	degF	180	50	250
40871		BRA03MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft bearing journal temperature	Ramgen AB PLC	degF	180	50	250
	DPBRA1	BRA11MPd		Shaft seal air flowmeter differential pressure by transducer	Boeing PLC	psia	25	14.7	45
7006		BRA11MPs	Pressure transducer, 0-30 psig, 4-20 mA, intrinsically safe, 1/2 NPT	Aft bearing air purge supply pressure with sealed transducer	Ramgen AB PLC	psia	20	14.7	40
6947		BRA21MPs	Pressure transducer, 0-30 psig, 4-20 mA, general purpose, 1/4 NPT	Aft bearing cavity pressure must be acquired with a sealed transducer	Ramgen AB PLC	psia	15.5	14.7	20
40961		BRA22MPs	Pressure transducer, 0-15 psig, 4-20 mA, general purpose, 1/4 NPT	Aft bearing cavity pressure must be acquired with a sealed transducer	Ramgen AB PLC	psia	15.5	14.7	20
6609		BRA23MPs	Pressure transducer, 0-25 psig, 4-20 mA, general purpose, 1/4 NPT	Aft bearing cavity pressure must be acquired with a sealed transducer	Ramgen AB PLC	psia	15.5	14.7	20
40871		BRF01MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward bearing journal temperature	Ramgen AB PLC	degF	180	50	250
40871		BRF02MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward bearing race temperature stainless sheath K-type 0 deg.	Ramgen AB PLC	degF	180	50	250
40871		BRF03MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward bearing race temperature stainless sheath K-type 0 deg.	Ramgen AB PLC	degF	180	50	250
7006		BRF21MPs	Pressure transducer, 0-30 psig, 4-20 mA, intrinsically safe, 1/2 NPT	Forward bearing cavity pressure taken with a sealed pressure transducer	Ramgen AB PLC	psia	20	14.7	20
6947		BRF22MPs	Pressure transducer, 0-30 psig, 4-20 mA, general purpose, 1/4 NPT	Forward bearing cavity pressure taken with a sealed pressure transducer	Ramgen AB PLC	psia	20	14.7	20
6609		BRF23MPs	Pressure transducer, 0-25 psig, 4-20 mA, general purpose, 1/4 NPT	Forward bearing cavity pressure taken with a sealed pressure transducer	Ramgen AB PLC	psia	20	14.7	20
		COL01MCv		Back pressure valve position signal 4-20 mA control signal form 1 channel of an Allen-Bradley control board	Ramgen AB PLC	mA	20	4	20
41087	P0901	COL01MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exhaust collector plenum pressure 0 deg.	Boeing Hyscan Pres Sys	psia	150	14.7	165
40070	TRTD04	COL01MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Discharge total temperature in the collector 0 deg.	Boeing PLC	degF	650	50	800
		COL02MCv		Core flow throttle valve position feedback signal	Ramgen AB PLC	mA	20	4	20
41087	P0902	COL02MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exhaust collector plenum pressure 90 deg.	Boeing Hyscan Pres Sys	psia	150	14.7	165
40070	TRTD05	COL02MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Discharge total temperature in the collector 90 deg.	Boeing PLC	degF	650	50	800
41087	P0903	COL03MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exhaust collector plenum pressure 180 deg.	Boeing Hyscan Pres Sys	psia	150	14.7	165
40070	TRTD06	COL03MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Discharge total temperature in the collector 180 deg.	Boeing PLC	degF	650	50	800
41087	P0904	COL04MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exhaust collector plenum pressure 270 deg.	Boeing Hyscan Pres Sys	psia	150	14.7	165
40070	TRTD07	COL04MTs	RTD, 100 ohm platinum, european cal, 1/8 in. dia., 6 in. length	Discharge total temperature in the collector 270 deg.	Boeing PLC	degF	650	50	800

40033		CSA01MAo	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Case vibration case aft theta (in the bearing plane)	Ramgen AB PLC	ips	0.5	0	5
40033		CSA01Mar	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Case vibration case aft r (in the bearing plane)	Ramgen AB PLC	ips	0.5	0	5
40033		CSA01Maz	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Case vibration case aft z (in the bearing plane)	Ramgen AB PLC	ips	0.5	0	5
40527		CSA11MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Aft frame stanchion metal temperature	Ramgen AB PLC	degF	<80	50	80
40527	T0102	CSA12MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Aft frame stanchion metal temperature	Boeing TC(k) Sys	degF	<80	50	80
40527		CSA13MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Aft frame stanchion metal temperature chiller input	Independent	degF	<80	50	80
40871	T0104	CSA21MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft frame metal temperature near bearing	Boeing TC(k) Sys	degF	<350	50	350
40871	T0105	CSA22MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft frame metal temperature near bearing	Boeing TC(k) Sys	degF	<350	50	350
	T0106	CSA23MTm		Aft frame metal temperature near bearing location 3	Boeing TC(k) Sys	degF	<350	50	350
		CSA31MTi		Aft frame cooling water supply temperature	Ramgen AB PLC	degF	50		
		CSA41MTo		Aft frame cooling water return temperature	Ramgen AB PLC	degF	55		
40033		CSF01MAo	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Case vibration case fore theta (in the bearing plane)	Ramgen AB PLC	ips	0.5	0	5
40033		CSF01Mar	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Case vibration case fore r (in the bearing plane)	Ramgen AB PLC	ips	0.5	0	5
40033		CSF01Maz	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Case vibration case fore z (in the bearing plane)	Ramgen AB PLC	ips	0.5	0	5
40527	T0107	CSF11MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Forward frame stanchion metal temperature	Boeing TC(k) Sys	degF	<80	50	80
40527	T0108	CSF12MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Forward frame stanchion metal temperature	Boeing TC(k) Sys	degF	<80	50	80
40527	T0109	CSF13MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Forward frame stanchion metal temperature	Boeing TC(k) Sys	degF	<80	50	80
40871	T0110	CSF21MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward frame bearing hub metal temperature	Boeing TC(k) Sys	degF	<350	50	350
40871	T0111	CSF22MTm	Thermocouple, k-type, 0.040 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward frame bearing hub metal temperature	Boeing TC(k) Sys	degF	<350	50	350
	T0112	CSF23MTm		Forward frame bearing hub metal temperature	Boeing TC(k) Sys	degF	<350	50	350
40013	P1005	DCA11MPs	Scanivalve Hyscan 100 psid module, atmospheric reference	Aft wheel space static pressure distribution shroud hub position 0 deg.	Boeing Hyscan Pres Sys	psia	20	14.7	35
40922	T0208	DCA11MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft wheel space shroud surface temperature near the disk hub at 0 deg.	Boeing TC(k) Sys	degF	150	50	260

6934		DCA12MPs	Pressure transducer, 0-60 psig, 4-20 mA, general purpose, 1/4 NPT	Aft wheel space static pressure distribution shroud hub position 120 deg.	Ramgen AB PLC	psia	20	14.7	35
40922	T0209	DCA12MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft wheel space shroud surface temperature near the disk hub at 240 deg.	Boeing TC(k) Sys	degF	150	50	260
40013	P1006	DCA21MPs	Scanivalve Hyscan 100 psid module, atmospheric reference	Aft wheel space static pressure distribution shroud middle position 0 deg.	Boeing Hyscan Pres Sys	psia	20	14.7	35
40922	T0210	DCA21MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft wheel space shroud surface temperature near the disk center at 0 deg.	Boeing TC(k) Sys	degF	150	50	260
6579		DCA22MPs	Pressure transducer, 0-100 psig, 4-20 mA, general purpose, 1/4 NPT	Aft wheel space static pressure distribution shroud middle position 120 deg.	Ramgen AB PLC	psia	20	14.7	35
40922	T0211	DCA22MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft wheel space shroud surface temperature near the disk center at 240 deg.	Boeing TC(k) Sys	degF	150	50	260
40013	P1007	DCA31MPs	Scanivalve Hyscan 100 psid module, atmospheric reference	Aft wheel space static pressure distribution shroud rim position 0 deg.	Boeing Hyscan Pres Sys	psia	20	14.7	35
40922	T0212	DCA31MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft wheel space shroud surface temperature near the disk rim at 0 deg.	Boeing TC(k) Sys	degF	150	50	260
6579		DCA32MPs	Pressure transducer, 0-100 psig, 4-20 mA, general purpose, 1/4 NPT	Aft wheel space static pressure distribution shroud rim position 120 deg.	Ramgen AB PLC	psia	20	14.7	35
40922	T0213	DCA32MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Aft wheel space shroud surface temperature near the disk rim at 240 deg.	Boeing TC(k) Sys	degF	150	50	260
40013	P1012	DCF11MPs	Scanivalve Hyscan 100 psid module, atmospheric reference	Forward wheel space static pressure distribution shroud at the disk hub 0 deg.	Boeing Hyscan Pres Sys	psia	20	14.7	50
6579		DCF12MPs	Pressure transducer, 0-100 psig, 4-20 mA, general purpose, 1/4 NPT	Forward wheel space static pressure distribution shroud at the disk hub 120 deg.	Ramgen AB PLC	psia	20	14.7	50
40013	P1013	DCF21MPs	Scanivalve Hyscan 100 psid module, atmospheric reference	Forward wheel space static pressure distribution shroud at the disk mid-line 0 deg.	Boeing Hyscan Pres Sys	psia	20	14.7	50
40922	T0214	DCF21MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward wheel space shroud temperature at the middle 0 deg.	Boeing TC(k) Sys	degF	150	50	260
6934		DCF22MPs	Pressure transducer, 0-60 psig, 4-20 mA, general purpose, 1/4 NPT	Forward wheel space static pressure distribution shroud at the disk mid-line 120 deg.	Ramgen AB PLC	psia	20	14.7	50
40922	T0215	DCF22MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward wheel space shroud temperature at the middle 120 deg.	Boeing TC(k) Sys	degF	150	50	260
40013	P1014	DCF31MPs	Scanivalve Hyscan 100 psid module, atmospheric reference	Forward wheel space static pressure distribution shroud at the disk rim 0 deg.	Boeing Hyscan Pres Sys	psia	20	14.7	50
40922	T0216	DCF31MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward wheel space shroud temperature at the rim 0 deg.	Boeing TC(k) Sys	degF	150	50	260
6934		DCF32MPs	Pressure transducer, 0-60 psig, 4-20 mA, general purpose, 1/4 NPT	Forward wheel space static pressure distribution shroud at the disk rim 120 deg.	Ramgen AB PLC	psia	20	14.7	50

40922	T0217	DCF32MTs	Thermocouple, k-type, 0.040 dia., exposed junction, 304 stnl sheath, 425 degF mini connector, 120 in length	Forward wheel space shroud temperature at the rim 120 deg.	Boeing TC(k) Sys	degF	150	50	260
40033		DRV01MAo	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Drive vibration theta	Ramgen AB PLC	ips	0.05	0	1
40033		DRV01MAr	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Drive vibration r	Ramgen AB PLC	ips	0.05	0	1
40033		DRV01MAz	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Drive vibration z	Ramgen AB PLC	ips	0.05	0	1
	WDRV	DRV01MCW		Drive supply air calculated mas flow rate	Boeing PLC	lbm/sec	10.5	10	13
		DRV01MDx		Driven hub x displacement	Ramgen AB PLC	mils	45	20	80
		DRV01MDy		Driven hub y displacement	Ramgen AB PLC	mils	45	20	80
		DRV01MDz		Drive turbine shaft position, axial	Ramgen AB PLC	mils	45	20	80
		DRV01MDz		Drive turbine shaft axial position		mils	45	20	80
	P1201	DRV01MPf		Drive supply air fluid meter upstream pressure	Boeing Hyscan Pres Sys	psia	290	14.7	295
	TFMSX	DRV01MTf		Drive supply air fluid meter temperature	Boeing TC(k) Sys	degF	50	40	95
		DRV01MTm		Drive forward bearing temp	Ramgen AB PLC	degF	180	50	250
		DRV01MVo		Drive output RPM from drive sensor split	BSI turbine controller	rpm	41836	0	42000
	DRVSTP	DRV01MXi		drive emergency shutdown signal out	Boeing PLC	high/low			
		DRV01MXo		drive emergency shutdown signal out	Ramgen AB PLC	high/low			
		DRV02C		Rotor Drive power consumption	Ramgen AB PLC	hp	550	0	1200
	DRVVLV	DRV02MCv		Drive system control air snap valve control signal	Ramgen AB PLC	Volts			
		DRV02MTm		Drive aft bearing temp	Ramgen AB PLC	degF	180	50	250
		DRV02MVo		Drive output RPM from drive sensor split to Ramgen PLC	Ramgen AB PLC	rpm	41836	0	42000
	DRVVRPM	DRV03MVo		Drive output RPM from drive sensor split to Boeing PLC	Boeing PLC	rpm	41836	0	42000
	P1202	DRV11MPd		Drive supply air pitot tube static pressure	Boeing Hyscan Pres Sys	psi	290	14.7	295
	T0113	DRV11MTi		Rotor Drive supply temperature	Boeing TC(k) Sys	degF	350	50	400
	T0611	DRV11MTo		Rotor Drive exhaust temperature	Boeing TC(k) Sys	degF	50	-20	400
	T0114	DRV12MTi		Rotor Drive supply temperature	Boeing TC(k) Sys	degF	350	50	400
	T0612	DRV12MTo		Rotor Drive exhaust temperature	Boeing TC(k) Sys	degF	50	-20	400
	T0115	DRV13MTi		Rotor Drive supply temperature	Boeing TC(k) Sys	degF	350	50	400
	T0613	DRV13MTo		Rotor Drive exhaust temperature	Boeing TC(k) Sys	degF	50	-20	400
	T0116	DRV14MTi		Rotor Drive supply temperature	Boeing TC(k) Sys	degF	350	50	400
	T0614	DRV14MTo		Rotor Drive exhaust temperature	Boeing TC(k) Sys	degF	50	-20	400
	DPDRV	DRV21MPd		Drive supply air fluid meter differential pressure	Boeing PLC	psi	290	14.7	295
	TFMSX	DRV21MTi		Drive turbine heat exchanger supply air temperature; (secindary system MCV downstream temp)	Boeing TC(k) Sys	degF	50	40	95
	THXB	DRV21MTo		Drive turbine heat exchanger #1 exhaust air temperature	Boeing TC(k) Sys	degF	350	50	400
	THXA	DRV22MTo		Drive turbine heat exchanger #2 exhaust air temperature	Boeing TC(k) Sys	degF	350	50	400
	WRE	EXH01CWo		Exhaust flow through an ASME nozzle - Boeing calculated	Boeing PLC	lbm/sec	2.8	0	3.9
40009	P1106	EXH01MPf	Scanivalve Hyscan 15 psid module atmospheric reference	Exhaust air upstream pressure for the flow measuring device	Boeing Hyscan Pres Sys	psia	45	0	50
	T0311	EXH01MTf		Main intake air flow meter upstream temperature with 1/16 TC stnl sheath type E standard low noise connector	Boeing TC(k) Sys	degF	550	40	700
40009	P1105	EXH02MPf	Scanivalve Hyscan 15 psid module atmospheric reference	Exhaust air upstream pressure for the flow measuring device	Boeing Hyscan Pres Sys	psia	45	0	50

	T0615	EXH02MTf		Core rig exhaust air flow meter upstream temperature with 1/16 TC stnl sheath type E standard low noise connector	Boeing TC(k) Sys	degF	550	40	700
40009	P1104	EXH03MPf	Scanivalve Hyscan 15 psid module atmospheric reference	Exhaust air upstream pressure for the flow measuring device	Boeing Hyscan Pres Sys	psia	45	0	50
	T0513	EXH03MTf		Main intake air flow meter upstream temperature with 1/16 TC stnl sheath type E standard low noise connector	Boeing TC(k) Sys	degF	550	40	700
40009	P1103	EXH04MPf	Scanivalve Hyscan 15 psid module atmospheric reference	Exhaust air upstream pressure for the flow measuring device	Boeing Hyscan Pres Sys	psia	45	0	50
	T0514	EXH04MTf		Main intake air flow meter upstream temperature with 1/16 TC stnl sheath type E standard low noise connector	Boeing TC(k) Sys	degF	550	40	700
40009	P1110	EXH11MPd	Scanivalve Hyscan 15 psid module atmospheric reference	Core rig flow exhaust nozzle downstream pressure	Boeing Hyscan Pres Sys	psia	14.7	14.5	18
40009	P1107	EXH12MPd	Scanivalve Hyscan 15 psid module atmospheric reference	Core rig flow exhaust nozzle downstream pressure	Boeing Hyscan Pres Sys	psia	14.7	14.5	18
40009	P1108	EXH13MPd	Scanivalve Hyscan 15 psid module atmospheric reference	Core rig flow exhaust nozzle downstream pressure	Boeing Hyscan Pres Sys	psia	14.7	14.5	18
40009	P1109	EXH14MPd	Scanivalve Hyscan 15 psid module atmospheric reference	Core rig flow exhaust nozzle downstream pressure	Boeing Hyscan Pres Sys	psia	14.7	14.5	18
	DPRE	EXH21MPd		Core rig flow exhaust nozzle differential pressure	Boeing PLC	psid	3	0	5
		FAC01MXi		Facility shutdown signal in	Ramgen AB PLC	high/low			
		FAC01MXo		Facility shutdown signal out	Ramgen AB PLC	high/low			
6579		LOA01MPi	Pressure transducer, 0-100 psig, 4-20 mA, general purpose, 1/4 NPT	Aft bearing intake lube oil pressure	Ramgen AB PLC	psia	40	25	50
		LOA01MTi		Aft bearing oil input temperature	Ramgen AB PLC	degF	140	50	165
	T0701	LOA01MTo		Aft bearing oil exit temperature	Boeing TC(k) Sys	degF	185	150	250
		LOA01MWi		Aft bearing oil flow rate	Ramgen AB PLC	gpm	TBD	0	1
6934		LOA02MPi	Pressure transducer, 0-60 psig, 4-20 mA, general purpose, 1/4 NPT	Aft bearing intake lube oil pressure	Ramgen AB PLC	psia	40	25	50
	T0206	LOA02MTi		Aft bearing oil input temperature	Boeing TC(k) Sys	degF	140	50	165
		LOA02MTo		Aft bearing oil exit temperature	Ramgen AB PLC	degF	185	150	250
6579		LOF01MPi	Pressure transducer, 0-100 psig, 4-20 mA, general purpose, 1/4 NPT	Forward bearing intake lube oil pressure	Ramgen AB PLC	psia	40	25	50
		LOF01MTi		Forward bearing oil input temperature	Ramgen AB PLC	degF	140	50	165
	T0702	LOF01MTo		Forward bearing oil exit temperature	Boeing TC(k) Sys	degF	185	150	250
		LOF01MWi		Forward bearing oil flow rate	Ramgen AB PLC	gpm		0	1
6934		LOF02MPi	Pressure transducer, 0-60 psig, 4-20 mA, general purpose, 1/4 NPT	Forward bearing intake lube oil pressure	Ramgen AB PLC	psia	40	25	50
		LOF02MTi		Forward bearing oil input temperature	Ramgen AB PLC	degF	140	50	165
		LOF02MTo		Forward bearing oil exit temperature	Ramgen AB PLC	degF	185	150	250
	WBPL	LOS01CWo		Lube system exhaust air flow rate	Boeing PLC	lbm/sec	TBD		
		LOS01MCv		Lube oil pump on/off	Ramgen AB PLC	on/off			
		LOS01MLv		Oil reservoir level switch	Ramgen AB PLC	open/closed	open	closed	open
40009	P1101	LOS01MPf	Scanivalve Hyscan 15 psid module atmospheric reference	Lube oil system reservoir vent flowmeter upstream pressure	Boeing Hyscan Pres Sys	psia	15.8	14.7	18
	T0205	LOS01MTf		Lube oil system reservoir vent flowmeter upstream temperature	Boeing TC(k) Sys	psia	70	50	100
		LOS01MTs		Lube oil reservoir temperature	Ramgen AB PLC	degF	140	50	165
		LOS02MCv		Lube oil scavenge pump on/off	Ramgen AB PLC	on/off	on	off	on
		LOS02MLv		Scavenge oil reservoir level switch	Ramgen AB PLC	open/closed	open	closed	open
40009	P1102	LOS11MPd	Scanivalve Hyscan 15 psid module atmospheric reference	Lube oil system reservoir vent flowmeter downstream pressure	Boeing Hyscan Pres Sys	psia	14.8	14.7	16

	DPBL	LOS21MPd		Lube oil system reservoir vent flowmeter differential pressure	Boeing PLC	psid	1	0.5	3
		MTST01S		time relative	Boeing PLC				
		MTST01Sa		time absolute	Boeing PLC				
41089	PAMB	PAMB	Scanivalve Hyscan 30 psid module, vacuum reference	Cell ambient pressure	Boeing PLC	psia			
	PBDP	PBDP		Cell 1 Primary Burst Disk Pressure	Boeing PLC	psia			
	PBDS	PBDS		Cell 1 Secondary Burst Disk Pressure	Boeing PLC	psia			
41087	PCHK1	PCHK1	Scanivalve Hyscan 200 psid module, atmospheric reference	Ref Pressure for Exit Rakes	Boeing Hyscan Pres Sys	psia			
41087	PCHK10	PCHK10	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 10	Boeing Hyscan Pres Sys	psi			
41087	PCHK11	PCHK11	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 11	Boeing Hyscan Pres Sys	psi			
41087	PCHK12	PCHK12	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 12	Boeing Hyscan Pres Sys	psi			
41087	PCHK13	PCHK13	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 13	Boeing Hyscan Pres Sys	psi			
41087	PCHK14	PCHK14	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 14	Boeing Hyscan Pres Sys	psi			
41087	PCHK15	PCHK15	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 15	Boeing Hyscan Pres Sys	psi			
41087	PCHK16	PCHK16	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 16	Boeing Hyscan Pres Sys	psi			
41087	PCHK2	PCHK2	Scanivalve Hyscan 200 psid module, atmospheric reference	Ref Pressure for Exit Rakes Probe	Boeing Hyscan Pres Sys	psia			
41087	PCHK3	PCHK3	Scanivalve Hyscan 200 psid module, atmospheric reference	Ref Pressure for Cobra Probes	Boeing Hyscan Pres Sys	psia			
41087	PCHK4	PCHK4	Scanivalve Hyscan 200 psid module, atmospheric reference	Ref Pressure for Exit Statics	Boeing Hyscan Pres Sys	psia			
41087	PCHK5	PCHK5	Scanivalve Hyscan 200 psid module, atmospheric reference	EPS Module Check w/ applied Dead Weight Load	Boeing Hyscan Pres Sys	psi			
41087	PCHK6	PCHK6	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 6	Boeing Hyscan Pres Sys	psi			
41087	PCHK7	PCHK7	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 7	Boeing Hyscan Pres Sys	psi			
41087	PCHK8	PCHK8	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 8	Boeing Hyscan Pres Sys	psi			
41087	PCHK9	PCHK9	Scanivalve Hyscan 200 psid module, atmospheric reference	Reference pressure; module 9	Boeing Hyscan Pres Sys	psi			
	PREF	PREF		Local atmospheric pressure outside the facility	Boeing PLC	psia			
40012	P0101	PSN01MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet hub static 0 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0102	PSN02MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet hub static 90 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0103	PSN03MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet hub static 180 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0104	PSN04MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet hub static 270 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0105	PSN11MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet tip/shroud static 0 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0106	PSN12MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet tip/shroud static 90 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0107	PSN13MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet tip/shroud static 180 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30

40012	P0108	PSN14MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution inlet tip/shroud static 270 deg.	Boeing Hyscan Pres Sys	psia	15.02	10	30
40012	P0109	PSN21MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution exit hub static 0 deg.	Boeing Hyscan Pres Sys	psia	10	5	30
40012	P0110	PSN22MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution exit hub static 90 deg.	Boeing Hyscan Pres Sys	psia	10	5	30
40012	P0111	PSN23MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution exit hub static 180 deg.	Boeing Hyscan Pres Sys	psia	10	5	30
40012	P0112	PSN24MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN static pressure distribution exit hub static 270 deg.	Boeing Hyscan Pres Sys	psia	10	5	30
40012	P0201	PSN2CMPt	Scanivalve Hyscan 50 psid module, vacuum reference	Preswirl nozzle cobra probe center pressure	Boeing Hyscan Pres Sys	psia	14	10	30
40012	P0202	PSN2LMPt	Scanivalve Hyscan 50 psid module, vacuum reference	Preswirl nozzle cobra probe left pressure	Boeing Hyscan Pres Sys	psia	14	10	30
40012	P0203	PSN2RMPt	Scanivalve Hyscan 50 psid module, vacuum reference	Preswirl nozzle cobra probe right pressure	Boeing Hyscan Pres Sys	psia	14	10	30
6935		PSN91MPs	Pressure transducer, 30 inMercury to 15 psig, 4-20 mA, general purpose, 1/4 NPT	PSN survey plane tip/shroud static 45 deg. 95% length	Ramgen AB PLC	psia	10	5	30
40012	P0113	PSN92MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN survey plane tip/shroud static 135 deg. 95% length	Boeing Hyscan Pres Sys	psia	10	5	30
40012	P0114	PSN93MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN survey plane tip/shroud static 225 deg. 95% length	Boeing Hyscan Pres Sys	psia	10	5	30
40012	P0115	PSN94MPs	Scanivalve Hyscan 50 psid module, vacuum reference	PSN survey plane tip/shroud static 315 deg. 95% length	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0204	PSNA1MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane hub static 45 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0205	PSNA2MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane hub static 135 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0206	PSNA3MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane hub static 225 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0207	PSNA4MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane hub static 315 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0208	PSNB1MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane tip static 315 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0209	PSNB2MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane tip static 135 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0210	PSNB3MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane tip static 225 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0211	PSNB4MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey plane tip static 45 deg	Boeing Hyscan Pres Sys	psia	10	5	30
41090	P0301	PSNC1MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static TE pitch location 0 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0302	PSNC2MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static TE pitch location 90 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0303	PSNC3MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud staticTE pitch location 180 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0304	PSNC4MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud staticTE pitch location 270 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0305	PSND1MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 25% pitch location 0 deg.	Boeing Hyscan Pres Sys	psia	10	5	16

41090	P0306	PSND2MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 25% pitch location 90 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0307	PSND3MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 25% pitch location 180 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0308	PSND4MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 25% pitch location 270 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0309	PSNE1MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static mid pitch location 0 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0310	PSNE2MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static mid pitch location 90 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0311	PSNE3MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static mid pitch location 180 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0312	PSNE4MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static mid pitch location 270 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0313	PSNF1MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 75% pitch location 0 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0314	PSNF2MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 75% pitch location 90 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0315	PSNF3MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 75% pitch location 180 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
41090	P0316	PSNF4MPs	Scanivalve Hyscan 100 psid module, vacuum reference	PSN survey static pressure distribution exit tip/shroud static 75% pitch location 270 deg.	Boeing Hyscan Pres Sys	psia	10	5	16
	PTARE	PTARE		Burst disk pressure; primary air line; T'd into PBDP; redundant control parameter	Boeing PLC	psia			
	PTARES	PTARES		Burst disk pressure; secondary air line; T'd into PBDS; redundant control parameter	Boeing PLC	psia			
40012	PTC1	PTC1	Scanivalve Hyscan 50 psid module, vacuum reference	Station 1 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC10	PTC10	Scanivalve Hyscan 50 psid module, vacuum reference	Station 10 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC11	PTC11	Scanivalve Hyscan 50 psid module, vacuum reference	Station 11 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC12	PTC12	Scanivalve Hyscan 50 psid module, vacuum reference	Station 12 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC13	PTC13	Scanivalve Hyscan 50 psid module, vacuum reference	Station 13 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC14	PTC14	Scanivalve Hyscan 50 psid module, vacuum reference	Station 14 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC15	PTC15	Scanivalve Hyscan 50 psid module, vacuum reference	Station 15 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC16	PTC16	Scanivalve Hyscan 50 psid module, vacuum reference	Station 16 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC2	PTC2	Scanivalve Hyscan 50 psid module, vacuum reference	Station 2 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50

40012	PTC3	PTC3	Scanivalve Hyscan 50 psid module, vacuum reference	Station 3 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC4	PTC4	Scanivalve Hyscan 50 psid module, vacuum reference	Station 4 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC5	PTC5	Scanivalve Hyscan 50 psid module, vacuum reference	Station 5 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC6	PTC6	Scanivalve Hyscan 50 psid module, vacuum reference	Station 6 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC7	PTC7	Scanivalve Hyscan 50 psid module, vacuum reference	Station 7 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC8	PTC8	Scanivalve Hyscan 50 psid module, vacuum reference	Station 8 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
40012	PTC9	PTC9	Scanivalve Hyscan 50 psid module, vacuum reference	Station 9 of 16 station rig inlet pressure rake	Boeing Hyscan Pres Sys	psia	25	0	50
		REG01MPs		Aft WS back pressure regulator, dome pressure	Ramgen AB PLC	psig	65	0	150
40920		RTR01MDx	Proximity probe, 5 mm dia., 1/4-28 UNF threaded body, 2 in case length, unarmored, 0.5 meter cable	Shaft run-out aft/driven shaft end x	Ramgen AB PLC	mils	42200	0	42200
40920		RTR01MDy	Proximity probe, 5 mm dia., 1/4-28 UNF threaded body, 2 in case length, unarmored, 0.5 meter cable	Shaft run-out aft/driven shaft end y	Ramgen AB PLC	mils	42200	0	42200
40918		RTR01MDz	Proximity probe, 5 mm dia., 1/4-28 UNF threaded body, 1.5 in case length, unarmored, 0.5 meter cable	Rotor axial position from nominal TBD location	Ramgen AB PLC	mils	10	0	50
122415		RTR01MFz	Donut load cell, 0-5000 lbf FS range, model D derivative	Rotor thrust axial load forward direction	Ramgen AB PLC	lbf	-1039	-1400	800
40548		RTR01MvR	Hall effect pick up, 5/8-18 thread, 0-5V ouput, 5.5-25 VDC supply	Rotor RPM key phasor/encoder or mag pick up once per rev	Boeing Oscilloscope	rpm	41800	0	55000
40548		RTR02MvR	Hall effect pick up, 5/8-18 thread, 0-5V ouput, 5.5-25 VDC supply	Rotor RPM key phasor/encoder or mag pick up once per rev	Boeing Oscilloscope	rpm	41800	0	55000
	T0218	RTR11MTs		Flex coupling windage temperature	Boeing TC(k) Sys	degF	90	50	200
	YCBA	SED01MDj		Rotadata actuator angular position	Boeing PLC	deg	-84	90	-90
	XCBA	SED01MDr		Rotadata actuator radial position	Boeing PLC	inches	5.274	5.088	5.46
41087	P0701	SED01MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure rotor exit 45 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0702	SED02MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure rotor exit 135 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0703	SED03MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure rotor exit 225 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0704	SED04MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure rotor exit 315 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
		SED11MDj		Exit rake 1 angular position	Independent	deg	-84	90	-90
41087	P0604	SED11MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 1 station 1	Boeing Hyscan Pres Sys	psia	125	14.7	150
122509	T0301	SED11MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 1 station 1	Boeing TC(k) Sys	degF	551	50	800
41087	P0605	SED12MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 1 station 2	Boeing Hyscan Pres Sys	psia	125	14.7	150
122509	T0302	SED12MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 1 station 2	Boeing TC(k) Sys	degF	551	50	800
41087	P0606	SED13MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 1 station 3	Boeing Hyscan Pres Sys	psia	125	14.7	150
122509	T0303	SED13MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 1 station 3	Boeing TC(k) Sys	degF	551	50	800
		SED21MDj		Exit rake 2 angular position	Independent	deg	-84	90	-90

41087	P0613	SED21MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure exit rake 0 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0607	SED21MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 2 station 1	Boeing Hyscan Pres Sys	psia	127	14.7	150
122509	T0304	SED21MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 2 station 1	Boeing TC(k) Sys	degF	551	50	800
41087	P0614	SED22MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure exit rake 90 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0608	SED22MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 2 station 2	Boeing Hyscan Pres Sys	psia	127	14.7	150
122509	T0305	SED22MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 2 station 2	Boeing TC(k) Sys	degF	551	50	800
41087	P0615	SED23MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure exit rake 180 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0609	SED23MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 2 station 3	Boeing Hyscan Pres Sys	psia	127	14.7	150
122509	T0306	SED23MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 2 station 3	Boeing TC(k) Sys	degF	551	50	800
41087	P0616	SED24MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure exit rake 270 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
		SED31MDj		Exit rake 3 angular position	Independent				
41087	P0610	SED31MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 3 station 1	Boeing Hyscan Pres Sys	psia	127	14.7	150
122509	T0307	SED31MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 3 station 1	Boeing TC(k) Sys	degF	551	50	800
41087	P0611	SED32MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 3 station 2	Boeing Hyscan Pres Sys	psia	127	14.7	150
122509	T0308	SED32MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 3 station 2	Boeing TC(k) Sys	degF	551	50	800
41087	P0612	SED33MPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure rake 3 station 3	Boeing Hyscan Pres Sys	psia	127	14.7	150
122509	T0309	SED33MTt	Ram 2 exhaust rake, dual element, three immersion, Ram 2	Post inlet diffuser exh duct stagnation temp rake 3 station 3	Boeing TC(k) Sys	degF	551	50	800
40311		SED41MPs	Pressure transducer, 0-200 psig, 4-20 mA, general purpose, 1/8 NPT	Stationary exh duct hub static pressure 1/2 between rakes/ collector 0 deg.	Ramgen AB PLC	psia	125	14.7	150
		SED41MPt		Tip ring exit total pressure rake, hub station	Boeing Hyscan Pres Sys	psia	120	15	215
41087	P0706	SED42MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure 1/2 between rakes/collector 90 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
		SED42MPt		Tip ring exit total pressure rake, middle station	Boeing Hyscan Pres Sys	psia	120	15	215
41087	P0707	SED43MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure 1/2 between rakes/collector 180 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
		SED43MPt		Tip ring exit total pressure rake, tip station	Boeing Hyscan Pres Sys	psia	120	15	215
41087	P0708	SED44MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure 1/2 between rakes/collector 270 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0709	SED61MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure collector entrance plane 0 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0710	SED62MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure collector entrance plane 90 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150

41087	P0711	SED63MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure collector entrance plane 180 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0712	SED64MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Stationary exh duct hub static pressure collector entrance plane 270 deg.	Boeing Hyscan Pres Sys	psia	125	14.7	150
41087	P0801	SED7CMPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure survey performance probe pressure sensor	Boeing Hyscan Pres Sys	psia	127	14.7	150
122810	T0310	SED7CMTt	Ram 2 dual element performance probe with nulling, Ram 2	Post inlet diffuser exh duct stagnation temperature dual element probe temp sensor	Boeing TC(k) Sys	degR	551	0	800
41087	P0802	SED7LMPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure survey performance probe left nulling pressure (ALF)	Boeing Hyscan Pres Sys	psia	127	14.7	150
41087	P0803	SED7RMPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure survey performance probe right nulling pressure (ALF)	Boeing Hyscan Pres Sys	psia	127	14.7	150
41087	P0713	SED8CMPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure survey single cobra probe center probe	Boeing Hyscan Pres Sys	psia	127	14.7	150
41087	P0715	SED8LMPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure survey single cobra probe left probe	Boeing Hyscan Pres Sys	psia	127	14.7	150
41087	P0714	SED8RMPt	Scanivalve Hyscan 200 psid module, atmospheric reference	Post inlet diffuser exh duct stagnation pressure survey single cobra probe right probe	Boeing Hyscan Pres Sys	psia	127	14.7	150
41087	P0804	SEDC1MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure closest actuated position (i.e. variable axial dist from rotor) location 3	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0805	SEDC2MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure closest actuated position (i.e. variable axial dist from rotor) location 4	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0806	SEDC3MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure closest actuated position (i.e. variable axial dist from rotor) location 2	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0807	SEDC4MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure closest actuated position (i.e. variable axial dist from rotor) location 1	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0808	SEDF1MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure farthest actuated position (i.e. variable axial dist from rotor) location 1	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0809	SEDF2MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure farthest actuated position (i.e. variable axial dist from rotor) location 2	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0810	SEDF3MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure farthest actuated position (i.e. variable axial dist from rotor) location 3	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0811	SEDF4MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure farthest actuated position (i.e. variable axial dist from rotor) location 4	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0812	SEDM1MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure middle actuated position (i.e. variable axial dist from rotor) location 1	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0813	SEDM2MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure middle actuated position (i.e. variable axial dist from rotor) location 2	Boeing Hyscan Pres Sys	psia	150	15	250

41087	P0814	SEDM3MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure middle actuated position (i.e. variable axial dist from rotor) location 3	Boeing Hyscan Pres Sys	psia	150	15	250
41087	P0815	SEDM4MPs	Scanivalve Hyscan 200 psid module, atmospheric reference	Exit survey tip/shroud static pressure middle actuated position (i.e. variable axial dist from rotor) location 4	Boeing Hyscan Pres Sys	psia	150	15	250
		SEL01MPs		Bellmouth inflatable seal pressure	Ramgen AB PLC	psig	40	AS10xMPs+ 10	45
	TAMB	TAMB		Cell ambient temperature	Boeing TC(k) Sys	degR	70	40	95
	T0601	TBDP		Primary air system burst disk temperature	Boeing TC(k) Sys	degR			
	TCHK1	TCHK-1		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
	TCHK2	TCHK-2		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
	TCHK3	TCHK3		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
	TCHK4	TCHK4		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
	TCHK5	TCHK5		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
	TCHK6	TCHK6		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
	TCHK7	TCHK7		Reference Oven Temperature Check	Boeing TC(k) Sys	degR			
40033		TIP01MAr	Velocimeter, 0-1 ips, 1/4-28 mounting stud	Axial tip ring radial vibration	Ramgen AB PLC	ips	0.1	0	0.3
		TIP01MFz		Tip ring right actuator thrust	Ramgen AB PLC	lbf	1250	0	2000
		TIP01MTs		Tip ring coolant supply temperature	Ramgen AB PLC	degF	TBD	TBD	TBD
		TIP02MFz		Tip ring left actuator thrust	Ramgen AB PLC	lbf	1250	0	2000
		TIP02MTs		Tip ring coolant exhaust temperature	Ramgen AB PLC	degF	TBD	TBD	TBD
40731		TIP11MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap forward probe position location 1	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0401	TIP11MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 1 (most upstream) circ. pos 1	Boeing TC(k) Sys	degF	50	-50	250
40731		TIP12MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap forward probe position location 2	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0402	TIP12MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 1 (most upstream) circ. pos 2	Boeing TC(k) Sys	degF	50	-50	250
40731		TIP13MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap forward probe position location 3	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0403	TIP13MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 1 (most upstream) circ. pos 3	Boeing TC(k) Sys	degF	50	-50	250
40731		TIP14MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap forward probe position location 4	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0404	TIP14MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 1 (most upstream) circ. pos 4	Boeing TC(k) Sys	degF	50	-50	250
40731		TIP21MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap aft probe position location 1	Boeing Oscilloscope	mils	0.0015	0.000	0.015

40742	T0405	TIP21MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 2 (2nd most upstream) circ. pos 1	Boeing TC(k) Sys	degF		0	700
40731		TIP22MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap aft probe position location 2	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0406	TIP22MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 2 (2nd most upstream) circ. pos 2	Boeing TC(k) Sys	degF		0	700
40731		TIP23MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap aft probe position location 3	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0407	TIP23MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 2 (2nd most upstream) circ. pos 3	Boeing TC(k) Sys	degF		0	700
40731		TIP24MDr	Clearance probe, high temp turbine, style, 0.196 body, 2 in. 1/4-28 threaded body, Inconel/Hastelloy, -110 to 1600 degF operating range	Strake tip gap aft probe position location 4	Boeing Oscilloscope	mils	0.0015	0.000	0.015
40742	T0408	TIP24MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature axial location 2 (2nd most upstream) circ. pos 4	Boeing TC(k) Sys	degF		0	700
40742	T0409	TIP31MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 3 (2nd most downstream) width 1	Boeing TC(k) Sys	degF		0	700
40742	T0410	TIP32MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 3 (2nd most downstream) width 2	Boeing TC(k) Sys	degF		0	700
40742	T0411	TIP33MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 3 (2nd most downstream) width 3	Boeing TC(k) Sys	degF		0	700
40742	T0412	TIP34MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 3 (2nd most downstream) width 4	Boeing TC(k) Sys	degF		0	700
	ACT1	TIP41MCv		Tip clearance shroud actuator 1 commanded position	Parker actuator control	mA	4	4	20
40742	T0413	TIP41MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 4 (most downstream) width 1	Boeing TC(k) Sys	degF		0	700
	ACT2	TIP42MCv		Tip clearance shroud actuator 2 commanded position	Parker actuator control	mA	4	4	20
40742	T0414	TIP42MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 4 (most downstream) width 2	Boeing TC(k) Sys	degF		0	700
40742	T0415	TIP43MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 4 (most downstream) width 3	Boeing TC(k) Sys	degF		0	700
40742	T0416	TIP44MTm	Thermocouple, k-type, 0.062 dia., ungrounded junction, inconel sheath, 425 degF mini connector, 24 in. length	Tip shroud metal temperature location 4 (most downstream) width 4	Boeing TC(k) Sys	degF		0	700
		TIP51MCv		Tip clearance shroud actuator 1 commanded travel rate	Parker actuator control	mA	4	4	20
		TIP52MCv		Tip clearance shroud actuator 2 commanded travel rate	Parker actuator control	mA	4	4	20

40527	T0117	TIP81MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Tip ring flange metal temperature	Boeing TC(k) Sys	degF	80	40	100
40527	T0118	TIP82MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Tip ring flange metal temperature	Boeing TC(k) Sys	degF	80	40	100
40527	T0119	TIP83MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Tip ring flange metal temperature	Boeing TC(k) Sys	degF	80	40	100
40527		TIP91MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Tip outer guide metal temperature chiller input	Independent	degF	80	40	85
		TIP91MTm		Tip ring guide metal temperature	Ramgen AB PLC	degF	80	40	85
40527	T0417	TIP92MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Tip outer guide metal temperature	Boeing TC(k) Sys	degF	80	40	85
40527	T0418	TIP93MTm	Thermocouple, k-type, 0.062 dia, ungrounded junction, 304 stnl sheath, 425 degF mini connector, 48 in. length	Tip outer guide metal temperature	Boeing TC(k) Sys	degF	80	40	85
		TST01MO		imaging borescope view of the exhaust	Independent				
6935		TST01MPs	Pressure transducer, 30 inMercury to 15 psig, 4-20 mA, general purpose, 1/4 NPT	Transducer rack absolute pressure	Ramgen AB PLC	psia	14.7	14.3	15.1
		TST02MO		imaging external-camera viewing exhaust	Independent				
	TTC1	TTC1		Station 1 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC10	TTC10		Station 10 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC11	TTC11		Station 11 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC12	TTC12		Station 12 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC13	TTC13		Station 13 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC14	TTC14		Station 14 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC15	TTC15		Station 15 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC16	TTC16		Station 16 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC2	TTC2		Station 2 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC3	TTC3		Station 3 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC4	TTC4		Station 4 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC5	TTC5		Station 5 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC6	TTC6		Station 6 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC7	TTC7		Station 7 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC8	TTC8		Station 8 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120
	TTC9	TTC9		Station 9 of 16 station rig inlet temperature rake	Boeing TC(k) Sys	degF	70	0	120

41089	P0514	VAC01MPs	Scanivalve Hyscan 30 psid module, vacuum reference	Vacuum system reservoir pressure	Boeing Hyscan Pres Sys	psia	5	0	14.7
	TVAC	VAC01MTs		Facility vacuum plenum temperature	Boeing TC(k) Sys	degF	150	50	250
41089	PVAC	VAC02MPs	Scanivalve Hyscan 30 psid module, vacuum reference	Vacuum system reservoir pressure	Boeing PLC	psia	5	0	14.7
	WACPE	WAE01CW0		Aft wheel space cavity exhaust flow rate	Boeing PLC	lbm/sec	0.01	0	0.1
		WAE02MCv		Aft bearing air purge flow control valve position signal 4-20 control signal from 1 channel of an Allen Bradley board	Ramgen AB PLC	mA		4	20
40013	P1001	WAE11MPd	Scanivalve Hyscan 100 psid module, atmospheric reference	Aft wheel space purge discharge air venturi downstream pressure	Boeing Hyscan Pres Sys	CFM	17	14.7	100
40013	P1002	WAE11MPf	Scanivalve Hyscan 100 psid module, atmospheric reference	Aft wheel space purge flow meter upstream pressure discharge air	Boeing Hyscan Pres Sys	psia	40	14.7	100
		WAE11MPs		Regulator dome pressure, aft wheelspace back pressure regulator	Ramgen AB PLC	psia	70	20	100
	T0201	WAE11MTf		Aft wheel space purge discharge air flow meter temperature	Boeing TC(k) Sys	degF	150	50	350
	DPACE	WAE21MPd		Aft wheel space purge discharge air venturi differential pressure	Boeing PLC	psid	3	0	5
	WACPS	WAS01CWf		Aft wheel space supply flow rate	Boeing PLC				
		WAS01MCv		Aft wheel space cavity back pressure valve commanded valve position signal 4-20 mA signal from 1 channel of an Allen-Bradley control board	Ramgen AB PLC	mA		4	20
40087	P1004	WAS01MPf	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	upstream flow meter pressure for the aft wheel space purge air intake line	Boeing Hyscan Pres Sys	psia	100	14.7	125
	T0202	WAS01MTf		Aft wheel space intake air flowmeter temperature 1/16 stnl sheath K-type with standard low noise connector	Boeing TC(k) Sys	degF	150	50	350
40013	P1003	WAS11MPd	Scanivalve Hyscan 100 psid module, atmospheric reference	Aft wheel space intake air flow meter downstream pressure	Boeing Hyscan Pres Sys	CFM	20	14.7	35
	DPWAS1	WAS21MPd		Aft wheel space intake air flow meter differential pressure	Boeing PLC	psid	3	0	5
	WFCPE	WFE01CW0		Forward wheel space exhaust flow rate	Boeing PLC	lbm/sec	0.071	0	0.25
40013	P1009	WFE01MPf	Scanivalve Hyscan 100 psid module, atmospheric reference	Forward wheel space purge flow meter upstream pressure discharge air	Boeing Hyscan Pres Sys	psia	20	14.7	50
	T0203	WFE01MTf		Forward wheel space purge discharge air flow meter temperature	Boeing TC(k) Sys	degF	150	0	260
		WFE02MCv		Forward wheel space cavity back pressure valve commanded valve position signal 4-20 mA signal from 1 channel of an Allen-Bradley control board	Ramgen AB PLC	mA		4	20
40013	P1008	WFE11MPd	Scanivalve Hyscan 100 psid module, atmospheric reference	Forward wheel space purge discharge air venturi differential or downstream pressure	Boeing Hyscan Pres Sys	psia	17	14.7	50
	DPFCE	WFE21MPd		Forward wheel space cavity exhaust nozzle differential pressure	Boeing PLC	psid	3	0	5
	WFCPS	WFS01CWf		Forward wheel space supply flow rate	Boeing PLC	lbm/sec	0.05	0	0.25
		WFS01MCv		Forward wheel space cavity back pressure valve commanded valve position signal 4-20 mA signal from 1 channel of an Allen-Bradley control board	Ramgen AB PLC	mA		4	20
40087	P1011	WFS01MPf	Amplifier, proximity probe, 4 sensors can be ganged to reduce amplifier count	Forward wheel space purge intake air flow meter upstream pressure	Boeing Hyscan Pres Sys	psia	90	14.7	115

	T0204	WFS01MTf		Forward wheel space purge intake air flow meter temperature	Boeing TC(k) Sys	degF	50	0	120
40013	P1010	WFS11MPd	Scanivalve Hyscan 100 psid module, atmospheric reference	Forward wheel space purge intake air flow meter downstream pressure	Boeing Hyscan Pres Sys	psia	25	14.7	50
	DPWFS1	WFS21MPd		Forward wheel space purge intake air flow meter differential pressure	Boeing PLC	psid	3	0	5

**Pages 3A-20 thru 3A-60 intentionally left blank under the provisions of the
Energy Policy Act of 1992**

Technical Report

Static Stress and Deflection Analysis of the Rampressor Test Rig

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Introduction

The purpose of this analysis was the simulation of the physical behavior of the Rampressor test rig when it is subjected to a large imbalance load. This analysis is preliminary in nature because it uses a static load as a method of estimating the worthiness of the test rig when a dynamic load is applied to it. It is understood by Ramgen Power Systems Inc. and Optimal Analysis Inc. that the true behavior of the system can only be determined by doing a dynamic analysis. It was decided by Ramgen Power Systems Inc. that a static analysis would be sufficient for their needs, even though a dynamic analysis would more accurately simulate the true behavior of the test rig.

Discussion

The Rampressor test hardware is shown in Figure 1. It consists of a massive frame that has the test rig bolted to it. The frame is assumed to be very stiff when compared to the test rig, and can be replaced with rigid boundary conditions where the test rig bolts to the frame. The relevant components for the analysis are shown in Figure 2. The finite element model omitted some of these components and is shown in Figure 3.

While the behavior of the test rig to rotating imbalance is a dynamic phenomena, a static analysis was used as a preliminary way of gaining insight into the worthiness of the

Rampressor test rig. The limitations of an analysis like this should be kept in mind when using the results of this analysis. A dynamic analysis would take into account the amplification that occurs when a dynamic load is applied with a frequency that is in synchronization with a natural frequency of the test rig (and the effects of damping), while the static analysis does not.

Figure 1: Cutaway view of the entire Rampressor test rig.

Figure 2: Cutaway view of the analyzed test rig structure.

Figure 3: Finite element model used in the analysis of the Rampressor test rig.

Analysis Approach

The test rig is a complicated structure that has several components that are bolted to one another and slide along three shafts. Because this analysis is preliminary, the bolts were omitted from the model and the bolted sections were bonded together.

The problem as it was analyzed is linear, which means that the stresses and deflections are linear with respect to the forces that are applied to the structure. This is illustrated in Appendix V where two plots show the stresses that result when loads of 20,000 lbs and 40,000 lbs are applied to each bearing support.

Five static analyses were done, each at different rotational positions around the bearing support surfaces on the front and aft core rig frames. Using the coordinate system shown in Figure 3 as a guide, the angles at which the loads were applied were straight up (+X direction, hereafter referred to as “0 degrees”), straight down (-X direction, hereafter referred to as “180 degrees”), to the left (+Y direction, hereafter referred to as 90 degrees), and halfway between each of these (hereafter referred to as “45 degrees” and “135 degrees”). The load used in this analysis was 40,000 lbs applied to each bearing support, for a total of 80,000 lbs applied to the test rig during each analysis.

Operating temperature and reference temperature were both 600F. This was done to include the lower elastic modulus that accompanies this temperature and avoids faulty

thermal stresses due to the rigid mounts. A bigger and more complex model would be needed to include thermal deflections.

Due to the complex geometry of the test rig and the number of nodes that can be analyzed, a combination of element types were used in the analysis. Linear SOLID45 elements were used, along with higher order (SOLID92 and SOLID95) solid elements. Mapped and sweep meshing were used whenever possible to keep the number of nodes to a size that could be analyzed in a timely manner. The model had 281818 elements and 412295 nodes.

Material properties were provide by Ramgen Power Systems and are listed in Appendix I. The same material property was used for all components in the analysis.

A macro was written to apply pressure loads and run the solutions. Runit2.mac can be found in Appendix II. The user can edit the parameters at the top of the macro and vary the amount of bearing load that will be applied. Another macro, runpost.mac, was used to postprocess the results of the analysis and can be found in Appendix III. It generated plots and tabulated the loads across the bolted cross sections that were bonded together and listed these loads in a file (forceandmoment.txt). The output file can be found in Appendix IV.

The core rig frames slide along the three bars. This was included in the simulation by generating coincident nodes where the rig frame bearings contact the bars, rotating these nodes and the corresponding nodes on the bars into a cylindrical coordinates system that was placed along the axis of each bar, and coupling the rotated nodes in the radial (X) direction.

Rather than applying forces to the bearing surfaces inside the front and aft core rig frames, pressures were used in an attempt to minimize unrealistic stresses that would result from applying forces to a small number of forces. The amount of pressure that was needed to obtain the desired force at each angular position was determined by running separate analysis on the front and rear core rig frames.

Results

Depending on the options chosen when using runpost.mac to postprocess the results, a minimum of 190 plots can be generated for the 5 analyses. All plots are listed in Appendix VI. Copies of the plots are also included on the CDs that accompany this report, along with the ANSYS model and the macros that run the analysis and do the postprocessing.

The results for each analysis are listed in sections below that include a table of result data and several plots that are particularly relevant. Von Mises stress is listed in the tables, along with the maximum deflection for each component. The deflection data is of limited value because it does not include effects of thermal expansion. Both averaged

and unaveraged von Mises stresses are listed. Unaveraged von Mises stresses tend to be more accurate and more conservative, and should be used in most instances.

Due to unrealistic stress values in the immediate vicinity where pressure loads were applied to bearing support surfaces, the postprocessing macro unselects the first two layers of elements in this region. Stresses on the bearing support surfaces are unrealistic and should be ignored. If accurate answers are desired in this region it will be necessary to build a more detailed model that has the bearing races and contact elements between them and the bearing support surfaces.

The last column in each table includes comments regarding some of the analysis results. Due to the simplified geometry used in the model and other fundamental limitations (absence of actual bolted joints, etc.) some of the stresses that result must be interpreted with caution. The comments column contains observations that are of particular significance.

Load at Zero Degrees

Table 1 contains the results of the analysis when the bearing loads are at zero degrees, or straight up parallel with the global X axis that is shown in Figure 3. In this case, the loads are applied to the bearing support surfaces in the positive X direction.

Load On Each Bearing	Angle	Component	Max Displacement	Max Von Mises Stress (Averaged)	Max Von Mises Stress (Unaveraged)	Comments
40000	0	Aft Core Rig Frame	0.197645	33910	44840	<---This stress likely to be 80,000 when Kt is 1.0. To obtain actual stress in this location, an appropriate Kt for this geometry must be multiplied by 80,000 PSI.
		Aft End Frame	0.168142	52040	55115	
		Bearings	0.219324	64280	68952	
		Front Core Rig Frame	0.222454	108338	127977	
		Struts Only	0.18604	108338	127977	
		Front End Frame	0.157902	35303	44824	
		Mount Plates	0.001208	22976	24299	
		Mounts	0.00074	61676	73588	
		Rods	0.211635	92267	103154	
		Spacers	0.19377	60138	71852	

Table 1: Stresses and deflections for zero degree loads.

Examples of the results of the analysis are shown in Figures 4-6 below. All plots can be found in Appendix VI.

Figure 4: Total displacement of the model.

Figure 5: Averaged von Mises stress.

Figure 6: Unaveraged von Mises stress.

Comparing averaged and unaveraged results is often useful for determining if the mesh is adequately refined. Most of the results for this analysis show a difference of 6-12%, with the exception of the stress in the front core rig frame struts. Closer examination of the stresses in this area revealed that the stress in this region is likely to be closer to 80,000 PSI when the stress concentration factor $K_t = 1.0$. To obtain the stresses in this region it is necessary to obtain a value for K_t and multiply this time 80,000 PSI, or add geometry to the finite element model that allows for an accurate evaluation of stresses in this region.

Figure 7 shows the stress results in the region around the struts.

Figure 7: Piece of aft core rig frame show maximum stress.

Figure 8 shows a closer view of the high stress region.

Figure 8: Closer view of high stress region.

The high stress is on the edge of the strut. The lack of a fillet in this region makes it impossible to determine what the actual stress is, but a method for approximating it can be employed. By making a plot of the stress along the edge of the strut we can see how the singular regions on each end of the strut cause unrealistically high stresses. Figure 9 shows the path that stress data was plotted over.

Figure 9: Path that stress data was plotted over.

Figure 10 shows the graphed stress data. It is easy to see the spikes on each side of the curve that are caused by the singularities in the sharp corners where the strut meets the inner hub..

Figure 10: Graph of von Mises stress along the edge of the strut. The left side of the curve is the top of the strut, the right side of the curve is the bottom.

If we were to remove the spikes from the curve and extrapolate where the curve would cross the end of the graph we can get an idea what the stress is at that location when $K_t=1.0$. For the right side of the curve in Figure 10, we can see that it would cross around 80,000-85,000 if the singularity was not present. If we knew the concentration factor for the fillet geometry we would be able to get a reasonably good estimate of what the stress is in this location.

Load at 45 Degrees

When the load is applied at a 45 degree angle, the stresses change in different components as shown below in Table 2.

Load On Each Bearing	Angle	Component	Max Displacement	Max Von Mises Stress (Averaged)	Max Von Mises Stress (Unaveraged)	Comments
40000	45	Aft Core Rig Frame	0.158037	62467	125510	<--- This high unaveraged stress is due to the bonded connection at the flanges and can be ignored. Max stress in the core rig frame is likely to be close to 65,000 PSI.
		Aft End Frame	0.12838	75183	79728	
		Bearings	0.175168	50030	53363	
		Front Core Rig Frame	0.177481	78930	88016	
		Struts Only	0.150854	78930	88016	
		Front End Frame	0.113474	30872	37826	
		Mount Plates	0.001844	27324	30480	
		Mounts	0.001136	90829	104256	
		Rods	0.169988	72047	81863	
		Spacers	0.152216	91782	103182	

Table 2: Stresses and deflections for 45 degree loads.

Of particular interest is the wide discrepancy between the averaged and unaveraged stresses for the aft core rig frame. This is due to how the aft core rig frame is bonded to the aft end frame. A step in the geometry is present at this location, and the element quality is poor. A closer look at this region can be seen in Figures 11 and 12.

Figure 11: View of the aft core rig frame showing the side that is bonded to the aft end frame. The high stress region is in the lower left.

Figure 12: A close-up of the high stress region. It is obvious that the high stress region is due to two factors. One is the poor aspect ratio of the element in this spot, and the other is the fact that a spacer is bonded to the aft core rig frame in this location and is causing a singularity due to the stepped nature of the geometry.

Stresses in other regions of the model are valid and the unaveraged von Mises stress can be considered to be accurate.

Load at 90 Degrees

The results from the 90 degree case are listed below in Table 3.

Load On Each Bearing	Angle	Component	Max Displacement	Max Von Mises Stress (Averaged)	Max Von Mises Stress (Unaveraged)	Comments
40000	90	Aft Core Rig Frame	0.091379	69073	139903	<--- 70,000 is more likely due to singularity in this location. Identical situation as was seen in the results from 45 degrees. Maximum stress is likely to be 65,000 PSI.
		Aft End Frame	0.057691	59366	65010	
		Bearings	0.115102	31322	33061	
		Front Core Rig Frame	0.11614	93625	107529	
		Struts Only	0.104294	93625	103629	
		Front End Frame	0.027047	19184	24293	
		Mount Plates	0.001609	20873	20873	
		Mounts	0.001143	96068	111013	
		Rods	0.11341	48361	55723	
		Spacers	0.091089	81265	92003	

Table 3: Stresses and deflections for 90 degree loads.

The maximum stress in the aft core rig frame is flawed in the same way that was seen in the previous case where the angle was 45 degrees. As was seen in that case, the high stress is due to the spacer that is bonded to the aft core rig frame and tie it to the aft rig frame. The maximum stress is like to be in the 65,000 PSI zone rather than 139,903 PSI.

Load at 135 Degrees

The results from the 135 degree analysis are listed below in Table 4.

Load On Each Bearing	Angle	Component	Max Displacement	Max Von Mises Stress (Averaged)	Max Von Mises Stress (Unaveraged)	Comments
40000	135	Aft Core Rig Frame	0.157981	62946	125985	<--- Could be 130,000 when Kt=1.0. Need better geometry to get an accurate answer. Likely to be highest stress location in the model.
		Aft End Frame	0.128294	75178	79721	
		Bearings	0.17516	49908	53224	
		Front Core Rig Frame	0.177477	113473	126565	
		Struts Only	0.11347	113473	126565	
		Front End Frame	0.113455	30871	37817	
		Mount Plates	0.001844	27328	30483	
		Mounts	0.001137	90844	104272	
		Rods	0.169965	72040	81862	
		Spacers	0.152165	91868	103259	

Table 4: Stresses and deflections for 135 degree loads.

The stresses in the front core rig frame are very high in the region where one of the struts meets the inside hub. This is a sharp inside corner and any results obtained in this region

are not accurate. Using the same methodology that was employed in looking at the zero degree case, von Mises stress along the edge of the strut was plotted and an estimate was made of what the maximum stress might be if the stress concentration factor K_t was zero. Figures 13-17 show the area where maximum stress occurs and the graph of stress along the edge of the strut.

Figure 13: The location of maximum von Mises stress is indicated by the MX characters. They can be seen at the inner edge of the strut that is at the 4 o'clock position.

Figure 14: A close-up of the strut where maximum von Mises stress occurs.

Figure 15: The strut elements and adjacent elements were selected for further examination.

Figure 16: The location of maximum von Mises stress is at the upper edge of the strut.

Figure 17: Plotting von Mises stress along the edge of the strut shows that the averaged von Mises stress might be as high as 105,000 PSI with $K_t = 1.0$. Unaveraged von Mises stress could be in the 130,000 PSI range, given the mesh density of this model.

Load at 180 Degrees

The last load case had the load at 180 degrees, or downward in the negative X direction. The results are listed in Table 5.

Load On Each Bearing	Angle	Component	Max Displacement	Max Von Mises Stress (Averaged)	Max Von Mises Stress (Unaveraged)	Comments
40000	180	Aft Core Rig Frame	0.197637	124082	141157	<--- Located at oil hole. Likely to be less than this due to the distributed nature of the how the bearing race spreads out the load. Needs more detailed analysis to know what the stress is in this location.
		Aft End Frame	0.168152	52075	55150	
		Bearings	0.219297	64187	68842	
		Front Core Rig Frame	0.222425	75606	89025	
		Struts Only	0.18606	75606	89025	
		Front End Frame	0.157912	35325	44855	
		Mount Plates	0.001209	23000	24323	
		Mounts	0.00074	61707	73619	
		Rods	0.211609	92230	103113	
		Spacers	0.193765	60359	72093	

Table 5: Stresses and deflections for 180 degree loads.

The maximum von Mises stress occurs in an oil passage just below the bearing support in the aft core rig frame. This is shown below in Figure 18.

Figure 18: A view of the aft core rig frame that shows the maximum stress is located in an oil passageway just below the bearing support surface where the pressure load is applied. The pressure load is shown by the red arrows.

Figure 19: A closer view of the oil passage hole.

The high von Mises stresses are due to the presence of the bearing above the oil passage, but it's impossible to know whether this will be the stress on the actual part because the bearing race will distribute the load differently than the pressure load that was used in this model. It can be seen in Figure 19 that the high stress zone in the hole is significantly influence by the pressure load. Due to the geometry and how the aft core rig frame was meshed, pressure could only be placed on the section of the race support that is shown in Figure 19.

It is impossible to know for sure what the stresses will be in the region of this whole without doing a more detailed analysis that includes the bearing race and contact elements between the race and the aft core rig frame.

Bolt Loads – A Conservative Way of Sizing Bolts

The bolt loads can only be approximated because bolts were not included in the model. A method was used to obtain a reasonable and conservative estimate of maximum bolt loads for the static analyses that were performed in this study. The forces and moments acting on each bolted section were obtained for each analysis. There are three bolted sections where the core frame rigs are joined and three more where the aft core rig frame joins the aft rig frame.

The terminology used to identify the joints deserves some explanation. The three joints that are between the two core rig frames are identified by the label “Core”. Their location is defined by “Y Low Position”, “Y Mid Position”, and “Y High Position”. Low, mid, and high refer to the Y coordinate for this location as defined by the coordinate system that is shown in Figure 3. Therefore, the location that is defined by “Core High Y Position” would be the joint that is on the interface between the two core frames and has the highest (most positive) Y location. In Figure 3, this would be the position on the far left. The “Core Mid Y Position” location would be the middle one as seen in Figure 3, and the “Core Low Y Position” would be the joint that is located on the right side of Figure 3. Similarly, the bolted connections between the aft core rig frame and the aft end frame are identified by “End Low Y Position”, “End Mid Y Position”, and “End High Y Position”.

There are three forces and three moments at each joint. One method for sizing bolts would be to calculate the SRSS (square root of the sum of the squares) of the moments that are in plane and add this value to the axial force that is acting on the joint. The information in Appendix IV lists the forces and moments that are acting on each joint. It is assumed that a single bolt is used in the joint and is located 2 inches away from the rod axis. (The radius can be changed by editing the parameter “boltrad” in the postprocessing macro and running the macro again.)

If we assume that the entire load is carried by one bolt, the maximum bolt load is 61919 lbs (End Frame Y Position, 135 degree load). Assuming we use a grade 8 bolt with a

yield strength of 130,000 PSI, the bolt would need to have an area of $61919/130000=.476$ square inches. This is approximately the area that is available with a 7/8 inch diameter bolt. It is recommended that 4 of these bolts be used at each connection at 90 degree increments around the rod.

Conclusions

The maximum stress that can be expected in the test rig is located where the struts meet the inner hub and can be as high as 130,000 PSI multiplied by the stress concentration factor for this geometry. Since the geometry was not included in the model data that was provided by Ramgen, it is impossible to accurately define this any further.

If $K_t = 2.0$, the stress can be as high as 260,000 PSI. As high as this might appear, the situation is not really that severe because the strut has relatively low stresses elsewhere (see Figure 16) and when yielding occurs the rest of the strut will start to pick up the load. The real issue is probably not a stress problem as much as it is a strain problem. When plastic strain exceeds the allowable limit for the material being used, fracture will occur and at this point the problem becomes crack propagation oriented. It is common for turbine engine manufacturers to use averaging techniques when evaluating the suitability of a blade or strut. If we were to use this method for this problem the average stress in the strut would likely be 20,000 PSI times K_t .

To fully understand the Rampressor test rig we would need to do a more accurate analysis of the bolted joints, the interaction of the bearing races and the core rig frames, and a dynamic analysis that more accurately represents the truly dynamic nature of the loads that the test rig might be exposed to. If stresses go beyond yield, a plastic analysis could be done using the deformed geometry from the dynamic analysis and cycled several times to see what residual stress occurs and whether plastic deformation continues to take place after the first cycle.

APPENDICES

APPENDIX I

The material properties used in the analysis are listed below. Different material tables were used for each component, each of which used the same properties.

```
PROPERTY TABLE EX      MAT=          1  NUM. POINTS= 10
  TEMPERATURE      DATA  TEMPERATURE      DATA  TEMPERATURE      DATA
    70.000      0.28444E+08  200.00      0.27890E+08  300.00
0.27440E+08
    400.00      0.26980E+08  500.00      0.26520E+08  600.00
0.26060E+08
    700.00      0.25600E+08  800.00      0.25160E+08  900.00
0.24720E+08
    1000.0      0.24299E+08
```

```
PROPERTY TABLE NUXY  MAT=          1  NUM. POINTS= 10
  TEMPERATURE      DATA  TEMPERATURE      DATA  TEMPERATURE      DATA
    70.000      0.27191      200.00      0.27572      300.00      0.27979
    400.00      0.28469      500.00      0.29036      600.00      0.29675
    700.00      0.30386      800.00      0.31169      900.00      0.32025
    1000.0      0.32955
```

```
PROPERTY TABLE ALPX  MAT=          1  NUM. POINTS= 9  REFERENCE TEMP. =
600.00
  TEMPERATURE      DATA  TEMPERATURE      DATA  TEMPERATURE      DATA
    200.00      0.61000E-05  300.00      0.62300E-05  400.00
0.63000E-05
    500.00      0.63500E-05  600.00      0.64001E-05  700.00
0.64802E-05
    800.00      0.66003E-05  900.00      0.67710E-05  1000.0
0.70006E-05
```

```
PROPERTY TABLE DENS  MAT=          1  NUM. POINTS= 1
  TEMPERATURE      DATA  TEMPERATURE      DATA  TEMPERATURE      DATA
    0.0000      0.73240E-03
```

APPENDIX II

The macro runit2.mac that was used to run the analysis is listed below. It resumed a base model and applied boundary conditions and then ran the analysis. Files like this are very useful when running multiple load step solutions.

```
!  
!  runit2.mac  a macro that runs the test rig analysis model  
!              for a given load at  
!              angles of 0, 45, 90, 135, and 180 degrees.  
!  
!  resume the model  
resume,model20,db  
!  
!  input force load that will be applied to each bearing  
force=40000  
!  
!  assign pressures for each position  
pzerofront=10594.5146*(force/10000)  
p45front=6966.57298*(force/10000)  
p90front=10594.5177*(force/10000)  
p135front=13091.1762*(force/10000)  
p180front=8014.962326*(force/10000)  
!  
pzeroaft=14951.53386*(force/10000)  
p45aft=9830.297703*(force/10000)  
p90aft=72500.01813*(force/10000)  
p135aft=36388.49963*(force/10000)  
p180aft=14951.53609*(force/10000)  
!  
/solu  
!  
!eqslv,pcg,1e-6  
!  the sparse solver seems to be quicker for this model  
eqslv,sparse  
!  
!  we want to apply a pressure that gives us the desired load  
!  in 5 directions on each bearing.  
!  
!  for zero degrees we'll use areas 979 981 2328 2445 for the front  
!  bearing.  
!  
!  for 45 degrees we'll use areas 972 973 976 977 979 for the front  
!  bearing.  
!  
!  for 90 degrees we'll use areas 965 967 969 972 for the front  
!  bearing.  
!  
!  for 135 degrees we'll use areas 876 961 963 for the front  
!  bearing.  
!  
!  for 180 degrees we'll use areas 876 983 2260 2376 for the front  
!  bearing.  
!  
!
```

```

! delete all pressures
sfade,all,all,pres
sfede,all,all,pres
!
! first do the zero degree position
sfa,979,1,pres,pzerofront
sfa,981,1,pres,pzerofront
sfa,2328,1,pres,pzerofront
sfa,2445,1,pres,pzerofront
!
!
sfa,1073,1,pres,pzeroaft
sfa,1096,1,pres,pzeroaft
sfa,2779,1,pres,pzeroaft
sfa,2786,1,pres,pzeroaft
!
/title,Zero Degrees - Force = %force% Per Bearing
solve
!
! delete all pressures
sfade,all,all,pres
sfede,all,all,pres
!
!
! do the 45 degree position
sfa,972,1,pres,p45front
sfa,973,1,pres,p45front
sfa,976,1,pres,p45front
sfa,977,1,pres,p45front
sfa,979,1,pres,p45front
!
sfa,849,1,pres,p45aft
sfa,1078,1,pres,p45aft
sfa,1096,1,pres,p45aft
!
/title,45 Degrees - Force = %force%
solve
!
! delete all pressures
sfade,all,all,pres
sfede,all,all,pres
!
!
! do the 90 degree position
sfa,965,1,pres,p90front
sfa,967,1,pres,p90front
sfa,969,1,pres,p90front
sfa,972,1,pres,p90front
!
sfa,1026,1,pres,p90aft
!
/title,90 Degrees - Force = %force%
solve
!
! delete all pressures
sfade,all,all,pres
sfede,all,all,pres

```

```

!
!
! do the 135 degree position
sfa,876,1,pres,p135front
sfa,961,1,pres,p135front
sfa,963,1,pres,p135front
!
sfa,856,1,pres,p135aft
!
/title,135 Degrees - Force = %force%
solve
!
! delete all pressures
sfade,all,all,pres
sfede,all,all,pres
!
!
! do the 180 degree position
sfa,876,1,pres,p180front
sfa,983,1,pres,p180front
sfa,2260,1,pres,p180front
sfa,2376,1,pres,p180front
!
sfa,1041,1,pres,p180aft
sfa,2699,1,pres,p180aft
!
/title,180 Degrees - Force = %force%
solve
!
! now go to the postprocessor
/post1
!

```

APPENDIX III

The macro runpost.mac that was used to postprocess the results of the analysis is listed below. It generates plots and creates a text file that lists the forces and moments at each bolted joint, and what the maximum bolt load is likely to be.

```
!
! runpost.mac a macro that postprocesses the test rig results
!
! usage: runpost,key
!
!     where k=0 to generate plots and force summary file
!           =1 to only generate the force summary file
!
!
! set smin and smax boundaries on some stress plots
smin=0
smax=50000
!
! flag that activates plots using smin and smax
! 0=don't make extra plots with smin and max, 1=do make them
contflag=0
!
! flag to plot SDSG valuse
! 0=don't plot SDSG, 1=plot SDSG
sdsg=0
!
! estimated bolt radius from shaft
boltrad=2.0
!
allse
!
! enter the postprocessor
!
/post1
! set the view
!
/WIND,1, -1.0000      , 1.6660      , -1.0000      , 1.0000
/VIEW,1, 0.28741     , -0.74786     , 0.59842
/FOC, 1, 46.244      , -5.4349      , -31.281
/DIST,1, 28.090
/ANG, 1, 71.013      , ZS
/TYPE,1, 6
/EDG, 1, 0, 45.000
/CONT,1, 9,AUTO
!
! dump plots to files rather than the screen
/show, jpeg
!
! reverse white and black colors
/rgb,index,100,100,100,0
/rgb,index,80,80,80,13
/rgb,index,60,60,60,14
/rgb,index,0,0,0,15
```



```

!
!  remove the title
/title
!
!  set the triad to the lower left corner
/triad,lbot
!
*if,arg1,eq,0,then
!  select everything
allse
eplo
!  select each component and plot it
!
cmse,s,aftcorerigframe
eslv,s
eplo
!
cmse,s,aftendframe
eslv,s
eplo
!
cmse,s,bearings
eslv,s
eplo
!
cmse,s,frontcorerigframe
eslv,s
eplo
!
cmse,s,frontendframe
eslv,s
eplo
!
cmse,s,mountplates
eslv,s
eplo
!
cmse,s,mounts
eslv,s
eplo
!
cmse,s,rods
eslv,s
eplo
!
cmse,s,spacers
eslv,s
eplo
!
*endif
!
!  open a file that will contain the force and moment results
*cfopen,forceandmoment,txt
!
!  write a header for the file
*vwrite
(' ')

```

```

*vwrite
(' Loads on Spacers Between Core Rig Frames and Aft Core Rig Frame
and Aft End Frame')
*vwrite
(' ')
*vwrite
(' FX and FY are shear loads. FZ is a tension/compression load.')
*vwrite
(' ')
*vwrite
(' MX and MY are bending moments. FZ is a torsional load.')
*vwrite
(' ')
!
! make sure csys is zero
csys,0
!
! loop through the result sets and make plots
*do,i,1,5
!
allse
!
! load the result set
set,i
!
! get the /title of this analysis
*get,title1,active,,title,0,start,1
*get,title2,active,,title,0,start,9
*get,title3,active,,title,0,start,17
*get,title4,active,,title,0,start,25
*get,title5,active,,title,0,start,33
*get,title6,active,,title,0,start,41
*get,title7,active,,title,0,start,49
!
! select everything
allse
!
!
*if,arg1,eq,0,then
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
! select each component and plot it
!
! unselect the elements that have pressures applied to them
! plus one more set of elements
cmse,s,aftcorerigframe
asel,s,area,,849
asel,a,area,,856

```

```

asel,a,area,,1015
asel,a,area,,1026
asel,a,area,,1041
asel,a,area,,1054
asel,a,area,,1073
asel,a,area,,1078
asel,a,area,,1096
asel,a,area,,2540
asel,a,area,,2631
asel,a,area,,2690
asel,a,area,,2699
asel,a,area,,2715
asel,a,area,,2724
asel,a,area,,2741
asel,a,area,,2779
asel,a,area,,2786
nsla,s,1
esln
nsle
esln
einv
eslv,r
nsel,all
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
/auto
plns,s,eqv
ples,s,eqv
!
/WIND,1, -1.0000 , 1.6660 , -1.0000 , 1.0000
/VIEW,1, 0.28741 , -0.74786 , 0.59842
/FOC, 1, 46.244 , -5.4349 , -31.281
/DIST,1, 28.090
/ANG, 1, 71.013 , ZS
!
cmse,s,aftendframe
eslv,s
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif

```

```

!
allse
cmse,s,bearings
eslv,s
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
cmse,s,frontcorerigframe
asel,s,area,,876
asel,a,area,,961
asel,a,area,,963
asel,a,area,,965
asel,a,area,,967
asel,a,area,,969
asel,a,area,,972
asel,a,area,,973
asel,a,area,,976
asel,a,area,,977
asel,a,area,,979
asel,a,area,,981
asel,a,area,,983
asel,a,area,,2171
asel,a,area,,2190
asel,a,area,,2260
asel,a,area,,2277
asel,a,area,,2293
asel,a,area,,2306
asel,a,area,,2328
asel,a,area,,2357
asel,a,area,,2376
asel,a,area,,2398
asel,a,area,,2425
asel,a,area,,2438
asel,a,area,,2445
nsla,s,1
esln
nsle
esln
einv
eslv,r
nsel,all
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif

```

```

ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
! rear isometric view
/VIEW,1, 0.53333 , 0.10078 , -0.83988
/FOC, 1, 50.330 , 0.25218 , -41.194
/DIST,1, 14.681
/ANG, 1, 0.51726E-01,ZS
plns,s,eqv
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
/dsca,1,off
! struts only - isometric view
/VIEW,1, 0.19857 , -0.59282 , 0.78047
/FOC, 1, 47.786 , 0.17787 , -34.359
/DIST,1, 7.1314
/ANG, 1, 80.347 , ZS
vsel,s,volu,,47
vsel,a,volu,,345
eslv,s
nsle,s
csys,100
nsel,u,loc,y,-176.9,176.9
nsel,r,loc,x,4.782,5.460
nsel,r,loc,z,2,6.4
esln,r
nsle
csys
!
vsel,s,volu,,29,31
vsel,a,volu,,327,329
eslv,a
nsel,all
!
plns,s,eqv
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
allse
!
/WIND,1, -1.0000 , 1.6660 , -1.0000 , 1.0000
/VIEW,1, 0.28741 , -0.74786 , 0.59842
/FOC, 1, 46.244 , -5.4349 , -31.281
/DIST,1, 28.090
/ANG, 1, 71.013 , ZS
/dsca
!
cmse,s,frontendframe
eslv,s
plns,u,sum

```

```

plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
cmse,s,mountplates
eslv,s
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
cmse,s,mounts
eslv,s
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
cmse,s,rods
eslv,s
plns,u,sum
plns,s,eqv
*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
cmse,s,spacers
eslv,s
plns,u,sum
plns,s,eqv

```

```

*if,contflag,eq,1,then
/cont,1,9,smin,,smax
plns,s,eqv
/cont
*endif
ples,s,eqv
*if,sdsg,eq,1,then
ples,sdsg
*endif
!
*endif
!
! now get the forces and moments on each spacer
!
! there are 6 spacers. three between the rig frames, and three
between the
! aft core rig frame and the aft end frame. We'll get the forces
and moments
! that are acting on each one.
!
! we'll identify the spacers by their position and vertical (Y)
location.
!
! core rig frame spacers will be called coreylow, coreymid, and
coreyhigh.
!
! rig to end frame spacers will be called endylow, endymid, and
endyhigh.
!
! coreylow volume is 337
! coreymid volumes are 15 and 313
! coreyhigh volume is 39
!
! endylow volume is 419
! endymid volumes are 89 387
! endyhigh volume is 122
!
! *****
!
! turn on the working plane symbol so we can plot the spoint
location
!rn on the working plane symbol so we can plot the spoint location
!
wpstyle,,,,,,,,,1
! select everything
allse
! select the volume(s) for coreylow
vsel,s,volu,,337
!
! get the elements and nodes in these volumes
eslv,s
nsle
!
! get their centroid
centx=42.863
centy=-10.824
centz=-42.400

```

```

!
! put the workplane at this location and make a volume plot
! so we can confirm that spoint is in the proper location
wpave,centx,centy,centz,centx,centy,centz,centx,centy,centz
!vplot
!
! set the summation point to this location
spoint,,centx,centy,centz
!
! unselect nodes that are connected to the frontcorerig
cmse,s,frontcorerigframe
aslv
nsla,u,1
! sum the forces and moments that act on these elements
fsum
!
! get the forces and moments
*get,fsumx,fsum,,item,fx
*get,fsumy,fsum,,item,fy
*get,fsumz,fsum,,item,fz
*get,msumx,fsum,,item,mx
*get,msumy,fsum,,item,my
*get,msumz,fsum,,item,mz
!
! calculate shear forces, moment, and pull load on bolt
! the max bolt pull load is  $FX + ((msumx^2 + msumy^2)^{.5}) / boltrad$ 
msum=(msumx**2+msumy**2)**.5
ftotal=abs(fsumz)+msum/boltrad
!
! write these to a file
*vwrite
(' ')
*vwrite,i
(' Load Step ',f3.0)
*vwrite
(' ')
*vwrite,title1,title2,title3,title4,title5
(' ',5a8)
*vwrite
(' ')
*vwrite
(' Spacer at Core Y Low Position')
*vwrite
(' ')
*vwrite,fsumx,fsumy,fsumz
(' FX= ',f12.2,' FY= ',f12.2,' FZ= ',f12.2)
*vwrite,msumx,msumy,msumz
(' MX= ',f12.2,' MY= ',f12.2,' MZ= ',f12.2)
*vwrite
(' ')
*vwrite,msum
(' Total Moment= ',f12.2)
*vwrite,ftotal,boltrad
(' Max Total Force on a Single Bolt=',f12.2,' For a Bolt Radius of ',f12.2)
*vwrite
(' ')

```



```

!
! *****
! select everything
allse
! select the volume(s) for coreymid
vsel,s,volu,,15
vsel,a,volu,,313
!
! get the elements and nodes in these volumes
eslv,s
nsle
!
! get their centroid
centx=61.624
centy=.0040308
centz=-42.218
!
! put the workplane at this location and make a volume plot
! so we can confirm that spoint is in the proper location
wpave,centx,centy,centz,centx,centy,centz,centx,centy,centz
!vplot
!
! set the summation point to this location
spoint,,centx,centy,centz
!
! unselect nodes that are connected to the frontcorerig
cmse,s,frontcorerigframe
aslv
nsla,u,1
! sum the forces and moments that act on these elements
fsum
!
! get the forces and moments
*get,fsumx,fsum,,item,fx
*get,fsumy,fsum,,item,fy
*get,fsumz,fsum,,item,fz
*get,msumx,fsum,,item,mx
*get,msumy,fsum,,item,my
*get,msumz,fsum,,item,mz
!
! calculate shear forces, moment, and pull load on bolt
! the max bolt pull load is  $FX + ((msumx^2 + msumy^2)^{.5}) / boltrad$ 
msum=(msumx**2+msumy**2)**.5
ftotal=abs(fsumz)+msum/boltrad
!
! write these to a file
*vwrite
(' ')
*vwrite
(' Spacer at Core Y Mid Position')
*vwrite,fsumx,fsumy,fsumz
(' FX= ',f12.2,' FY= ',f12.2,' FZ= ',f12.2)
*vwrite,msumx,msumy,msumz
(' MX= ',f12.2,' MY= ',f12.2,' MZ= ',f12.2)
*vwrite
(' ')
*vwrite,msum

```

```

(' Total Moment= ',f12.2)
*vwrite,ftotal,boltrad
(' Max Total Force on a Single Bolt=',f12.2,' For a Bolt Radius of
',f12.2)
*vwrite
(' ')
!
! *****
! select everything
allse
! select the volume(s) for coreyhigh
vsel,s,volu,,39
!
! get the elements and nodes in these volumes
eslv,s
nsle
!
! get their centroid
centx=42.863
centy=10.824
centz=-42.400
!
! put the workplane at this location and make a volume plot
! so we can confirm that spoint is in the proper location
wpave,centx,centy,centz,centx,centy,centz,centx,centy,centz
!vplot
!
! set the summation point to this location
spoint,,centx,centy,centz
!
! unselect nodes that are connected to the frontcorerig
cmse,s,frontcorerigframe
aslv
nsla,u,1
! sum the forces and moments that act on these elements
fsum
!
! get the forces and moments
*get,fsumx,fsum,,item,fx
*get,fsumy,fsum,,item,fy
*get,fsumz,fsum,,item,fz
*get,msumx,fsum,,item,mx
*get,msumy,fsum,,item,my
*get,msumz,fsum,,item,mz
!
! calculate shear forces, moment, and pull load on bolt
! the max bolt pull load is  $FX + ((msumx^2 + msumy^2)^{.5}) / boltrad$ 
msum=(msumx**2+msumy**2)**.5
ftotal=abs(fsumz)+msum/boltrad
!
! write these to a file
*vwrite
(' ')
*vwrite
(' Spacer at Core Y High Position')
*vwrite,fsumx,fsumy,fsumz
(' FX= ',f12.2,' FY= ',f12.2,' FZ= ',f12.2)

```

```

*vwrite,msumx,msumy,msumz
('  MX= ',f12.2,'      MY= ',f12.2,'      MZ= ',f12.2)
*vwrite
('  ')
*vwrite,msum
('  Total Moment= ',f12.2)
*vwrite,ftotal,boltrad
('  Max Total Force on a Single Bolt=',f12.2,' For a Bolt Radius of
',f12.2)
*vwrite
('  ')
!
!
! *****
! select everything
allse
! select the volume(s) for endylow
vsel,s,volu,,419
!
! get the elements and nodes in these volumes
eslv,s
nsle
!
! get their centroid
centx=42.867
centy=-10.825
centz=-52.764
!
! put the workplane at this location and make a volume plot
! so we can confirm that spoint is in the proper location
wpave,centx,centy,centz,centx,centy,centz,centx,centy,centz
!vplot
!
! set the summation point to this location
spoint,,centx,centy,centz
!
! unselect nodes that are connected to the frontcorerig
cmse,s,aftcorerigframe
aslv
nsla,u,1
! sum the forces and moments that act on these elements
fsum
!
! get the forces and moments
*get,fsumx,fsum,,item,fx
*get,fsumy,fsum,,item,fy
*get,fsumz,fsum,,item,fz
*get,msumx,fsum,,item,mx
*get,msumy,fsum,,item,my
*get,msumz,fsum,,item,mz
!
! calculate shear forces, moment, and pull load on bolt
! the max bolt pull load is  $FX + ((msumx^2 + msumy^2)^{.5}) / boltrad$ 
msum=(msumx**2+msumy**2)**.5
ftotal=abs(fsumz)+msum/boltrad
!
! write these to a file

```

```

*vwrite
(' ')
*vwrite
(' Spacer at End Frame Y Low Position')
*vwrite,fsumx,fsumy,fsumz
(' FX= ',f12.2,' FY= ',f12.2,' FZ= ',f12.2)
*vwrite,msumx,msumy,msumz
(' MX= ',f12.2,' MY= ',f12.2,' MZ= ',f12.2)
*vwrite
(' ')
*vwrite,msum
(' Total Moment= ',f12.2)
*vwrite,ftotal,boltrad
(' Max Total Force on a Single Bolt=',f12.2,' For a Bolt Radius of
',f12.2)
*vwrite
(' ')
!
! *****
! select everything
allse
! select the volume(s) for endymid
vsel,s,volu,,89
vsel,a,volu,,387
!
! get the elements and nodes in these volumes
eslv,s
nsle
!
! get their centroid
centx=61.617
centy=0
centz=-52.764
!
! put the workplane at this location and make a volume plot
! so we can confirm that spoint is in the proper location
wpave,centx,centy,centz,centx,centy,centz,centx,centy,centz
!vplot
!
! set the summation point to this location
spoint,,centx,centy,centz
!
! unselect nodes that are connected to the frontcorerig
cmse,s,aftcorerigframe
aslv
nsla,u,1
! sum the forces and moments that act on these elements
fsum
!
! get the forces and moments
*get,fsumx,fsum,,item,fx
*get,fsumy,fsum,,item,fy
*get,fsumz,fsum,,item,fz
*get,msumx,fsum,,item,mx
*get,msumy,fsum,,item,my
*get,msumz,fsum,,item,mz
!

```

```

! calculate shear forces, moment, and pull load on bolt
! the max bolt pull load is  $FX + ((msumx^2 + msumy^2)^{.5}) / boltrad$ 
msum=(msumx**2+msumy**2)**.5
ftotal=abs(fsumz)+msum/boltrad
!
! write these to a file
*vwrite
(' ')
*vwrite
(' Spacer at End Frame Y Mid Position')
*vwrite,fsumx,fsumy,fsumz
(' FX= ',f12.2,' FY= ',f12.2,' FZ= ',f12.2)
*vwrite,msumx,msumy,msumz
(' MX= ',f12.2,' MY= ',f12.2,' MZ= ',f12.2)
*vwrite
(' ')
*vwrite,msum
(' Total Moment= ',f12.2)
*vwrite,ftotal,boltrad
(' Max Total Force on a Single Bolt=',f12.2,' For a Bolt Radius of ',f12.2)
*vwrite
(' ')
!
! *****
! select everything
allse
! select the volume(s) for endyhigh
vsel,s,volu,,122
!
! get the elements and nodes in these volumes
eslv,s
nsle
!
! get their centroid
centx=42.867
centy=10.825
centz=-52.764
!
! put the workplane at this location and make a volume plot
! so we can confirm that spoint is in the proper location
wpave,centx,centy,centz
!vplot
!
! set the summation point to this location
spoint,,centx,centy,centz
!
! unselect nodes that are connected to the frontcorerig
cmse,s,aftcorerigframe
aslv
nsla,u,1
! sum the forces and moments that act on these elements
fsum
!
! get the forces and moments
*get,fsumx,fsum,,item,fx
*get,fsumy,fsum,,item,fy

```

```

*get,fsumz,fsum,,item,fz
*get,msumx,fsum,,item,mx
*get,msumy,fsum,,item,my
*get,msumz,fsum,,item,mz
!
! calculate shear forces, moment, and pull load on bolt
! the max bolt pull load is  $FX + ((msumx^{**2} + msumy^{**2})^{**0.5}) / boltrad$ 
msum=(msumx**2+msumy**2)**.5
ftotal=abs(fsumz)+msum/boltrad
!
! write these to a file
*vwrite
(' ')
*vwrite
(' Spacer at End Frame Y High Position')
*vwrite,fsumx,fsumy,fsumz
(' FX= ',f12.2,' FY= ',f12.2,' FZ= ',f12.2)
*vwrite,msumx,msumy,msumz
(' MX= ',f12.2,' MY= ',f12.2,' MZ= ',f12.2)
*vwrite
(' ')
*vwrite,msum
(' Total Moment= ',f12.2)
*vwrite,ftotal,boltrad
(' Max Total Force on a Single Bolt=',f12.2,' For a Bolt Radius of ',f12.2)
*vwrite
(' ')
*vwrite
(' ')
!
! turn workplane symbol off
wpstyle,,,,,,,,0
!
*enddo
!
! close the file that contains the force and moment data
*cfclose
!
! list the forces and moments
*uilist,forceandmoment,txt
! reset the colors
/cmap,defa
! dump plot data to the screen
/show,term
/show,close
!
! all done

```

APPENDIX IV

The text file forcesandmoments.txt is shown below. It is generated by the postprocessing macro runpost.mac that is listed in APPENDIX III. This file lists the forces and moments at each bolted joint, and what the maximum bolt load is likely to be. Due to formatting limitations, some lines in the text below extend into a second line. The lines that are not indented are lines that belong to the line directly above it.

Loads on Spacers Between Core Rig Frames and Aft Core Rig Frame and
Aft End Frame

FX and FY are shear loads. FZ is a tension/compression load.

MX and MY are bending moments. FZ is a torsional load.

Load Step 1.

Zero Degrees - Force = 40000 Per Bearing

Spacer at Core Y Low Position

FX=	5498.36	FY=	317.61	FZ=	-3394.58
MX=	-10996.23	MY=	-27042.60	MZ=	-5037.89

Total Moment= 29192.79

Max Total Force on a Single Bolt= 17990.98 For a Bolt Radius of
2.00

Spacer at Core Y Mid Position

FX=	5922.40	FY=	0.00	FZ=	6789.15
MX=	-27.37	MY=	-8777.02	MZ=	23.87

Total Moment= 8777.06

Max Total Force on a Single Bolt= 11177.68 For a Bolt Radius of
2.00

Spacer at Core Y High Position

FX=	5498.36	FY=	-317.61	FZ=	-3394.58
MX=	10996.23	MY=	-27042.60	MZ=	5037.89

Total Moment= 29192.79

Max Total Force on a Single Bolt= 17990.98 For a Bolt Radius of
2.00

Spacer at End Frame Y Low Position

FX=	24946.86	FY=	-2428.90	FZ=	5570.86
MX=	23132.33	MY=	62872.20	MZ=	-56185.33

Total Moment= 66992.67

Max Total Force on a Single Bolt= 39067.20 For a Bolt Radius of
2.00

Spacer at End Frame Y Mid Position

FX=	6141.95	FY=	0.00	FZ=	-11141.68
MX=	0.00	MY=	-7042.19	MZ=	0.00

Total Moment= 7042.19

Max Total Force on a Single Bolt= 14662.77 For a Bolt Radius of 2.00

Spacer at End Frame Y High Position

FX=	24946.86	FY=	2428.90	FZ=	5570.86
MX=	-23132.33	MY=	62872.20	MZ=	56185.33

Total Moment= 66992.67

Max Total Force on a Single Bolt= 39067.20 For a Bolt Radius of 2.00

Load Step 2.

45 Degrees - Force = 40000

Spacer at Core Y Low Position

FX=	5276.44	FY=	6879.29	FZ=	2203.30
MX=	25192.00	MY=	-19012.90	MZ=	-6809.63

Total Moment= 31561.49

Max Total Force on a Single Bolt= 17984.05 For a Bolt Radius of 2.00

Spacer at Core Y Mid Position

FX=	4174.10	FY=	5199.38	FZ=	4762.53
MX=	22341.28	MY=	-6199.52	MZ=	-9439.75

Total Moment= 23185.49

Max Total Force on a Single Bolt= 16355.28 For a Bolt Radius of 2.00

Spacer at Core Y High Position

FX=	2515.77	FY=	6459.66	FZ=	-6965.84
MX=	41108.01	MY=	-19880.75	MZ=	673.91

Total Moment= 45663.04

Max Total Force on a Single Bolt= 29797.36 For a Bolt Radius of 2.00

Spacer at End Frame Y Low Position

FX=	36755.81	FY=	20552.15	FZ=	25595.96
MX=	28832.37	MY=	66303.93	MZ=	-98925.99

Total Moment= 72301.57
Max Total Force on a Single Bolt= 61746.75 For a Bolt Radius of 2.00

Spacer at End Frame Y Mid Position
FX= 4374.35 FY= -1010.97 FZ= -7863.54
MX= -12859.32 MY= -4986.65 MZ= 4448.30

Total Moment= 13792.35
Max Total Force on a Single Bolt= 14759.71 For a Bolt Radius of 2.00

Spacer at End Frame Y High Position
FX= -1478.43 FY= 23939.94 FZ= -17732.37
MX= -4072.35 MY= 22985.47 MZ= -19140.07

Total Moment= 23343.43
Max Total Force on a Single Bolt= 29404.08 For a Bolt Radius of 2.00

Load Step 3.

90 Degrees - Force = 40000

Spacer at Core Y Low Position

FX= 1976.11 FY= 9417.59 FZ= 6454.67
MX= 46416.99 MY= -162.88 MZ= -4765.10

Total Moment= 46417.27
Max Total Force on a Single Bolt= 29663.31 For a Bolt Radius of 2.00

Spacer at Core Y Mid Position
FX= -6.89 FY= 7329.75 FZ= -56.36
MX= 30862.26 MY= -25.56 MZ= -13054.76

Total Moment= 30862.27
Max Total Force on a Single Bolt= 15487.50 For a Bolt Radius of 2.00

Spacer at Core Y High Position
FX= -1968.76 FY= 9466.35 FZ= -6398.32
MX= 46956.44 MY= -761.75 MZ= -4287.35

Total Moment= 46962.62
Max Total Force on a Single Bolt= 29879.63 For a Bolt Radius of 2.00

Spacer at End Frame Y Low Position

FX=	26995.60	FY=	31465.92	FZ=	30643.62
MX=	17753.54	MY=	31061.92	MZ=	-83896.65

Total Moment= 35777.52
 Max Total Force on a Single Bolt= 48532.38 For a Bolt Radius of 2.00

Spacer at End Frame Y Mid Position					
FX=	41.92	FY=	-1399.11	FZ=	26.13
MX=	-17849.43	MY=	20.34	MZ=	6608.01

Total Moment= 17849.44
 Max Total Force on a Single Bolt= 8950.85 For a Bolt Radius of 2.00

Spacer at End Frame Y High Position					
FX=	-26996.18	FY=	31378.81	FZ=	-30669.74
MX=	17441.97	MY=	-30494.99	MZ=	-83408.72

Total Moment= 35130.71
 Max Total Force on a Single Bolt= 48235.10 For a Bolt Radius of 2.00

Load Step 4.

135 Degrees - Force = 40000

Spacer at Core Y Low Position

FX=	-2492.80	FY=	6455.56	FZ=	6876.63
MX=	40751.62	MY=	19358.37	MZ=	374.03

Total Moment= 45115.86
 Max Total Force on a Single Bolt= 29434.55 For a Bolt Radius of 2.00

Spacer at Core Y Mid Position					
FX=	-4155.47	FY=	5161.34	FZ=	-4765.23
MX=	21167.11	MY=	6168.88	MZ=	-8936.52

Total Moment= 22047.71
 Max Total Force on a Single Bolt= 15789.09 For a Bolt Radius of 2.00

Spacer at Core Y High Position					
FX=	-5315.72	FY=	6919.52	FZ=	-2111.40
MX=	24926.40	MY=	19516.61	MZ=	-7057.02

Total Moment= 31657.92
 Max Total Force on a Single Bolt= 17940.36 For a Bolt Radius of 2.00

Spacer at End Frame Y Low Position

FX=	1422.80	FY=	23893.61	FZ=	17765.92
MX=	-3901.77	MY=	-22734.94	MZ=	-19412.94

Total Moment= 23067.32

Max Total Force on a Single Bolt= 29299.58 For a Bolt Radius of 2.00

Spacer at End Frame Y Mid Position

FX=	-4385.95	FY=	-962.27	FZ=	7862.43
MX=	-12328.70	MY=	4995.57	MZ=	4972.00

Total Moment= 13302.35

Max Total Force on a Single Bolt= 14513.61 For a Bolt Radius of 2.00

Spacer at End Frame Y High Position

FX=	-36685.55	FY=	20479.46	FZ=	-25628.31
MX=	28961.29	MY=	-66554.21	MZ=	-99176.91

Total Moment= 72582.50

Max Total Force on a Single Bolt= 61919.56 For a Bolt Radius of 2.00

Load Step 5.

180 Degrees - Force = 40000

Spacer at Core Y Low Position

FX=	-5528.78	FY=	-304.14	FZ=	3348.58
MX=	11451.32	MY=	27926.92	MZ=	5493.74

Total Moment= 30183.53

Max Total Force on a Single Bolt= 18440.34 For a Bolt Radius of 2.00

Spacer at Core Y Mid Position

FX=	-5868.39	FY=	0.00	FZ=	-6697.13
MX=	26.99	MY=	8738.42	MZ=	-23.65

Total Moment= 8738.46

Max Total Force on a Single Bolt= 11066.36 For a Bolt Radius of 2.00

Spacer at Core Y High Position

FX=	-5528.78	FY=	304.14	FZ=	3348.58
MX=	-11451.32	MY=	27926.92	MZ=	-5493.74

Total Moment= 30183.53
Max Total Force on a Single Bolt= 18440.34 For a Bolt Radius of
2.00

Spacer at End Frame Y Low Position
FX= -24936.46 FY= 2373.04 FZ= -5545.23
MX= -23384.58 MY= -63368.38 MZ= 56627.87

Total Moment= 67545.47
Max Total Force on a Single Bolt= 39317.97 For a Bolt Radius of
2.00

Spacer at End Frame Y Mid Position
FX= -6240.82 FY= 0.00 FZ= 11090.50
MX= 0.00 MY= 7037.09 MZ= 0.00

Total Moment= 7037.09
Max Total Force on a Single Bolt= 14609.05 For a Bolt Radius of
2.00

Spacer at End Frame Y High Position
FX= -24936.46 FY= -2373.04 FZ= -5545.23
MX= 23384.58 MY= -63368.38 MZ= -56627.87

Total Moment= 67545.47
Max Total Force on a Single Bolt= 39317.97 For a Bolt Radius of
2.00

APPENDIX V

The two plots below show that the stress results for two loads are linearly related to the magnitude of these loads. The stresses that occur when a load of 40,000 lbs is applied to each bearing is exactly twice the stresses that occur when a load of 20,000 lbs is applied. Similarly, the maximum displacement is twice as high.

Figure A: von Mises stress where a 40,000 lb load is applied to each bearing at zero degrees. Maximum averaged von Mises stress is 92267 PSI. Maximum displacement is .211635 in.

Figure B: Maximum von Mises stress when a 20,000 lb load is applied to each bearing at zero degrees. The maximum averaged von Mises stress is 46134 PSI. Maximum displacement is .105818 in.

APPENDIX VI

Plots of the model and results are shown below in the order they are created. A complete set of plots can also be found on the CD that accompanies this report. The item that is being plotted is labeled in the upper left corner of each plot. The first load step is the zero degree case, the second load step is the 45 degree case, the third load step is the 90 degree case, the fourth load step is the 135 degree case, and the fifth load step is the 180 degree case.

1

ELEMENTS

MAT NUM

ANSYS

PLOT NO. 5

1

ELEMENTS

MAT NUM

ANSYS

PLOT NO. 6

1

NODAL SOLUTION

ANSYS

PLOT NO. 325

STEP=5
SUB =1
TIME=5
USUM (AVG)
RSYS=0
DMX =.740E-03
SMX =.740E-03

180 Degrees - Force = 40000

Yr.	Series	Job No.

DESIGN SUBSTANTIATION MEMO

Engine Model Rampressor 2	Engine Area Code HPC Rotor	Layout or Part No. 122310, 122307, 122318, 122329	Date 03-15-06
Subject: Rotor, Cover Plate, Snap Ring Design			

DISTRIBUTION:

Dave Taylor, Dan Bolduc, Brian Chisholm

INTRODUCTION:

This DSM records the rotor, cover plate and snap ring detail design.
The objectives for this Design Phase are:

- Complete rotor design in the area of the cover plate and snap ring.
- Design cover plate to retain segments and maintain sealing against segment and rotor preventing flow from segment to segment bleed cavities.
- Design snap ring to retain the cover plate.

Prepared by: D. Bolduc	Approved by: M. Ciak	Approved by: Dave Taylor

Yr	Series	Job No.

CONCLUSIONS:

The final geometry satisfies all design requirements for the rotor, cover plate and snap ring.

DISCUSSION:

Rotor:

The area of the rotor that was designed by MTU-AENA is the interface with the cover plate and snap ring. The remainder of the rotor is designed by Ramgen. There is a snap fit to center of the cover plate to the rotor. There is a loose fit on the snap ring in the rotor groove to allow for assembly. There are two alignment pin holes in the rotor to accept the pins in the cover plate for anti-rotation. Balancing areas are defined for detail (shown below) and assembly balancing (Disk post material removal). There is a gap (A) of .006"-.026" between the rotor and cover plate to maintain clearance. The anti-rotation pin diametral fit in the rotor (B) is .009"L-.023"L. Material of the rotor is Ti-6-2-4-2.

Cover Plate:

The cover plate was designed to have a diametral .000" T to .004" L fit against the rotor (D) with an engagement of .009"-.059". This fit is designed to allow cold assembly and tighten up at engine run. The vertical segment sealing arm is designed to sit flat against the segments and the rotor rim face to seal air from passing from cavity to cavity. The cover plate also has a pin with a diametral press fit (C) of .0003"T-.0013"T as an anti-rotation feature. There is a clearance of .008"-.038" between the end pin and the rotor (E). Knife edge seal definition for the cover plate was provided by Ramgen. There is a tab machined into the face of the cover plate to serve as an anti-rotation for the snap ring. Clearance from the cover plate ID to the rotor OD is diametral .019"-.039" (H). Cover plate and A/R pin material is Ti-6-2-4-2. There are 2 cover plates per assembly located fore and aft of the rotor.

Yr	Series	Job No.

Snap Ring:

The snap ring is designed to retain the cover plate. The ring is compressed into the rotor groove and when the cover plate is installed the ring expands into the groove of the cover plate. The snap ring has a fit of .005 L - .013 L in the rotor groove. There is capacity for the snap ring to be compressed into the rotor groove completely .005"-.025" clear (G) for assembly of the cover plate. It is designed to spring into place when installed and sit against the ID groove in the cover plate with .012"-.032" (F) engagement. Material of the snap ring is Ti-6Al-4V. There are 2 snap rings per assembly located fore and aft of the rotor.

AREAS OF CONCERN:

None

RECOMMENDATIONS:

None

Yr	Series	Job No.

REFERENCES:

Drawings:

Rotor: 122310 Rev B

Cover Plate: 122307 Rev A

Snap Ring: 122318 Rev A

Dowel Pin: 122329 N/C

Title: Ramgen Rampressor 2 Rig Structural Analysis

Part Name	Rampressor Rig	Subject	Structural & Life Analysis		
Scheme	N/A	Drawing No.	N/A	DSS No.	N/A
Project:	Ramgen Rampressor 2	Project-No.:	N/A	Number of pages:	19

Abstract

A structural analysis of the Ramgen Rampressor 2 test rig was conducted. Both 2D and 3D analysis were conducted using ANSYS. Iterations on coverplate geometries and initial assembly gaps were conducted to optimize fillet stresses and coverplate to segment gaps.

The final geometry satisfied all design requirements for the rotor. A final modification on coverplate geometry was done due to manufacturing issues. This final geometry satisfied the design requirements and improved the coverplate/segment sealing.

Certification Doc. Yes ☐ No ☒
 Language code en Module name Module No.
 Key words Ramgen, Compressor, ANSYS,
 Suggested synonyms
 Additional documentation disk thinning 6-22-05.ppt, disk thinning 6-28-05.ppt, 3d rotor 8-9-05.ppt, disk coverplate 9_19.ppt

Classification

	Prepared Brian Chisholm MTU AENA	Checked Lydia Geer MTU AENA	Approved	Accepted Dave Taylor RAMGEN
Date / Signature	March 17, 2006	March 17, 2006		

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Title sheet:

Report for action:

Report for information:

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1 Overview

A structural analysis of the Rampressor 2 compressor was conducted at MTU AENA under contract for RAMGEN. This analysis was performed in conjunction with design work on the rotor system that was also performed at MTU AENA.

A general arrangement of the rig is shown in Figure 1. The rotor assembly consists of a solid bore disk which also serves as the drive shaft for the rig, compressor segments that slide into slots in the disk, 2 coverplates that trap the segments in the disk and 2 snap rings that retain the coverplates. The front and rear coverplates and snaprings are identical and interchangeable.

Figure 1 – Ramgen Rampressor 2 Rotor General Arrangement

Both 2d and 3d analysis were conducted using the ANSYS finite element code. Mechanical and thermal loading for the models was provided by RAMGEN. Geometry was provided both by RAMGEN and by the MTU AENA design department.

The 2d analysis was used for initially sizing of the disk and coverplates. This analysis provided rapid turn around for iterations on disk thickness and coverplate geometry. Coverplate thicknesses, fits, and radii were modified during these preliminary iterations.

The 3d analysis was used to calculate disk slot and segment dovetail stresses as well as look at coverplate fillet stresses and sealing characteristics.

2 Design Requirements

The following design requirements were provided by RAMGEN:

Aerodynamic Design Point	41,837 RPM
Maximum Rotor Speed	46,021 RPM (120% ADP)
LCF Speed	44,766 RPM (107% ADP)
Burst Requirement	120% of Maximum Rotor Speed
Life Requirement	5,000 cycles at LCF Speed
Coverplate Sealing	The coverplate should maintain a line on line seal against the segments and disk from 95% speed (39,745 rpm) to 107% speed (44,766 rpm).

3 Materials

The rotor, coverplates, segments, and snap rings are all made from titanium alloys. The rotor and coverplates are made from Ti 6-2-4-2 alloy. The original design had Ti 6-4 coverplates however the LCF life requirements required the switch to Ti 6-2-4-2. The snap rings and segments are Ti 6-4 alloy. The detailed analysis of the segments was conducted by RAMGEN. The only portion analyzed by MTU AENA was the dovetail section.

4 Loading

The models were loaded with thermal, mechanical, and aerodynamic loading. The mechanical loading was input with both centrifugal loads from the rotor speed and forces of assembly due to interference fits. The method of applying the assembly loads is defined more completely in the details of the analysis.

Thermal loading was provided by RAMGEN. The thermal load was provided via an ANSYS database and result file. A single steady state time point was provided and is shown in Figure 2. The segment was not included in this thermal file. Temperatures for the segments were extrapolated from the rim and coverplate temperatures.

Aerodynamic loading was supplied as axial loading per segment. These were applied as point loads to the section of the segments that were modeled. A more complete description of this loading can be found in the details of the analysis.

Figure 2 – Ramgen Rampressor 2 Rig Temperature Profile

5 2D Analysis

5.1 Model Description

A 2-dimension finite element model was created to do most of the sizing of the major components. The model included the rotor, coverplates, snap rings, and a 2-D representation of the segments. This model is shown in Figure 3. The model was made up of 4 noded axisymmetric and plane stress elements. It contains a total of 11,302 elements and 11,821 nodes.

The blade segments are represented by plane stress elements. The total thickness of the plane stress elements is equivalent to the neck thickness of all the segments. The density of the segment elements is set to zero. The total mass of the segment is applied at its CG by a series of mass elements that are tied by rigid springs to the top of the segment elements (Figure 5). The wide rim and low CG location of the segment prevented use of a single mass element to represent the total segment mass. Mass elements were used instead of radial forces so that the loads would change with rotor speed input.

Figure 3 – Ramgen Rampressor 2 – 2D ANSYS Structural Model

5.2 Boundary Conditions

The model is constrained axially at the forward roller bearing (Figure 4). The shaft section of the rotor is solid. ANSYS applies a radial constraint at all nodes that lie on the centerline ($x=0$). A total axial load of 1039 lbs forward was applied to the segment. The load was applied to the mass elements nodes and evenly distributed across the segment.

The blade segment was tied radially to the rotor at the Z-plane. This was done via ANSYS coupled nodes in the X direction. This allowed the segment to slide axially in the blade slot due to forces applied to it by the coverplates and axial aerodynamic loads and those induced by rim rolling.

The remaining components are held together using contact elements and soft springs. Contact elements were created between the disk and coverplate, snap rings and coverplates, snap rings and disk, and coverplate and segments. These control axial or radial (depending on orientation) between the components. Soft springs were created between the coverplates and the segment at the OD. These springs helped with model convergence until the coverplates and segment contacts seated in the analysis.

Figure 4 – 2D Model Boundary Conditions

5.3 Load Cases

Four separate load cases were run.

- 1) Cold assembly – 68°F with no axial aerodynamic load and a small rotational speed
- 2) ADP – 41,837 RPM with aerodynamic load and temperatures
- 3) LCF – 44,766 RPM (107% ADP) with aerodynamic load and temperatures
- 4) Max Operational Speed – 46,021 RPM (110% ADP) with aerodynamic load and temperatures

5.4 Results

The results for the final design are presented here. Several design iterations were performed however only the final design is documented here.

5.4.1 Burst Calculations

Burst calculations were made at the maximum operational speed load case. Burst calculations were made using average tangential stress, average temperature, material ultimate strength, and a material utilization factor (MUF). The material utilization factor is derived from test data and the value for Ti 6-2-4-2 is 0.9. The equation for burst margin is:

$$BM = ((UTS * MUF) / ATS)^{1/2}$$

BM = Burst Margin

UTS = Ultimate Strength at Average Disk Temperature

MUF = Material Utilization Factor (0.9 for Ti 6-2-4-2)

ATS = Average Disk Hoop Stress

The entire rotor was not used to calculate the average hoop stress of the disk. Only the in-plane elements were used and the elements in the shaft areas were ignored (Figure 5). The hoop stresses in these elements is usually very low and artificially lowers the average. Also these areas are rarely involved in burst events.

The hoop stress plot for the elements used in the burst calculation are shown in Figure 6. The input and results for the rotor at 110% of the ADP speed are:

Average Disk Temperature = 284°F

Minimum Ultimate Tensile Strength = 113 KSI

Average Disk Hoop Stress = 55.7 KSI

Burst Margin = 1.35

Burst Speed = 62,326 RPM

Figure 5 – Ramgen Rampressor 2 – Burst Calculation Elements

Figure 6 – Ramgen Rampressor 2 – Burst Elements Hoop Stress

5.4.2 LCF Calculations

The LCF calculations were done using the loads from load case 2 which was run at the LCF rotor speed of 44,766 RPM for maximum stress and load case 1 (cold assembly) for minimum stress. Segment pull, aerodynamic loads, and temperature affects were included. Lifting was done using the MTU Aero Engines LCF system for Ti 6-2-4-2 alloy. Inputs to the LCF program are the maximum and minimum principal stresses and the corresponding temperatures. The calculated lives are B.1 lives (1 part in 1000) to a 1/32" crack length.

The results for the rotor and coverplates are shown in figures 7 and 8. All locations satisfy the 5000 LCF cycle requirement.

Figure 7 – Ramgen Rampressor 2 – Coverplate Area LCF Calculations – 44,766 RPM

Figure 8 – Ramgen Rampressor 2 – Web and ID LCF Calculations – 44,766 RPM

6 3D Analysis

6.1 Model Description

A 3D model of the rim section of the disk was constructed along with the ID section of a segment and sections of the coverplates. (Figures 9 and 10). This model was used to assess the LCF life of the disk slot and segment dovetail.

The model was run at the LCF rotor speed of 44,766 RPM. Additionally the segment axial and circumferential loads were included. The blade mass from the missing segment was applied to the model by point mass elements that were attached to the 3D segment using rigid beams.

Contact elements were used on the bearing surfaces of the disk and segment. They were also applied between the coverplate segments and disk and the coverplate segments and the blade segment that was included in the model.

Displacement boundary conditions at the ID the disk were taken from the 2D finite element analysis described in section 5. These conditions were mapped to the 3D model using the ANSYS sub-modeling process. Similarly the 3D temperature profile was created by sweeping the 2D profile to 3D. The applied temperature distribution is shown if figure 11.

Figure 9 – Ramgen Rampressor 2 – 3D Finite Element Model

Figure 10 – Ramgen Rampressor 2 – 3D Finite Element Model Coverplate Area Detail

C

Figure 11 – Ramgen Rampressor 2 – 3D Finite Element Model Temperature Distribution

An additional submodel was created of the disk attachment slot. Displacements were applied from the single slot model shown above as well as the temperatures using the ANSYS submodeling options. This model (Figure 12) is a much finer mesh giving a better stress distribution around the high stress area in the disk slot.

Figure 12 – Ramgen Rampressor 2 – 3D Finite Element Attachment Submodel

6.2 Results

The dovetail slot stresses from the single slot model that includes the segment and coverplates is shown in Figure 13. The 105.4 KSI stress is well within acceptable limits for the titanium material. The mesh in this area is fairly coarse and the submodel was run to get a better stress definition in the high stress area.

Figure 13 – Ramgen Rampressor 2 – 3D Finite Element Attachment Stresses – 44,766 RPM

The results from the submodel are shown in Figure 14. The finer mesh resulted in a higher predicted stress (120 KSI vs 105 KSI) in the submodel. The 120 KSI predicted stress however results in an acceptable predicted LCF life for the disk attachment (47,000 cycles B.1).

Figure 13 – Ramgen Rampressor 2 – 3D Finite Element Submodel Stresses – 44,766 RPM

The stresses in the coverplates and disk undercut areas were compared between the 2D and 3D models as a check to be sure the models were behaving similarly. The 3d stresses in these areas (Figures 14 and 15) are comparable to the 2D stresses in the same area under the same loading conditions. The 3D stresses are slightly lower than the 2D stresses but this is probably due to the courser mesh in the 3D model. Submodels of the coverplate and disk undercut area were not created but could have been used as a further check of the validity of the 3D model.

Figure 14 – Ramgen Rampressor 2 – 3D Finite Element Model Coverplate Stresses – 44,766 RPM

Figure 15 – Ramgen Rampressor 2 – 3D Finite Element Model Disk Undercut Stresses – 44,766 RPM

Stresses in the fillet of the segment dovetail were also reviewed (Figure 16). The mesh in this area is coarse so the actual stress will be higher than that shown. No submodel was created for this area. The stresses calculated are well below a level that would cause any concern in the rig and it was concluded that no further analysis needed to be conducted in this area.

Figure 16 – Ramgen Rampressor 2 – 3D Finite Element Model Segment Dovetail Stresses
44,766 RPM

7 Coverplate/Disk/Segment Sealing

A requirement of the design of the rotor was that the coverplate seal against the disk and segment. The secondary air flow system around the segment attachments required that the ID and OD be sealed as well as possible during the operation of the rig. This sealing was assessed in both the 2D and 3D models.

The 2D model showed a small gap at the ID between the coverplate and the disk while maintaining contact at the OD and ID of the segment and between the coverplate and disk at the ID. This was caused by the thermals causing the coverplate to grow towards the disk until contact was made. The OD hammerhead then closed the gap between the coverplate and segment at the OD causing the coverplate to bow resulting in the small gap (Figure 17). This small gap was deemed to be acceptable and a similar gap was seen at the rear coverplate as well.

Figure 17 – Ramgen Rampressor 2 – 2D Finite Element Model Coverplate Gap – 41,837 RPM (ADP)

The 3D analysis shows a slightly different result (Figures 18 and 19). The 3D analysis shows a small gap at the ID between the forward coverplate and the disk. It seals on segment at the ID and OD. The gap is caused by the axial gas load pushing the coverplate off the disk. The rear coverplate however seals tightly against the disk and segment at both the ID and OD.

Figure 18 – Ramgen Rampressor 2 – 3D Finite Element Model Coverplate ID Gap 41,837 RPM (ADP)

Figure 19 – Ramgen Rampressor 2 – 3D Finite Element Model Coverplate OD Gap
41,837 RPM (ADP)

8 Coverplate Redesign

A change was made to the coverplate design to ease manufacturing. The selected manufacturer of the coverplates felt that the thin OD face that seals against the segment was going to be hard to keep flat during manufacturing. To ease this problem the coverplate was redesigned with a tapered OD wall and a thicker ID section (Figure 20). The taper allowed the OD wall to be manufactured thicker to reduce distortion concerns and still allow the coverplate to seal against the segment. The thicker ID section maintained the stress level in the disk and coverplate.

The stress results from the new 2D analysis are shown in Figure 21. The disk stresses did not change and the coverplate stress still meets the LCF requirements of the design.

The coverplate sealing is shown in Figure 22. The 2D prediction shows a small gap at the ID between the disk and coverplate however a better seal along the length of the coverplate is obtained due to the taper of the OD wall.

No additional 3D analysis was conducted on this geometry change. It was deemed acceptable based on the results of the 2D analysis.

Figure 19 – Ramgen Rampressor 2 – 3D Finite Element Model Redesigned Coverplate Geometry

Figure 20 – Ramgen Rampressor 2 – 3D Finite Element Model Redesigned Coverplate Stresses
44,766 RPM

Figure 21 – Ramgen Rampressor 2 – 3D Finite Element Model Redesigned Coverplate Sealing
41,439RPM

9 Data Files

The following data files were used to perform the above documented analysis.

ramgen_2d_final_6242cp.db – ANSYS Database–2d model of final geometry with tapered coverplates

ramgen_2d_final_6242cp.inp – ANSYS control file – file to run analysis of final 2d model

ramgen_course_3d_ansys.db – ANSYS database – 3d model of rim segment

ramgen_course_3d_ansys.inp – ANSYS control file – file to run analysis of course 3d model

ramgen_submodel_3d.db – ANSYS database – 3d sub-model of disk slot for final stress analysis

ramgen_submodel_3d.inp – ANSYS control file – file to run analysis of slot sub-model

**Pages 3A-277 thru 3A-329 intentionally left blank under the provisions of
the Energy Policy Act of 1992**

Bearings • Roulements • Baleros

ADDENDUM TO

ANALYSIS & DESIGN OF MAINSHAFT BALL & ROLLER BEARINGS FOR THE *RAMGEN* SUPERSONIC COMPRESSOR RIG #2 (Incorporating *REVISION A* dated 7-13-04) July 15, 2004

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AUGUST 8, 2005

For

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INTRODUCTION

Design of the Rig #2 roller and split inner race ball bearings presented in the earlier report, dated July 15, 2004, included details of the radial stiffnesses of the ball and roller bearings computed by the author and which varied from about $0.5 \text{ E}6$ – $1.3 \text{ E}6$ and $1.1 \text{ E}6$ – $2.7 \text{ E}6$ respectively, depending on loads and speeds, in Table A of that report. Rotor dynamics computed by Barbour-Stockwell for Ramgen using these stiffness data indicated that values of $3 \text{ E}6$ or greater at all operating conditions were required to limit radial loads on the bearings. Accordingly, review of and necessary changes to the design of both bearings were requested by Ramgen with the requirement to raise the stiffnesses.

In order to minimize effects on the overall rig design, the review was limited to the internal geometries of the bearings, i.e., the (nominal) external dimensions were not to be changed. It was found during the early stages of this work that the $0.46875''$ ($7/16''$) diameter of the 12 balls of the ball bearing had to be raised to $12 @ 0.500''$, the other diameter usable in this size of bearing, to provide any hope of increasing the stiffness to the required level. In general, it was also found that considerable raceway curvature and contact angle changes were required in the ball bearing in turn dictating reduced internal clearance and the necessity for extremely careful review and control of differential thermal operating condition effects on internal clearance to obviate seizure. To facilitate this it was found necessary to change lubrication of the ball bearing from oil supplied into the bearing between the cage ID and the inner race OD by jet to the "under race" method. There oil is fed into the interior of the bearing via specially provided grooves in the locknut, shaft bearing journal and bearing inner race split face. This approach reduces temperature differences particularly between the shaft and inner race of the bearing.

Initially, it was assumed that a "trilobe" out-of-round outer race configuration change to the roller bearing would provide the solution for it. Parametric variations with reasonable values of out-of-round race and OD and internal clearances, however, failed to produce acceptable values of life and outer race bending stress even at the lower radial loads computed at the required stiffness. It was therefore necessary to revert to the original "round" configuration and determine over what range of operating clearance the required stiffness was obtained and, in effect, work "backwards" to establish the room temperature, as-manufactured clearance tolerance range. One option was a negative, or interference, operating clearance which would require increased emphasis on thermal effects. It was realized subsequently, however, that this approach would be totally unsatisfactory from the rig operation standpoint. The roller bearing would in effect act like a clamp on the shaft restricting its axial movement from axial differential thermal expansion and/or changes in rotor thrust. This could result in overload or insufficient thrust load on the ball bearing and fatigue or skidding wear failure respectively. Which scenario occurred would depend on operating "history", in other words, what axial position relative to the ball bearing the rotor shaft was in at startup, what speed and thrust changes had occurred, and the temperature changes of all components affecting rotor axial position – an extremely time-dependent situation. An almost infinite number of possibilities results with little or no chance to anticipate and for which to cater.

Other detail changes were found to be required or accepted during the course of this work as no Ramgen drawings were created based on the 7-13-04 drawing details provided in the referenced report and incorporating changes agreed subsequently by Ramgen and the author for use in procurement by Ramgen from vendor/manufacture Cerobear. The author did provide such information informally to Mr. Wemhoner at a meeting in San Diego on 10-12-04 but some details were not reflected in the early issues of the internal drawings produced by Cerobear virtually coincident with the beginning of manufacture. On the other hand, Cerobear has been extremely cooperative in meeting quite restrictive tolerances on some dimensions which have facilitated what has proven to be an extremely difficult design problem. It is believed by the author that the stiffness requirements imposed by the rotor dynamics subcontractor were excessively high resulting in bearing designs which require an inordinate amount of care in assuring satisfactory operation, not to mention the strong advisability of holding additional bearings of each type in a partly finished stage to permit, in as timely a fashion as possible, internal geometry "adjustments" if found to be required after initial rig operation.

PROCEDURE

The criticality of the internal geometry, particularly of the ball bearing, cannot be overemphasized in achievement of a successful outcome of this redesign effort. Also highlighted is the absolute necessity of employing a coupled bearing and heat transfer analysis to predict as closely as possible operating fits and clearance. For the ball bearing the major parameters involved were contact angle, inner and outer raceway curvature conformities (relative to ball diameter), and the tolerances on each. The contact angle exercises great influence on both stiffness and life. The curvatures affect life, including which race dictates life (usually the inner race), heat generation, and radial stiffness. Contact angle and curvatures taken together give the bearing's internal radial clearance. Too little clearance at room temperature renders the bearing susceptible to seizure during operation because of differential thermal expansion as well as fit-up and speed effects. Some variables work at cross purposes, for instance contact angle. The lower the contact angle, the greater is the radial stiffness but the lower the life. Conversely, with a realistic tolerance on contact angle, its maximum value might produce a lower than desirable (minimum) stiffness. There is also a complex relation between contact angle and curvature(s) as far as heat generation is concerned. Also, the higher the internally-generated temperatures, the lower is (at least the computed) life and, ultimately, the greater the likelihood of overall thermal instability and bearing seizure from lubrication failure. In addition, it was desirable to minimize the effects of fit-up on operating internal clearance by reducing as much as possible manufacturing tolerances on bearing bore and outside diameters as well as the mating shaft journal and housing bore diameters.

The first task, then, was to determine at least roughly a new set of internal geometry for the ball bearing and invite the manufacturer, Cerobear, to provide manufacturable tolerances on each of the parameters so that the computed effects of minimum and maximum values could be determined. Investigating the extremes was considered essential since it had become obvious that internal radial clearance would be on the low

side, and assurance of satisfactory operation at all conditions was needed. When stiffness, life, and fit-up effects had been evaluated over the speed and load variations specified and the geometry extremes, then (new) nominal geometry could be established with all the variables optimized as much as possible. During the course of these deliberations, it was found that Cerobear were able to maintain very close tolerances on raceway curvatures so it was only necessary to evaluate analytically what would be the nominal curvatures. Life and internal clearance considerations resulted in the outer race curvature being considerably less conforming than the inner with the result that the outer was statistically more likely to fail (in classical fatigue if that ever happened!) than, as is usual, the inner. It was also found that, for reasons not completely understood, computer solutions reaching fully converged iteration were often difficult to achieve, it being necessary to repeat cases with slight changes in thrust and/or radial loading. As it usually was the intent to explore upper and lower bounds of safe/satisfactory loads over the ranges of contact angles and speeds, many such incremental load changes were required to verify that a real limit had been reached and not just a mathematical instability, or whatever was involved. In any case, no geometry/load condition was considered viable unless full convergence was attained and all predicted operating parameters were satisfactory. Because the oil supply method for the ball bearing was changed from jet to under-race, the inner race fit-up on the shaft was altered as a result of revised differential thermals although the race growths due to speed-induced centrifugal effects were the same.

For the roller bearing, again Cerobear had to provide the smallest manufacturable internal radial clearance tolerance as well as, both Ramgen and Cerobear, the dimensions affecting fit-up. This would permit minimizing bearing clearance, the reason being to maximize radial stiffness. The task here was to identify the minimum room temperature clearance which would give the slightest positive operating clearance (i.e., as close to but not equal to or less than zero) so that the roller bearing, in operation, would not clamp down on its inner race and therefore the rotor shaft, restricting its axial movement as dictated by thrust load and/or differential thermal expansion. It was found that, as undesirable as this might be, the room temperature clearance, taking into account the inner race fit-up on its journal, was in contrast still tight. This in turn would dictate special consideration at installation of the rotor assembly into the housing assembly during build of the machine and most likely during rig shutdown during testing. The problem being that assurance would have to be provided that the rotor was positioned with the ball bearing "loaded" in the correct sense prior to each start-up. Because the clearance was the only parameter being changed, except perhaps fit-up, roller bearing analyses were easily accomplished along with the ball bearing studies mentioned above since the whole assembly was modeled for the bearing computer code SHABERTH with its coupled heat transfer capability (up to 100 nodes).

OPERATION

Operating characteristics below were investigated satisfactorily for bearing internal geometry with the following (nominal) values, these being the most critical for stiffness, life, and heat-generated temperatures.

For the ball bearing: Contact angle 16°
 Inner race curvature 51.3%
 Outer race curvature 56.1%

For the roller bearing: Internal radial clearance 0.0003"

Intended operating conditions: Speed 41837 - 46200 rpm
 Thrust load 1650 lb - 2500 lb (on the assumption of
 approximately 850 lb thrust variation)
 Radial stiffness $\geq 3 \text{ E6 lb/in}$ (target value)
 Life 100 hr. (minimum)
 Radial loads varying with speed and bearing per
 Barbour-Stockwell predicted rotor dynamics
 response shown in Figures 1 and 2 and
 approximated, for each position, as:
 41837 rpm - 75 lb.
 44766 rpm - 120 lb.
 46200 rpm - 150 lb.

Because the roller bearing operating fit-up resulted in a tight fit at room temperature, it will be necessary to ensure that the ball bearing clearance is taken up in the direction of normal thrust load prior to rig operation in each instance. The reason for this is to ensure that the ball bearing runs no risk of being unloaded during start-up. For initial start-up, rig assembly was instructed above with this in mind. During each rig shutdown, it will be necessary to pressurize the rotor system pneumatically to provide a forward thrust load to ensure that the ball bearing clearance is taken up in the same fashion and remains so when the roller bearing cools and immobilizes the rotor axially. Establishment of rotor axial position using output from a proximity probe statically would facilitate this procedure. Note that it may be necessary to take component temperatures into account depending on probe location relative to the bearing if a determination during or after running is desired. This could be by a plot of displacement versus (a significant) temperature. As long as the roller bearing is not applying a "clamping" effect on the rotor, the required pneumatic load to push the rotor forward should be minimal. If the rotor is not forward in the ball bearing clearance statically and the roller bearing is tight and immobilizing the rotor, gradually increasing pneumatic thrust until the proximity probe output matches the known displacement would be required.

To establish a recommendation for the maximum water-in-oil concentration, Exxon Mobil was contacted to see if they could/would advise a value for new oil. (Seldom if ever have such data been published for any oil.) For DTE 832, the Ramgen oil of choice, Mobil still doesn't have an "advertised" spec requirement but suggested that the "clear and bright" spec requirement means that if there were more than 300 - 400 ppm (parts per million) it would be visible. They added that new oil would probably therefore have less than 300 ppm. As an in-service limit 1000 ppm would be the max recommended. In Appendix A data can be found (pp. 214 - 220) to quantify the effect of water concentration in oil on bearing fatigue life. Some caution is required in using that data, though, as the then-available studies did not include the more modern turbomachinery

lubricants which have additives to inhibit water effects. As mentioned on the excerpted p. 220, the higher the bearing stress, the less the effect of water on life, probably because oil temperature in the contacts is higher promoting evaporation. Nevertheless, it is recommended that water content be a maximum of 1000 ppm, as suggested by Exxon Mobil for their DTE 832 as an in-service limit, or, if it is found that bearing/oil-out temperatures are less than 212° F, say, 650 ppm which is half way between 1000 and the 300 ppm Exxon Mobil expects of a new oil. If this range proves completely onerous, the maximum limit might be raised as high as 2000 ppm which is the limit employed by an industrial turbomachinery manufacturer familiar to both Ramgen and the author.

In the following section, pp. 220 – 228, of Appendix A can be found a discussion of the drastic effects on bearing life of particulate contaminants in the oil, even including material softer than the very hard bearing steels. See p. 227 in particular. This illustrates the importance of scrupulous cleaning of all rig oil-wetted components before assembly and initial operation plus the necessity for the best possible oil filtration, e.g. a 3 µm filter downstream of the pump.

RESULTS

The manufacturing (total) tolerances established for the bearings were as follows.

I Ball bearing tolerances

- Cerobear - Contact angle 2.5°
- Curvatures 0.4%
- Inner race bore 0.0002"
- Outer race O.D. 0.0003"
- Ramgen - Shaft journal 0.0002"
- Housing bore 0.0004"

II Roller bearing tolerances

- Cerobear - Internal radial clearance 0.0003"
- Inner race bore 0.0002"
- Outer race O.D. 0.0003"
- Ramgen - Shaft journal 0.0002"
- Housing bore 0.0004"

Applied to the nominal values above, the following geometry was established:

For the ball bearing: Cerobear drawing - Fig 3A	0500025H
Contact angle	15° - 17.5°
Inner race curvature	51.1% - 51.5%
Outer race curvature	55.9% - 56.3%
For the roller bearing: Cerobear drawings - Fig. 3B bearing	0500035F
- Fig. 3C roller	0200693
- Fig. 3D cage	0500040A
Internal radial clearance	0.0003" – 0.0006"

The Tables I and II following show the results at geometrical and fit-up extremes, including some outside specified tolerances for perspective, to provide guidance as to thrust and/or radial load/speed operating conditions which give suitable lives and temperatures and define as far as could be determined the limits of satisfactory operation as well as the resulting radial stiffnesses. While there is a table for each bearing, in many instances results were developed with a single computed case. To make the assessment matrix at all manageable, in terms of the number of cases to be run, cases were limited to those of minimum and maximum speeds and (ball bearing) contact angles. The data show that at the minimum speed of 41837 rpm, thrust load on the ball bearing should be between about 1450 lb. and just over 2000 lb. At maximum speed of 46200 rpm, thrust load on the ball bearing should be between about 1700 lb. and 2700 lb. For each case, the room temperature and operating fit-up (including change in radial internal clearance from 70°F, in metric and English) are given along with the computed temperature (in both °C and °F) of each bearing component and the oil-out temperature (for an oil-in temperature of 70°F in every case), the B-10 life in hours, and the radial stiffness in lb/in. (The metric values are included because the computer code outputs them and their conversion is facilitated in Excel.)

Appendix B presents example printouts of 2 cases noted in Table I. Input data are given at the end of each along with the complete output. Identification of the bearing nodes of prime interest listed across the head of the Tables I and II is given in the table footnotes. Also, Figure 4 illustrates the complete node layout which corresponds to the output temperatures at the end of each set of output. In Table I some entries are in **bold** characters to highlight them as meriting special consideration. In the case of **bold** temperatures thermocouple data should be monitored very closely at that operating condition to make sure that a runaway situation does not develop. In the one highlighted case quite a low life accompanies the higher temperatures. For stiffnesses (with fit-up within drawing tolerances, i.e. no case number with an <*>) below 2.5 E6, the *close* ones are *italicized* (2.45 E6 and above) while those **below** are in **bold** type. The max and min thrust loads solved for each speed are also in **bold** type.

HANDLING PROCEDURES

The most important objectives to be achieved in specifying handling procedures for rolling contact (i.e. ball and roller) bearings are avoidance, actually prevention and elimination, of damage of any kind and corrosion of the rolling contact surfaces. Either of these can dramatically reduce the rolling contact fatigue life of those surfaces. The Hertzian contact stresses, as they are called, over the very small areas of contact between the balls and rollers in operation, with which the operative subsurface fatigue is associated, are generally in the 200,000 to 300,000 psi (about 1400 to 2100 N/mm, the units of Shaberth stress output) range and are therefore bound to escalate to enormous (destructive) values if there are any stress concentrations present. The latter can be caused, for example, by even the slightest indentation arising from a blow applied to assist installation, dropping the bearing on the floor, even the smallest corrosion pit resulting from moisture on an internal surface for as little as an hour or two, and foreign particles in the lubricating oil. (This can be a factor even if, after initial rig operation for a

very short duration, oil which has entrained manufacturing debris from wetted components and supplied to bearings is changed.) Nowadays bearings do not usually fail in rolling contact fatigue, the "classical" mode of failure, because of the great strides in development of bearing metallurgy, advanced lubricants now available, sophisticated analytical techniques for design, etc. This may not be entirely true, however, where computed fatigue life has been traded off (analytically reduced toward the minimum desired test rig life) for one or more other operating parameters such as bearing radial stiffness and internal clearance as in the present case.

Bearings should therefore be retained in their original packaging until actually required for installation in the machinery in which they are to operate. When the packaging is opened they should only be handled by personnel wearing white cotton gloves, as lint-free as possible. Examples of sources of suitable gloves would be:

[1] Conservation Resources International, LLC

5532 Port Royal Rd.

Springfield, VA 22151

800.634.6932

Items LFCG-SM (small), LFCG-LG (large)

[2] Electron Microscopy Sciences

PO Box 550

1560 Industry Rd.

Hatfield, PA 19440

215.412.8450

Items 71119-01 (small), 71119-03 (medium), 71119-05 (large)

Cotton is preferred to the more slippery nylon to avoid dropping the bearings. If the gloves are not a snug fit, apply a "cinch" at the wrists.

If the bearings are not to be immediately installed, they should be transferred to a steady surface that has been scrupulously cleaned immediately before and protected from contamination of any kind. The bearings should then be placed on that surface and immediately covered with transparent wrap or similar material until required for actual installation. The reasons here are:

[1] To prevent fingerprint moisture from being transferred to any bearing surface

[2] To prevent dirt, dust or other contaminants from entering the bearings

If bearings are to be reused after operation in a machine and its subsequent disassembly, handling practices should be identical to those above. Unless there is evidence of corrosion when they are being removed, corrosion protection with the lubricant used in the machine should be adequate. (Otherwise, a vendor of oils for the aerospace industry should be contacted to obtain a small quantity of a suitable corrosion preventive oil compatible with the DTE 832 operating lubricant.) As soon as possible after removal the bearings should be enclosed in transparent wrap, as above, only being unwrapped as necessary for inspection and verification of suitability for reuse. Again, any handling should be only by personnel wearing white cotton gloves.

INSTALLATION PROCEDURES

The installation – and removal – of bearings start with the above handling practices and proceed in such a way as to reinforce and continue them.

In most cases, bearing races are very close if not tight fits on shaft journals and in housing bores. Even if there is a minimal clearance, e.g. 0.0005", between a bearing outer race O.D. and its mating housing bore, this may for all intents and purposes be considered "tight". Any attempt to simply slide the race in will in most cases result in its being cocked and not properly seated thereby requiring its (usually difficult) removal and reinstallation with a better, surer method.

There are three ways to install an outer race into its housing unless the clearance between the components is 0.001" (on diameter) or more in which case it can probably be slipped in as discussed above.

- [1] Heat the housing (bore) sufficiently to increase its diameter due to thermal expansion so that it is at least 0.001" greater than the bearing O.D. (obtain the actual dimension from the bearing vendor's inspection data tag for the actual bearing being installed) so that it can be inserted into the bore and properly seated axially against the housing shoulder. Ensure that there is a very light film of oil on the race O.D. either as supplied or that to be used in the machine. Be careful not to heat any component excessively but if any significant contact between parts occurs during installation, temperatures will quickly equalize; therefore err on the side of extra clearance. After temperatures have cooled, lightly tap at several locations around the exposed race endface as close to the O.D. as possible, with a brass or non-metallic "drift", to ensure that the race remains properly axially seated against the housing shoulder. Also squirt clean oil onto all the interior surfaces of rolling elements raceway, cage, etc., as applicable. Limit ingress of foreign material, moisture, etc. by use of transparent wrap or other suitable material.
- [2] Immerse the race in liquid air or liquid nitrogen so as to reduce its O.D. by thermal expansion (contraction in this case) so/until it is at least 0.001" less than the housing bore and it can then be slipped into that bore. If any contact between parts occurs during installation, temperatures will quickly equalize; therefore err on the side of extra clearance. Ensure that there is a very light film of oil as is to be used in the machine applied around the housing bore. After temperatures have warmed, lightly tap at several locations around the exposed race endface as close to the O.D. as possible, with a brass or non-metallic "drift", to ensure that the race remains properly axially seated against the housing shoulder. Also squirt clean oil onto all the interior surfaces of rolling elements raceway several times as and until temperatures equalize to ensure that any condensing moisture is not allowed to remain on bearing surfaces. Limit ingress of foreign material, moisture, etc. by use of transparent wrap or other suitable material.

This method requires more care subsequent to the actual installation because moisture will condense on the cold surfaces until all are at room temperature.

- [3] If the bearing race O.D. and housing bore are between, say, 0.0005" and 0.001" clearance, it is probably feasible to assemble the race into its bore by lightly tapping at several locations around the exposed race endface as close to the O.D. as possible, with a brass or non-metallic "drift", gradually advancing the race axially into its correct position by continuing the process around the circumference. Ensure that there is a very light film of oil on the race O.D. and/or housing bore either as supplied or that to be used in the machine. The race being tapped **MUST be the one being installed** – do not attempt to install a ball bearing outer race, for example, by pushing or tapping on the inner race. Ensure that the race has been properly axially seated against the housing shoulder. Limit ingress of foreign material, moisture, etc. by use of transparent wrap or other suitable material.

Installation of the inner races is a little more type-specific, particularly with a split inner race ball bearing as designed and manufactured for the Rig #2. Its inner race is in two halves with a "keeper" in the bore at the interface, the half which faces away from the disk when installed has a "puller groove", and close inspection will reveal a "V" marking part of which is on one half and the mating part on the other. The purpose of this marking is to show the circumferential alignment of the two halves when the dimensional inspection measurements were taken so that, when it is installed, assurance can be had that the bearing internal characteristics will be exactly as intended. (This is not to say that during operation one may not "creep" around its journal with respect to the other. Usually, though, the fit-up has been specified so that this will not happen but, if it does, it would be very unusual for it to result in a problem.)

The "keeper" is only to position the race halves for packaging, shipping, and handling convenience. Remove the "keeper" before installing the races as it could/will block oil supply through the slots in the interfacing surface on the forward half race. With the outer race assembly with balls and cage installed as above in the housing, the rear half of the inner race (without the puller groove) is installed onto the rotor shaft journal. The method chosen can be similar to either [1] or [2] above, i.e., the bearing inner race can be heated sufficiently to produce a clearance between its bore (obtain the actual dimension from the bearing vendor's inspection data tag for the actual bearing being installed) and the shaft O.D. or the shaft can be cooled and the ring slipped into position. Again, assure oil films are present but perhaps more sparingly here. In either case, assure that the race is still tight against the shaft shoulder when room temperatures have been reached and apply oil as required..

Extreme care is then required to centralize and keep centralized (radially) the rotor as it is introduced to the ball bearing outer race assembly, bringing the aft half of the inner race into contact with the balls. What is perhaps most critical is not allowing the corner of the inner race to scratch or dent the (ceramic) balls. It may be that a teflon, nylon, wood, etc.

assembly aid in the form of a hollow cone with its maximum diameter matching that of the corner in question would be helpful for this step. Keeping axial force on the rotor to maintain its position in the ball bearing and with rotor and housing centerlines coincident, the forward half of the ball bearing can be installed. This step will probably require heating of the race even if the aft half was installed by freezing the rotor journal. Again, if any contact between parts occurs during installation, temperatures will quickly equalize; therefore err on the side of extra clearance. Restrain axial movement of the rotor as the race is pushed on and, using appropriate thread lube, install the bearing locknut and assure that the halves of the inner race are in intimate contact at their split face. Ideally, the halves of the bore "V" marking should be aligned. It may be convenient to carry out appropriate drop checks to check this (including removal of the locknut as it need not be finally torque-tightened at this stage). The rotor must be carefully supported with centerlines aligned until the roller bearing installation has been completed. This assumes that rig assembly is carried out with the rotor centerline horizontal. If assembly to the point where the bearings have been completely installed with the centerline vertical has been accomplished, their installation may prove to be considerably simplified.

It may be convenient, on the other hand, to install the complete ball bearing on its shaft journal and then insert the bearing/rotor assembly into the housing bore. This would probably most conveniently be begun with the rotor axis vertical. Again, the "keeper" must be removed from the bearing bore. The halves of the inner race must be kept tight together at all times to obviate damage to the balls by the corners at the race split line. If the (whole) bearing is heated to install it on the shaft, the temperature must not exceed temperature limit of the Victrex PEEK cage, probably not much more than 300° F, otherwise the shaft journal will have to be cooled. During fitting of the bearing onto the journal it may be necessary to apply clamping to the inner race halves to keep them tight together. The locknut should then be installed for the same purpose and to ensure that the races are tight against the shaft shoulder to the point when temperatures have cooled to room temperature values. Installation of the roller bearing inner race at this point would probably be desirable. The housing bore for the ball bearing will then have to be heated and assembly completed as described above. Again, maintaining the rotor centerline as close to that of the housing bearing bores will be important.

The installation of the roller bearing inner race is similar and can be either of methods [1] or [2] above. For this particular bearing, in the room temperature installed condition, it has less than zero radial internal clearance. This is so that when it warms up during operation its clearance will be minimal to maximize its radial stiffness. So that when the rig first starts up the roller bearing does not "rob" the ball bearing of sufficient thrust for it to operate properly by, in effect, grabbing on to it, the rotor has to be pushed forward during the installation procedure while (artificially) maintaining a clearance in the roller bearing. How to accomplish this **continuously** until the roller bearing has been completely installed may take very careful consideration. This dictates, then, that the ball bearing assembly be complete before the final assembly of the roller bearing. The choice must be made whether to install the inner race on the rotor (at the same time as or before the rear half inner race of the ball bearing) first or as the last step. Installation of the

tapered rotor component adjacent to the inner race will most likely prove extremely helpful.

If the roller bearing inner race has been fitted to the rotor, both the roller bearing outer race assembly and the housing bore area will have to be heated to permit their installation over the roller bearing inner race (obtain the actual internal radial clearance dimension from the bearing vendor's inspection data tag for the actual bearing being installed for help in determining the required temperature). In this case, take care not to exceed the temperature limit of the Victrex PEEK cage, probably not much more than 300° F and/or provision of heat shielding would be advisable.

If the roller bearing outer race, cage and roller assembly has been installed in the housing, both the rotor journal and bearing race will have to be frozen but to different degrees to permit the race to be fitted on the journal (obtain the actual bore dimension from the bearing vendor's inspection data tag for the actual bearing being installed) and passed through the rollers. It may be necessary to heat the outer race assembly and housing as above, depending on actual stack-up.

Removal of the bearings from their assembled positions in the machine if there is intention to reuse them must be done at least as carefully as at their installation. This includes maintaining the rotor centerline alignment with that of the housing. Removal of the roller bearing outer race assembly by heating will be required. Unless the roller bearing requires replacement, the inner race can be left on its journal, otherwise remove it using an appropriate size puller tool engaging the side of the race facing the disk at the grooves in the shoulder provided for that purpose.

With the rotor restrained from forward movement and using a 3-prong puller tool, remove the forward half of the inner race of the ball bearing by engaging the prongs in the puller groove. Withdraw the rotor from the ball bearing outer race, cage and ball assembly. Unless the ball bearing requires replacement, the rear half of the inner race can be left on its journal, otherwise remove it using an appropriate size puller tool engaging the side of the race facing the disk at the grooves in the shoulder provided for that purpose. With a heavy press fit, use of a 3-prong puller tool is preferred. If the race is left in position, suitably protect it from corrosion and damage.

The outer race assemblies can be removed by lightly tapping at several locations around the exposed race endface as close to the O.D. as possible, with a brass or non-metallic "drift", gradually advancing the race axially out of its installed position by continuing the process around the circumference. Probably more desirable is use of 3-prong pullers with long-reach prongs which can be inserted through the rolling element-cage assemblies and reacted by a bar arrangement resting on the housing outboard of the bearing O.D.'s.

RAMPRESSOR 2 – FINAL DESIGN EFFORT

FIGURE 1

RAMPRESSOR & COUPLING - 50% BALL BEARING STIFFNESS – 50% ROLLER BEARING STIFFNESS

BALL BEARING FORCES

- DISK IMBALANCE = 0.22 GM.IN. / PLANE
- COUPLING IMBALANCE = 0.157 GM.IN
- BALL BRG. STIFFNESS = 1.375E6 LBF/IN.
  - 2500 LBF THRUST LOAD
  - 1/2 JONES PREDICTION
- ROLLER BRG STIFFNESS = 1.50E6 LBF/IN.
  - 100 LBF RADIAL LOAD
  - 1/2 PREDICTED
- CRITICALS AT 24,000 & 53,000 RPM

Speed	Calc	Use, lb, N
11927	63	75 3336
11916	55	120 5338
11920	20	150 6672

FIGURE 2

RAMPRESSOR 2 – FINAL DESIGN EFFORT

RAMPRESSOR & COUPLING - 50% BALL BEARING STIFFNESS – 50% ROLLER BEARING STIFFNESS

ROLLER BEARING FORCES

- DISK IMBALANCE = 0.22 GM.IN. / PLANE
- COUPLING IMBALANCE = 0.157 GM.IN
- BALL BRG. STIFFNESS = 1.375E6 LBF/In.
  - 2500 LBF THRUST LOAD
  - 1/2 JONES PREDICTION
- ROLLER BRG STIFFNESS = 1.50E6 LBF/In.
  - 100 LBF RADIAL LOAD
  - 1/2 PREDICTED
- CRITICALS AT 24,000 & 53,000 RPM

Speed	Calc.	Use, lb. N
11,637	58	75 333.6
22,764	58	120 533.8
46,200	114	150 667.2

11,637

TABLE I RAMGEN RIG 2 BALL BRG SHABERTH DATA

CASE NO.	SPEED RPM	THRUST LB.	RADIAL LOAD LB.	CONTACT ANGLE DEGREES	INNER RACE FIT ON SHAFT MM	I/RACE FIT TYPE	OUTER RACE FIT IN HO USING MM	O/RACE FIT TYPE	C
1**	41837	1348	100	15	-0.0356	NOM	-0.00889	NOM	69.435
2		1461	75	15	-0.0356	NOM	0.0051	ABOVE	70.214
3		1676	75	15	-0.0356	NOM	0.0051	ABOVE	69.46
4		1448	75	15	-0.0406	TIGHTTEST	-0.0152	TIGHT	67.987
5		1464	75	15	-0.0406	TIGHTTEST	0.0152	ABOVE	70.737
6		1673	75	15	-0.0356	NOM	0	LOOSEST	69.236
7		2089	100	15	-0.0356	NOM	-0.00889	NOM	115.034
8		2183	175	15	-0.0356	NOM	-0.00889	NOM	118.71
9		1399	75	17.3	-0.0356	NOM	-0.00889	NOM	94.129
10		1501	75	17.3	-0.0356	NOM	-0.00889	NOM	95.044
11		1907	75	17.3	-0.0356	NOM	-0.0076	~NOM	71.684
12		1903	75	17.3	-0.0406	TIGHTTEST	-0.0152	TIGHT	70.899
13		1973	75	17.3	-0.0356	NOM	-0.0076	~NOM	71.643
14		1687	75	17.3	-0.0406	TIGHTTEST	-0.0152	TIGHT	71.561
15		2029	75	17.3	-0.0356	NOM	-0.00889	NOM	71.743
16		2027	100	17.3	-0.0356	NOM	-0.00889	NOM	71.607
17		2048	125	17.3	-0.0356	NOM	-0.00889	NOM	108.987
18		2013	150	17.3	-0.0356	NOM	-0.00889	NOM	106.981
19		2041	200	17.3	-0.0356	NOM	-0.00889	NOM	116.538
20	46200	1757	100	17.3	-0.0356	NOM	-0.00889	NOM	114.163
21		1959	100	17.3	-0.0356	NOM	-0.00889	NOM	84.966
22		2197	150	17.3	-0.0356	NOM	-0.0089	~NOM	83.831
23		1934	150	15	-0.0356	NOM	-0.0076	~NOM	79.467
24		1941	150	15	-0.0305	LOOSEST	0	LOOSEST	80.752
25		2399	150	17.3	-0.0305	LOOSEST	0	LOOSEST	85.244
26		1814	150	15	-0.0356	NOM	-0.0076	~NOM	80.002
27		2170	150	15	-0.0356	NOM	-0.0076	~NOM	79.196
28		2169	150	15	-0.0406	TIGHTTEST	-0.0152	TIGHT	78.533
29		1923	150	15	-0.0406	TIGHTTEST	-0.0152	TIGHT	78.022
30		1706	150	15	-0.0406	TIGHTTEST	-0.0152	TIGHT	79.07
31		2464	150	17.3	-0.0406	TIGHTTEST	-0.0152	TIGHT	82.558

108.269	227	151.273	304	212	-4.25E-03	-0.00017	8.08E-02	0.00318	2.62E+06	32
109.333	229	153.080	308	291	-3.76E-03	-0.00015	8.13E-02	0.00320	2.54E+06	33
107.869	226	150.533	303	238	-5.37E-03	-0.00021	7.97E-02	0.00314	2.59E+06	34
108.053	226	150.880	304	258	-5.46E-03	-0.00021	7.96E-02	0.00313	2.59E+06	35
108.132	227	151.187	304	254	-5.34E-03	-0.00021	7.97E-02	0.00314	2.58E+06	36
108.787	228	152.429	306	240	-4.13E-03	-0.00016	8.09E-02	0.00319	2.58E+06	37
108.715	228	152.270	306	238	-4.49E-03	-0.00018	8.05E-02	0.00317	2.59E+06	38
107.836	226	150.567	303	194	-5.26E-03	-0.00021	7.98E-02	0.00314	2.63E+06	39
107.97	226	150.873	304	185	-5.53E-03	-0.00022	7.95E-02	0.00313	2.65E+06	40
108.148	227	151.478	305	167	-5.08E-03	-0.00020	8.00E-02	0.00315	2.67E+06	41

TABLE II RAMGEN RIG 2 ROLLER BRG

CASE ID	SPEED RPM	RADIAL LOAD LB.	RT RADIAL CLEARANCE MM	INTERNAL RAD CLR TYPE	INNER RACE FIT ON SHAFT MM	I/RACE FIT TYPE	OUTER RACE FIT IN HO USING MM	O/RACE FIT TYPE
A	41837	75	0.00762	0.00030	-0.0229	LOOSEST	0.0229	TIGHTTEST
B		75	0.00381	0.00015*	-0.0279	NOM	0.0152	BELOW
C		500	0.00762	0.00030	-0.033	TIGHTTEST	0.0229	TIGHTTEST
D		500	0.01524	0.00060	-0.0229	LOOSEST	0.0406	LOOSEST
E		150	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
F		500	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
G		1000	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
H	46200	150	0.00762	0.00030	-0.0229	LOOSEST	0.0229	TIGHTTEST
I		500	0.00762	0.00030	-0.0229	LOOSEST	0.0229	TIGHTTEST
J		500	0.01524	0.00060	-0.0229	LOOSEST	0.0406	LOOSEST
K		500	0.00762	0.00030	-0.033	TIGHTTEST	0.0229	TIGHTTEST
L		500	0.00762	0.00030	-0.033	TIGHTTEST	0.0152	BELOW
M		1000	0.00762	0.00030	-0.033	TIGHTTEST	0.0152	BELOW
N		1000	0.00762	0.00030	-0.0229	LOOSEST	0.0229	TIGHTTEST
O		200	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
P		400	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
Q		600	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
R		800	0.01143	0.00045	-0.0279	NOM	0.03175	NOM
S		1000	0.01143	0.00045	-0.0279	NOM	0.03175	NOM

* DENOTES OUTSIDE DRAWING TOLERANCE

** NODE IDENTIFICATION

- 21 SHAFT
- 22 INNER RACE
- 23 INNER RACEWAY
- 24 ROLLER
- 25 CAGE
- 26 OUTER RACEWAY
- 27 OUTER RACE
- 28 HOUSING
- 40 OIL OUT

RT = ROOM TEMPERATURE
INLET OIL TEMPERATURE = 70F ALL CAS ES

7	2 8			4 0			B.10 LIFE HR.	Δ INTERNAL CLEARANCE		RADIAL IN.	OPERATING C/EA RANGE MM	RADIAL IN	RADIAL STIFFNESS LB/IN	CASE ID
	F	C	F	C	F	MM.		IN.						
206	62.855	83.307	145	82.21E+06	4.85E-03	0.00019	1.25E-02	0.00049	1.38E+06	A				
216	65.793	103.012	150	22610	-1.63E-02	-0.00064	-1.24E-02	-0.00049	4.47E+06	B				
209	63.843	91.394	147	76770	-8.41E-03	-0.00033	-7.91E-04	-0.00003	2.24E+06	C				
207	63.144	88.507	146	40460	4.81E-03	0.00019	2.01E-02	0.00079	2.19E+06	D				
220	66.728	92.959	152	6.15E+05	5.49E-03	0.00022	1.69E-02	0.00067	1.48E+06	E				
219	66.337	96.207	151	14620	2.19E-03	0.00009	1.36E-02	0.00054	2.20E+06	F				
221	67.099	97.047	153	1447	2.30E-03	0.00009	1.37E-02	0.00054	2.54E+06	G				
216	65.68	89.379	150	1.29E+06	3.74E-03	0.00015	1.14E-02	0.00045	1.48E+06	H				
218	66.378	94.887	151	48970	6.09E-04	0.00002	8.23E-03	0.00032	2.21E+06	I				
219	66.441	95.026	152	32330	6.93E-03	0.00027	2.22E-02	0.00087	2.19E+06	J				
221	67.088	97.918	153	66730	-9.32E-03	-0.00037	1.70E-03	0.00007	2.25E+06	K				
228	69.202	110.048	157	6494	-2.24E-02	-0.00088	-1.48E-02	-0.00058	4.63E+06	L				
227	68.764	108.354	156	3139	-2.23E-02	-0.00088	-1.47E-02	-0.00058	4.53E+06	M				
222	67.35	96.845	153	5093	8.44E-05	0.00000	7.70E-03	0.00030	2.54E+06	N				
237	71.143	100.611	160	2.44E+05	7.65E-03	0.00030	1.91E-02	0.00075	1.54E+06	O				
236	70.799	100.085	159	23890	7.42E-03	0.00029	1.89E-02	0.00074	2.10E+06	P				
231	69.642	102.993	157	6796	4.42E-03	0.00017	1.59E-02	0.00062	2.28E+06	Q				
234	70.555	103.573	159	2585	4.46E-03	0.00018	1.59E-02	0.00063	2.42E+06	R				
232	69.876	103.343	158	1233	3.86E-03	0.00015	1.53E-02	0.00060	2.54E+06	S				

NODE OUTPUT TEMPERATURES

**

3A-349

P573 input $\frac{1}{2} \times 1000 \times 1000 \times 1000$ mm
 Effective length = $9 - 2 \times (.375)^2$. $9 - 1.5 = 7.5$ mm
 Net length = $4.075 \times 2 = 8.15$ mm

Allgemeintoleranzen ISO 2768 Teil 1					
Nennmaß- bereich	0,5 bis 3	über 3 bis 6	über 6 bis 30	über 30 bis 120	über 120 bis 400
Genaui- keitsgrad					
ISO 2768-m	±0,1	±0,1	±0,2	±0,3	±0,5

	Datum	Name	Artikelnummer VZR7.5X9.0000002		Allgemeintoleranzen DIN2768 - mK	
Gez.	10.06.02	Ni Zuckel	Bezeichnung Zylinderrolle VZR CN 7.5x9.0 G1 ZB nach Cerobear Werksnorm		CEROBEAR	
Gepr.	19.6.02					
Freig.	13.06.02					
Werkstoff Si3N4			Zeichnung Nr. 0200693		Maßstab: 10:1	
Zust.	Änderung/Revision 3A-352		Datum	Name		

Figure 3D

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FIG 4

RAMGEN RIG 2 SHAMBERTH HEAT
TRANSFER ANALYSIS NODE LAYOUT

COMPUTED FLOW OF
TEMPERATURE OF
COMPUTED COMPONENT
TEMPERATURE OF
FLOW/INPUT OF

PIPE FLOW
COOLING AIR FLOW
WATER RIG AIR FLOW

G.W. HOSANG

APPENDIX A

STLE Life Factors for Rolling Bearings

**Erwin V. Zaretsky
Editor**

1992

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* Chapter coordinator.

lubricants (according to J.L. Tevaarwerk of Battelle Memorial Institute, Columbus, Ohio). Once the traction coefficient is known, a life factor can be obtained from figure 5.16.

5.7 Contaminants

5.7.1 Water Corrosion Environments

Grunberg and Scott (1958) were among the first to identify the effect of water on premature failure due to fatigue of rolling-element bearings in outdoor appliances. They conducted tests in a four-ball fatigue tester on balls made from EN31 (similar to AISI 52100) and balls made from 12-percent-chromium stainless steel at 6.9 GPa (1000 ksi) with five lubricants at 1500 rpm. The lubricants were tested as received or were treated in one of the following ways:

- (1) Dried over metallic sodium for 24 hr before testing
- (2) Saturated with water before testing. A sample of the oil was thoroughly shaken with an equal volume of water and then allowed to settle for 24 hr. The oil was carefully decanted for testing.
- (3) Supersaturated with water during testing. To ensure excess water during testing, a small quantity (1 ml) was added to the 16 ml in the lubricant bath immediately after the start of each test. The fatigue life of the EN31 balls was considerably reduced as the water content was increased in all the lubricants studied. However, the stainless steel balls exhibited no reduction. These results suggest that corrosion fatigue was the cause of the reduced life effect.

In an effort to counter the deleterious effect of water in lubricants, Grunberg and Scott (1960) conducted further rolling-element fatigue tests in their four-ball fatigue tester under the same test conditions. Two different mineral oils were used as the test lubricants and were treated as previously described. An additional test series was performed with the lubricants supersaturated with water during the test (treatment 3), except that 1/2 ml of water was used instead of 1 ml and 3 percent isoamyl alcohol was added. The alcohol partially counteracted the deleterious effects of the water even though the alcohol lowered the viscosity considerably.

Grunberg and Scott (1960) tested six dewatering agents: a 3 percent isoamyl alcohol, oleic acid, triethanolamine, an imidazoline derivative, a sarcosine derivative, and a phenodin-acetic acid. Both oleic acid and triethanolamine lowered the fatigue life when saturated with water before testing (treatment 2) and raised the fatigue life under conditions of high saturation (treatment 3). Only the imidazoline derivative was effective in extending fatigue life under all conditions tested. The sarcosine derivative and the phenodin-acetic acid shortened fatigue life. Grunberg and Scott (1960) suggest that only the isoamyl alcohol (an aliphatic alcohol) and the imidazoline derivative (a water-

soluble corrosion inhibitor) prevented action by the water dissolved in the oil. These agents acted within the lubricant, whereas oleic acid and triethanolamine produced an adsorbed monolayer on the ball surface. This monolayer resulted in fatigue lives independent of water contamination.

However, Grunberg and Scott (1960) also found that corrosion fatigue played some part in the rolling-element fatigue process. It was not inhibited by water-soluble anticorrosives and dewatering agents other than the imidazoline derivative. Furthermore, stressing the surface layer of the bearing steel produced a high concentration of vacancies in the metal. If a ready source of hydrogen ions or atoms had been available, vacancy-induced diffusion of hydrogen into the surface would have occurred. The increase in hydrogen concentration in the surface would have produced "hydrogen embrittlement" and accelerated the fatigue process.

Scott (1962) continued work and found that alcohols higher than C_5 were less effective than the lower chain alcohols in inhibiting the effect of water-saturated oil. Under supersaturation there was little effect of any of the alcohols. However, in determining the effect of concentration, he found that 1 percent isoamyl alcohol was adequate to cope with water-saturated conditions (treatment 2) but that 5 percent was more effective under supersaturated conditions (treatment 3).

Scott (1962) also reevaluated the effect of the additives studied earlier. He found that an imidazoline "0" additive supply used with other dewatering agents was effective in a mineral oil containing "tritiated" water (i.e., including the radioactive isotope of hydrogen). Both tritium and hydrogen were adsorbed by the steel surface. The amount of adsorbed tritium increased with the number of stress cycles. These experimental results appeared to confirm the hypothesis that hydrogen penetration occurs in the water-accelerated spalling of rolling-element bearings. Scott also found that the behavior of the adsorbed hydrogen as determined by radioactivity measurements was similar to hydrogen embrittlement of ferrous materials. In both instances, diffusible as well as residual hydrogen was present in hydrogen adsorption on plastically deformed steels. The residual hydrogen contents increased with the extent of plastic deformation, but the amount of diffusible hydrogen decreased.

Schatzberg and Felsen (1968) conducted four-ball fatigue tests at four stress levels from 6.4 to 9.0 GPa (925 to 1300 ksi) with squalene, a refined paraffinic hydrocarbon, as the lubricant. Squalene was tested in a dry condition with a water content less than 10 ppm and in a wet condition with a water content of 100 ppm. The maximum dissolved water content for squalene at the system operating temperature of 52 to 54 °C (126 to 129 °F) was 160 ppm. Hence, the water concentration for the wet condition was approximately 65 percent of saturation. The 0.013-m (0.5-in.) diameter balls were AISI 52100 with a Rockwell C hardness of 62 to 63. The races were all made from AISI 52100 with a Rockwell C hardness of 59 to 60. The test speed was 1730 rpm. The results (fig. 5.18) show a decrease of 32 to 48 percent in fatigue life due to the increased water content at all stress levels. The pronounced influence on fatigue

Figure 5.18.—Rolling-element fatigue life as function of dissolved water content. From Schatzberg and Felsen (1968).

of such a small amount of water was explained by postulating a capillary crack condensation hypothesis. According to this hypothesis, dissolved water condensed into surface microcracks, which acted as fine capillaries. This process led to aqueous corrosion and hydrogen embrittlement within the crack which, when coupled with dynamic stress, shortened the time for the crack to grow to critical size. In addition, mild wear superimposed upon the fatigue process appeared to interact with it. This interaction was more pronounced in some cases than in others and was increased by the presence of dissolved water and oxygen. Dissolved oxygen in the lubricant was, on the basis of the results at one stress level, also found to cause a small but measurable reduction in fatigue life.

In another presentation of their work Schatzberg and Felsen (1969) added that the dissolved water in the lubricant which is forced into the capillaries condenses into the crack tips to form a water-rich phase. Supporting evidence for this hypothesis was found in the observed shift, owing to the presence of water, toward more fatigue spalls located at the "poles" of the balls. In that region more nonmetallic inclusion fibers intersect the surface with angles approaching 90° than anywhere else on the surface as a result of the original forging operation during ball manufacture. Therefore, more hard inclusion-martensite matrix interfaces occur in that region. These interfaces are potential surface defects. Microcracks formed at these interfaces become the capillaries into which water condenses. Water can also condense into the tips of grinding furrows, which are small enough to permit capillary condensation. This condensed water can initiate additional corrosion-generating surface defects.

Fein (1970) reported the 1968 unpublished results of R.C. Miller shown in table 5.8 that at 204°C (400°F) life decreases with increasing moisture content for 204-size ball bearings operating at 2.5 GPa (360 ksi). This occurs even though the bearing operating temperature is above the boiling point of

TABLE 5.8.—EFFECT OF MOISTURE IN TETRAESTER OILS ON ROLLING-ELEMENT FATIGUE LIFE OF 204-SIZE BALL BEARINGS

[Speed, 10 000 rpm; maximum Hertz stress, 2.4 GPa (350 ksi); outer-race temperature, 204 °C (400 °F); R.C. Miller's unpublished 1968 data in Fein (1970).]

Moisture content, ppm	Additive package		
	A	B	C
	L_{50} life, hr		
50	130	---	68
225	120	110	52
500	35	25	48

Figure 5.19.—Inhibition of water-accelerated rolling-element fatigue in four-ball fatigue tester. From Schatzberg (1971).

water. Schatzberg (1971) stated that Miller's results (in Fein, 1970) also showed that different additive packages present in the lubricant respond differently to the influence of water, implying that some additives can partially inhibit the effect.

Schatzberg (1971) also showed with the four-ball fatigue tester that adding 0.1 vol% isopropylaminoethanol (IPAE) completely counteracted the detrimental influence of 1 percent seawater emulsified with a petroleum-base hydraulic fluid containing rust inhibitors (fig. 5.19). No further benefit was obtained by using 0.25 vol% IPAE. It can be seen in figure 5.19 that the 1 percent water content resulted in reductions in the 10- and 50-percent

(L_{10} and L_{50}) fatigue lives of 48 and 60 percent, respectively. The steeper Weibull slope for the water-in-oil emulsion was deemed to be indicative of a chemical influence on the fatigue process. In explanation, Schatzberg (1971) noted that anodic and cathodic regions are present in the surface microcracks (which may develop early in rolling-element fatigue), the fresh metal surface at the crack tip being anodic. The ensuing corrosion results in the formation (in the absence of oxygen) of ferrous ions, ferrous hydroxide, and hydrogen ions. Reduction of hydrogen ions in a cathodic region of the crack results in the formation of atomic hydrogen, which can dissolve in the steel. He added that only the water in the seawater will dissolve in the oil because salt is not soluble in oil. Schatzberg (1971) believed that the IPAIE condensed from solution into the propagating subcritical microcracks along with the water. The amino alcohol is a proton acceptor, or basic substance; that is, the two extra electrons on the nitrogen atom will attract and accept protons. Consequently, any protons of hydrogen ions formed within the crack will be captured by the IPAIE.

Cantley and Widner (Schatzberg, 1971) suggested in their discussion of that work that the nitrogen atom could also adsorb onto the anodic sites within the crack to prevent the formation of a corrosion cell.

Felsen et al. (1972) extended this work with angular-contact ball bearing tests that showed reductions in rolling-element fatigue life as high as 80 percent during the flood-lubricated tests with 1 vol% seawater added to various lubricants as a contaminant. Again, the contaminant seawater was observed to produce greater Weibull slopes that became steeper as the amount of dissolved water increased. It might be observed that reported Rockwell C hardnesses of 68 for the balls and 66 for the races used in this testing were beyond the usual AISI 52100 hardnesses (62 to 64 for most balls and races) in standard practice. Such high ball hardness in particular raises the possibility of as-manufactured (brittle) cracking.

Polk and Rowe (1975) developed an optical method for measuring the extent of cracking in test specimens that had failed by fatigue under rolling-contact conditions. The fatigue tester comprised two hemispherically crowned rollers loaded against a specimen rod positioned between them. They tested both AISI 52100 and AISI M-50 steel specimens with a number of lubricants. Three tests included 0.05 percent water contamination in mineral base oil, which led to a higher cracking rate with a resultant decrease in fatigue life. The addition of 1 vol% IPAIE to the wet base oil decreased the average rate at which each secondary crack tended to branch and grow from the main crack by about 40 percent and increased the 50-percent (L_{50}) life by 20 percent. In the nominally dry base oil the IPAIE again decreased the average rate at which the secondary cracks grow and increased the fatigue life, showing that the additive inhibited the effects not only of added water but also of trace amounts that were present initially as well.

Cantley (1977) tested tapered roller bearings with varying amounts of water-contaminated SAE 20 mineral oil. The results obtained are shown in

TABLE 5.9. — EFFECT OF WATER
ON ROLLING-ELEMENT FATIGUE
LIFE OF 45-mm-BORE
TAPERED BEARINGS

[Additive content, 2-isopropylaminoethanol;
maximum Hertz stress, 2.03 GPa (294 ksi);
speed, 2700 rpm; oil outlet temperature,
66 °C (150 °F); elastohydrodynamic film
thickness, 0.29 μm (11.4 $\mu\text{in.}$);
from Cantley (1977).]

Lubricant type	Water concentration, ppm	L_{159}^a	Confidence limits ^b
SAE 20	25	2.59	1.73 to 3.43
	100	1.00	0.59 to 1.59
	400	.52	0.42 to 0.61
SAE 5 ^c	400	0.66	0.45 to 0.77

^aNormalized to 100 ppm.

^bIf the lower confidence limit of one group is greater than the upper confidence limit of another group, significant difference in life between the two groups exists at a 90-percent confidence level.

^cOil outlet temperature, 43 °F (110 °C).

table 5.9. Included is a result with SAE 5 oil with IPAE, which produced the same improvement that it did in the tests of Schatzberg (1971) and Polk and Rowe (1975). No explanation could be offered for this effect except that throughout much of the testing the additive decomposed into insoluble precipitate on the bearing surfaces and also on the bearing test rig housing surfaces. Hamaguchi et al. (1977), using optical interferometry to measure the EHD film thickness, determined that it was independent of the water dispersed in the oil.

Murphy et al. (1978), using a rotating-beam fatigue tester, determined the effectiveness of 0.01 wt% additives in mitigating the effects that water in SAE 20 mineral oil had on the fatigue life of steel. These results suggested that combinations of the reaction product of tetraethylene pentamine and alkyl succinic anhydride (TPA) with morpholine and IPAE with morpholine were effective in sequestering water at the steel surface.

Armstrong et al. (1978) reported an extensive test program to evaluate water-accelerated fatigue in ball bearings lubricated with circulating turbine oils. The oils were first screened in a rotating-beam fatigue rig, where bearing steel samples were exposed to the lubricating oil environment under severe rotating and bending conditions. As-received oils contained 40- to 60-ppm water, which was found to decrease to approximately 20 ppm unless water was added. For "wet" tests the water content was maintained at 400 to 600 ppm. Next, the oils were tested in a four-ball fatigue tester at high stress levels of 7.52 GPa (1090 ksi) at 54 °C (130°F). The initial water contaminant level was 0.5 percent so that 500 ppm would remain at the end of the test. Finally, confirmation

tests were made on two oils in a "unisteel" thrust ball bearing fatigue rig at the much lower stress level of 3.90 GPa (565 ksi). The acceleration of fatigue failure by water contamination was again confirmed in all three test series. For the particular type of lubricants and test methods used, it appeared that the reduction of fatigue life by small amounts of water was inversely related to the stress employed. Water in the oil caused a greater reduction in life in the unisteel bearing rig at the lower stress level. Likewise, the beneficial effect of the additives in the base oil was greater at the lower stress level than at high stress. The Weibull statistics of the unisteel tests showed that oil contaminated with 0.5 percent water had a three times longer L_{50} fatigue life than the corresponding wet base oil without the additive package.

Cantley (1977) offered the following relation of relative life to water concentration, which has been modified herein as follows:

$$LF = \left(\frac{25}{\text{ppm}} \right)^{0.6} \quad (5.14)$$

where ppm is the water concentration in parts per million and is equal to or greater than 25. Where $\text{ppm} \geq 250$, $LF = 0.25$.

When using equation (5.14), the bulk temperature of the oil, the oil-in temperature, or both should be less than 100 °C (212 °F). At higher temperatures it must be assumed that water will not be in the oil. Also, it is assumed that there is no effective additive package in the oil to inhibit water effects on fatigue and that the water will not be dissipated during operation. Equation (5.14) as now written further implies that at water concentrations of less than 25 ppm, the water will have no effect, although this may not necessarily be the case.

Figure 5.20 relates water absorption characteristics of various lubricants to relative humidity (Beercheck, 1978). When the relative humidity is known, this figure can be used in conjunction with equation (5.14) to approximate the values for ppm.

5.7.2 Particle Contamination and Filtration

It has been long recognized that lubricant contamination can affect bearing life, reliability, and performance. It is the number, size, and material properties of particles entering the Hertzian contact of the rolling element and race that affect bearing life. The nature of the particles in the oil is a function of several processes:

- (1) Manufacturing processes (swarf, chips, grit)
- (2) Internal generation, including wear debris and chemical attack of surfaces
- (3) Ingression from the environment (sand, dust)
- (4) Maintenance activities (making/breaking fittings, new oil)
- (5) Lubricant breakdown products (sludges, precipitates, coke)

Figure 5.20.—Water absorption characteristics of various lubricants as a function of relative humidity. From Beercheck (1978).

Along with selected test contaminants, typical particle size distributions from three different mechanical systems are shown in figure 5.21. The greater number of smaller particles in each of the lubrication systems is due in part to wear mechanisms that generate smaller particles and in part to removal processes that tend to remove larger particles. For comparison purposes two AC test dusts were used to represent mineral-derived contaminants such as silica sand. In addition to wear-generated particles Steele (1984) reported a wide range of particulate levels in unused turbine oils.

Hirano and Yamamoto (1958–59) reported that contaminants added to various lubricants could initiate scuffing in rubbing contacts. Dalal et al. (1974, 1975) reported that ball bearing lives over 50 times the Anti-Friction Bearing Manufacturers' (AFBMA) calculations were achieved by operating with pre-filtered ultraclean lubricant wherein the only source of metallic contamination was the bearing run-in. However, Dalal et al. (1974, 1975) did not run a control test lot of bearings without filtration to determine the exact improvement in life due to filtration. They later induced race damage in their test bearings with a hardness indenter in an attempt to simulate contaminant-caused indentations in service, but obtained longer fatigue lives than expected, although 7 of the 14 bearings tested failed at the Vickers indentation.

Fitzsimmons and Clevenger (1977) carried out an extensive test program with tapered roller bearings, using a variety of lubricants and contaminants with controlled particle sizes, types, and concentrations. They noted that two-body wear occurs when a hard rough surface plows a series of grooves in an opposing

Figure 5.21.—Particle contamination as function of oil filtration for mechanical systems. From Needelman and Zaretsky (1991).

softer surface. They stated that solid contaminants in lubricants are conducive to three-body abrasive wear, which occurs when particles are introduced between sliding surfaces and abrade material off both surfaces. They added the caveat that a certain amount of abrasive wear may be tolerable but is very application sensitive. As an example, noise in gears resulting from looseness in the bearing system rather than surface failure may constitute “failure” of the application.

Wear in tapered roller bearings normally occurs on the surfaces where there is combined rolling and sliding contact, such as the roller ends and cone (inner race) large-end rib. Fitzsimmons and Clevenger (1977) found that wear of these surfaces increased linearly with the concentration of hard contaminant particles. In addition, they stated that using extreme-pressure oils to prevent adhesive wear in gear systems (scoring, galling, etc.) may, in the presence of contaminants, increase abrasive wear, as experienced in actual wear tests on bearings. In their discussion of this work Kirnbauer and Farris (in Fitzsimmons and Clevenger, 1977) reported that 3- μm filtration prevents circulation of the hard particles that cause abrasive wear.

Oil filtration is a means to control and eliminate particle contamination from the lubrication system. Figure 5.22 shows particle size distribution for various absolute filter ratings as a function of the filtration factor β , where

Figure 5.22.—Particle size as a function of filter parameter for different filter ratings. From Needelman and Zaretsky (1991).

$$\beta = \frac{NU_X}{ND_X} \quad (5.15)$$

and NU_X and ND_X are the average number of particles upstream and downstream of the filter, respectively, whose particle size is greater than X μm . The data of figure 5.22 were generated by using AC fine test dust. A 3- μm filter is defined as $\beta_3 = 200$. A 25- μm filter is defined as $\beta_{25} = 200$. This means that for a 3- μm filter 99.5 percent of the particles greater than 3 μm are removed from the system. For a 25- μm filter, 99.5 percent of the particles greater than 25 μm are removed (Needelman and Zaretsky, 1991).

The distribution of β factors measured on a test stand is not necessarily the same as that obtained from operating machinery in the field. However, experience has shown that the test-stand rating qualitatively predicts differences in contamination levels in actual machine operation. As an example, according to Needelman and Zaretsky (1991) a 3- μm filter will keep the T-53 gas turbine lubrication system approximately two orders of magnitude cleaner than a 40- μm filter (fig. 5.22). Additionally, the T-700 gas turbine engine with 3- μm lubrication filters used in the Blackhawk and Seahawk helicopters has no scheduled oil changes for its 2000-hr time between removals. Along with protecting the base stock and additives, finer filtration can inhibit the blockage of oil passages

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and the coating of heat exchanger surfaces by deposits of varnish, resins, and coke (oil oxidation products). It can also minimize chemical attack of bearing surfaces by organic acids and oxidation products.

The removal efficiency E_X of any particle size X can be related to the β factor as follows:

$$E_X = 1 - \frac{1}{\beta_X} \quad (5.16)$$

From this relation it becomes apparent that there is a large particle size above which essentially nothing passes, an intermediate particle size at which a reasonable fraction of the particles are captured and the rest pass downstream, and a smaller size particle below which particles freely pass and accumulate to copious quantities (Needelman and Zaretsky, 1991).

Two independent investigations were conducted to determine the effect of oil filtration on rolling-element bearing life, that of Loewenthal et al. (1978, 1979, 1982) at the NASA Lewis Research Center, Cleveland, Ohio, and that of Bhachu et al. (1981) at the Imperial College, London, England.

Bhachu et al. (1981) used a gear test machine to generate wear debris. The gear wear debris, verified by ferrography to be representative of that found in helicopter gearboxes, was used as the contaminant. Rolling-element fatigue tests were conducted on 25-mm-bore AISI 52100 steel roller bearings having a 2.96-kN (665-lbf) radial load. For each test series the gear oil flow was passed through one of four possible filters of different ratings from (3 to 40 μm) or through an electromagnetic separator and was continuously supplied to a parallel-roller, bearing fatigue tester. Elastohydrodynamic film thicknesses and Λ ratios during testing are shown along with the test results in table 5.10. Significantly, tests run with 40- μm filtration for only 30 min before switching to 3- μm filtration showed substantially the same lives as if all running had been with 40- μm filters. Apparently the early damage could not be healed, at least in these small roller bearings. These results show that life increased with improved filtration, whereby it was suggested by Needelman and Zaretsky (1991) that

$$LF = 3.5(FR)^{-0.55} \quad (5.17)$$

The post-test inner-race measurements for 40- μm filtration showed greater out of roundness than in the unrun bearing. A gradual improvement was observed with finer filtration down to the 8- μm rating and then marked improvement for 3 μm , at which point there was virtually no out of roundness. Below the 3- μm level the measurement was similar to that of the unrun bearing. Bhachu et al. (1981) suggested that particles smaller than 3 μm were too small to have any effects on roundness and merely passed through the contacts. These data strongly suggest that the lack of contamination contributes to improvement

TABLE 5.10.—EFFECT OF OIL FILTRATION ON ROLLING-ELEMENT FATIGUE LIFE OF
25-mm-BORE ROLLER BEARINGS
[Radial load, 2.96 kN (665 lbf); original race surface finish, 0.38 μm . (15.2 $\mu\text{in.}$); from Blachut et al. (1981).]

Test filter rating (where $\beta \geq 200$), μm	Composite surface roughness after testing, σ , μm	Film parameter after testing, Λ	Experimental life, millions of inner-race revolutions				Weibull slope	Predicted L_{10} life from eq. (5.17), millions of inner-race revolutions
			10-Percent life, L_{10}	90-Percent confidence limits	50-Percent life, L_{50}	90-Percent confidence limits		
(a) Calculated film thickness, 0.58 μm (23.2 $\mu\text{in.}$)								
40	0.41	1.4	1.5	1.0 to 2.0	2.1	1.9 to 2.3	5.8	1.9
25	.36	1.6	2.5	2.0 to 3.0	3.0	2.9 to 3.1	9.4	2.5
8	.32	1.8	4.5	3.4 to 5.8	5.9	5.4 to 6.3	7.0	5.5
3	.26	2.2	8.0	5.5 to 11.6	12.0	10.7 to 13.5	4.7	8.0
2.5	.22	2.6	6.5	3.6 to 11.8	12.4	9.9 to 15.5	2.9	8.8
Magnetic	.20	2.9	5.0	2.9 to 8.5	10.0	8.3 to 12.0	3.2	---
(b) Calculated film thickness, 0.99 μm (39.6 $\mu\text{in.}$)								
40	0.41	2.4	1.8	-----	3.3	-----	3.1	2.2
3	.26	3.8	9.3	-----	16.2	-----	3.4	49.3
40/3	.41	2.4	1.7	-----	3.1	-----	3.3	2.2

^aLife prediction normalized to 3- μm filter.

in bearing race surface finish during operation. There appears to be a correlation between the Λ ratio after testing and the rolling-element fatigue life as evidenced by the data of table 5.10.

Loewenthal et al. (1978, 1979, 1982) performed a series of tests to measure the quantitative effects of filtration on rolling-element fatigue life. Four levels of filtration were investigated by using full-flow filters rated at 3, 30, 49, and 105 μm . The 3- μm filter used for these tests had been developed to replace the original 40- μm nominal filter for a helicopter gas turbine lubrication system. The new filter elements were found not only to provide a much cleaner lubricant with less component wear, but also, contrary to prior belief, to greatly extend the time between filter and oil changes as discussed previously.

The test bearings were 65-mm-bore deep-groove ball bearings run at 15 000 rpm under a radial load of 4.58 kN (1030 lbf), which produced a maximum Hertz stress of 2.4 GPa (350 ksi). The lubricant contaminant rate was 0.125 gram per hour per bearing. The test environment was designed to simulate an aircraft lubrication system comprising multiple bearings, pumps, and other components commonly found in such systems. The test temperature was 74 °C (165 °F). The test lubricant was a MIL-L-23699 type that produced a Λ ratio of 3.3. The test contaminant was similar to particulate matter found in the lubricant filters of 50 JT8D commercial engines (Jones and Loewenthal, 1980). Because this engine has a number of carbon-graphite bearing sump seals, replication of oil contaminants in engines with the "windback" type of labyrinth seal may demand a slightly different composition. On the other hand, increasing demands being placed on aircraft seal systems are dictating the use of face seals and other advanced concepts that usually employ carbon graphite or even ceramics, with the potential of generating extremely hard particles.

The results of these tests are summarized in table 5.11. As with the work of Bhachu et al. (1981), improved filtration extended bearing life. However, for the contaminated tests no statistical difference in life was obtained between the 3- and 30- μm filters. Wear was severe when the 105- μm filter was used with contamination. The 105- μm test series was suspended after 448 test hours on each bearing. No fatigue failures were encountered. On the basis of the test results for 3- and 49- μm filters, the following relation was suggested by Needelman and Zaretsky (1991):

$$LF = 1.8(FR)^{-0.25} \quad (5.18)$$

Table 5.12 compares equation (5.17) with equation (5.18) for various filter ratings. The resultant life factors can be used to adjust the calculated bearing L_{10} or catalog life to account for the filtration level in the lubrication system. It is not recommended that a life factor less than 0.5 be used with no filter. Furthermore, equation (5.18) appears to give more conservative results than equation (5.17). Although there may be differences in the effect of particle damage between roller and ball bearings as reflected by these two equations, it

TABLE 5.11.—COMPARISON OF 65-mm-BORE DEEP-GROOVE BALL BEARING FATIGUE LIFE RESULTS WITH AN ULTRACLEAN LUBRICANT AND WITH DIFFERENT LEVELS OF FILTRATION IN A CONTAMINATED LUBRICANT

[Radial load, 4580 N (1030 lbf); speed, 15 000 rpm; temperature, 74 °C (165 °F); test lubricant, MIL-L-23699 type; film parameter, Λ , 3.3; from Loewenthal et al. (1978, 1979, 1982).]

Lubricant condition	Test filter rating (where $\beta \geq 200$), μm	Experimental life, hr		Weibull slope	Failure index ^a	Confidence number, ^b percent	
		10-Percent life, L_{10}	50-Percent life, L_{50}			L_{10}	L_{50}
Ultraclean	3	1099	1741	4.1	5/9	--	--
Clean	49	672	2276	1.5	9/32	76	--
(baseline)	3	505	993	2.8	10/16	93	99
Contaminated	30	594	857	5.1	11/16	96	99
Contaminated	49	367	533	5.1	20/32	99	99
Contaminated	^c 105	-----	-----	-----	-----	---	--

^aNumber of fatigue failures out of number of bearings tested.

^bProbability (expressed as a percentage) that bearing fatigue life in a given test series will be inferior to the life obtained with ultraclean lubrication. A 90-percent or greater confidence number is considered statistically significant.

^cTest series was suspended after 448 test hours on each of the test bearings because of excessive bearing wear. No fatigue failures were encountered.

TABLE 5.12. — LIFE FACTOR BASED ON FILTER RATING

[From Needelman and Zaretsky (1991).]

Absolute filter rating (where $\beta \geq 200$), FR, μm	Life factor, LF	
	Roller bearing (eq. (5.17))	Ball bearing (eq. (5.18))
3	1.9	1.4
6	1.3	1.2
8	1.1	1.1
10	1.0	1.0
12	.9	1.0
25	.6	.8
40	.5	.7
^a 49	.4	.7
^a 60	.4	.6
^a 105	.3	.6

^aFor filter ratings less than 40 μm or with no filtration a life adjustment factor less than 0.5 is not recommended.

is recommended that equation (5.18) be used to adjust life calculations. It is also recognized that the contacts within these two bearings are small (3.5-mm ball bearing and 3.7-mm cylindrical bearing) so that extrapolating the life factor to larger bearings is not recommended (Lorösch, 1985).

The technology for improving oil filtration is commercially available. By minimizing the number of harmful particles entering a rolling-element bearing, oil filtration can substantially extend bearing life. In addition to machine-generated wear debris the comparatively high contamination of new oil also demands good filtration.

No data taken under controlled conditions exist in the open literature that compare grease lubrication, which entraps wear debris, with the same bearings oil lubricated with or without oil filtration. It is also suggested that for long-term application without regreasing the bearing, grease-lubricated bearings be considered the same as oil-lubricated bearings without filtration.

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APPENDIX B

FOR006

RUN ON APR-24-2005 AT 15:24:03.67

*** P R O G R A M /P573D14 ** GEO. W. HOSANG A L O U E T T E TECHNOLOGY ** P R O G R A M /P573D14 ***

RAMGEN RIG 2 108 ROLL 208 SPLIT I/R BALL HYBRID BRGS REDESIGN 573RAMGEN2HT22.DAT

Correct Alpha;Fits OIL 70F MOBIL DTE832 HT MODEL -2HT14,15X.DAT + U/race Lube #1

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIED IN MILLIMETERS, TEMPERATURES IN DEGREES CELSIUS, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CELSIUS. REF AFAPL-TR-76-89 & -90, SKF AL79P007L, SOLAR SR84-M-2134-00.

THIS DATA SET CONTAINS 2 BEARINGS

BEARING NO. (1) -

BALL BEARING

BEARING NO. (2) -

CYLINDRICAL ROLLER BEARING

THE MAXIMUM NUMBER OF MAIN LOOP ITERATIONS ALLOWED IS 61 AND THE RELATIVE ACCURACY REQUIRED IS .00010

MAX ITERATION NUMBER IN A CALL BEAR STATEMENT IN SHABE =150

MAX ITERATION NUMBER IN A CALL BEARC STATEMENT IN SONR1 =200

SOLUTION LEVEL =2

SURFACE FINISH CONVERSION FACTOR (FMULT) = 1.111

Y.M. CORRECTION FLAG (EPFLAG) = .10E+01

B E A R I N G D A T A

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	12	.000	60.000	.06405	-15.000	41837.	0.
2	14	.000	54.000	.00381	.000	41837.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	OUTER RING LAND RIDING	.381000	3.0000	65.0000	.254	.004000
2	OUTER RING LAND RIDING	.317500	3.0000	60.0000	.203	.003000

S T E E L D A T A

* NOTE * FOR HEAT TRANSFER CASES, LIFE FACTOR(S) MAY NEED MANUAL ITERATION BASED ON COMPUTED TEMPERATURE(S)

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	CRONIDUR 30	15.000	CRONIDUR 30	15.000
2	CRONIDUR 30	15.000	CRONIDUR 30	15.000

R O L L I N G E L E M E N T D A T A
BEARING NUMBER (1) TYPE - BALL BEARING

BALL DIAMETER OUTER RACEWAY CURVATURE INNER RACEWAY CURVATURE

12.7000
BEARING NUMBER (2) TYPE - CYLINDRICAL ROLLER BEARING .561
FOR006
.513

ROLLER DIAMETER	ROLLER LENGTH	ROLLER END SPHERE RADIUS	ROLLER INCL. ANGLE	AXIAL PLAY OUTER RING	INNER RING	OUTER RING	FLANGE ANGLE INNER RING
7.5000	9.0000	.0000	.000	.0000	.0000	.000	.000

EFF. LENGTH	OUTER RACEWAY FLAT LENGTH	CROWN RAD.	EFF. LENGTH	INNER RACEWAY FLAT LENGTH	CROWN RAD.	NO. OF AXIAL LAMINAE
7.4760	4.4450	254.000	7.4760	4.4450	254.000	2

S U R F A C E D A T A

BEARING NUMBER	OUTER RACEWAY FLAT LENGTH	CROWN RAD.	EFF. LENGTH	INNER RACEWAY FLAT LENGTH	CROWN RAD.	NO. OF AXIAL LAMINAE
1	.1020	.1020	.0127	1.500	1.500	1.500
2	.1020	.1020	.0508	1.500	1.500	1.500

BEARING NUMBER	OUTER RACEWAY FLAT LENGTH	CROWN RAD.	EFF. LENGTH	INNER RACEWAY FLAT LENGTH	CROWN RAD.	NO. OF AXIAL LAMINAE
1	.1020	.1020	.0127	1.500	1.500	1.500
2	.1020	.1020	.0508	1.500	1.500	1.500

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MOBIL DTE 832	29.60	.8600	7.20E-04	.130
2	MOBIL DTE 832	29.60	.8600	7.20E-04	.130

BEARING NUMBER	DESIGNATION	WALTHER EQUATION A	WALTHER EQUATION B	AUX TEMP COEFF BETA
1	MOBIL DTE 832	10.015990	3.582929	.015952
2	MOBIL DTE 832	10.015990	3.582929	.015952

L U B R I C A T I O N A N D F R I C T I O N D A T A

BEARING NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELEM.+ RACEWAY)	ASPERITY FRICTION COEFFICIENT
1	.50	.1000E-02	.10
2	.50	.1000E-02	.10

F I T D A T A A N D M A T E R I A L P R O P E R T I E S

BEARING NUMBER	COLD FITS SHAFT	(MM TIGHT) HOUSING	SHAFT	INNER RING	OUTER RING	HOUSING
1	.0356	.0089	36.0000	18.0000	18.0000	36.0000
2	.0279	-.0318	30.0000	15.0000	15.0000	30.0000

BEARING NUMBER	SHAFT I.D.	BEARING BORE	EFFECTIVE DIAMETERS INNER RING AVE. O.D.	OUTER RING AVE. I.D.	BEARING O.D.	HOUSING O.D.
1	.000	40.000	50.000	66.000	80.000	190.500
2	.000	40.000	48.000	64.000	68.000	190.500

BEARING NUMBER (1)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
1	113763.0	213734.0	310275.0	213734.0	193050.0
2	.3200	.3000	.2600	.3000	.2700

WEIGHT DENSITY
COEFF. OF THERMAL EXP. 4.540 7.806 FOR006 7.778
.00000770 .00001053 3.190 .00001080
.00000200

BEARING NUMBER (2) SHAFT INNER RING ROLL. ELEM. OUTER RING HOUSING
MODULUS OF ELASTICITY 113763.0 213734.0 310275.0 213734.0 193050.0
POISSONS RATIO .3200 .3000 .2600 .3000 .2700
WEIGHT DENSITY 4.540 7.806 3.190 7.806 7.778
COEFF. OF THERMAL EXP. .00000770 .00001053 .00000200 .00001053 .00001080

STEADY STATE TEMPERATURE CALCULATION. ITERATION LIMIT 155, ABSOLUTE ACCURACY 2.00 DEGREES
UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

NODE POINTERS
BRG SHAFT I. RING I. RACE ROLL EL. O. RACE O. RING HOUSING BULK FLANGE
1 11 12 13 14 16 17 18 45 17
2 21 22 23 24 26 27 28 42 27
NODES WHERE BEARING HEAT IS GENERATED
BRG INNER RACE OUTER RACE CAGE DRAG FLANGE
1 13 14 16 15 45 17
2 23 24 26 25 50 27

H E A T T R A N S F E R C O E F F I C I E N T S
TYPE INDEX COEFFICIENTS

CONDUCTION 1 46.7000
CONDUCTION 2 55.8000
CONDUCTION 3 27.7000
CONDUCTION 4 17.3000
CONDUCTION 5 26.0000
CONDUCTION 6 34.3000
CONDUCTION 7 2500.00
CONDUCTION 8 1500.00
CONDUCTION 9 500.000
CONDUCTION 10 17.4000

H E A T T R A N S F E R C O E F F I C I E N T S
TYPE INDEX COEFFICIENTS

FOR006

FORCED CONVECTION	21	.622000 .134000 .000000 .000000	.333000 .406000E-01 .000000	.333000 845.000 .000000	.476300E-02 1913.00 401.372	.212000 .000000 .000000
FORCED CONVECTION	22	.600000 .133100 .406000E-01 -3.55741	.500000 .000000 66.0000 .830000E-03	.310000 830.000 .982000E-02 -.190000	.540000E-01 1985.00 1403.03	51.9000 27.0000 .253808E+10
FORCED CONVECTION	23	.600000 .133100 .406000E-01 -3.55741	.500000 .000000 66.0000 .830000E-03	.310000 830.000 .982000E-02 -.190000	.540000E-01 1985.00 2006.04	106.100 27.0000 .253808E+10
FORCED CONVECTION	24	.600000 .133100 .406000E-01 -3.55741	.500000 .000000 66.0000 .830000E-03	.310000 830.000 .982000E-02 -.190000	.794000E-02 1985.00 5231.50	106.100 27.0000 .253808E+10
FORCED CONVECTION	25	.300000 .261000E-01 .000000 .000000	.570000 .185100E-04 .000000 .000000	.000000 1.17700 .000000 .000000	.635000E-02 1004.00 1054.50	345.000 .000000 .000000
FORCED CONVECTION	26	.300000 .261000E-01 .000000 .000000	.570000 .185100E-04 .000000 .000000	.000000 1.17700 .000000 .000000	.635000E-02 1004.00 1112.54	379.000 .000000 .000000
FORCED CONVECTION	27	.860000E-01 .288000E-01 .185100E-04 .254215	.700000 .000000 66.0000 .110500E-05	.350000 1.10500 .203300E-04 -.350000	.500000E-01 1005.00 6.64546	106.100 27.0000 .670648
FORCED CONVECTION	28	.300000 .288000E-01 .185100E-04 .254215	.570000 .000000 66.0000 .110500E-05	.000000 1.10500 .203300E-04 -.570000	.525000E-01 1005.00 .308466	51.9000 27.0000 .670648
FORCED CONVECTION	29	.300000 .288000E-01 .185100E-04 .254215	.570000 .000000 66.0000 .110500E-05	.000000 1.10500 .203300E-04 -.570000	.794000E-02 1005.00 1.04464	106.100 27.0000 .670648

H E A T T R A N S F E R C O E F F I C I E N T S
INDEX

FLUID FLOW

41

FOR006

TYPE OF HEAT TR.	INDEX	NODE	NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
FLUID FLOW	42	11	AND 1	10.0000	20.0000	30.5000
		1	AND 2	10.0000	20.0000	15.5000
FLUID FLOW	43	2	AND 3	15.0000	30.0000	14.0000
		1	AND 4	29.0000	58.0000	20.5000
FLUID FLOW	44	48	AND 52	72.7500	29.5000	35.0000
		1	AND 5	29.0000	58.0000	31.0000
FLUID FLOW	45	67	AND 10	97.0000	62.0000	3.0000
		1	AND 9	94.0000	62.0000	3.0000
FLUID FLOW	46	10	AND 60	91.0000	62.0000	3.0000
		1	AND 52	87.5000	62.0000	16.5000
FLUID FLOW	47	48	AND 3	58.0000	2.0000	60.0000
		1	AND 6	58.0000	2.0000	60.0000
FLUID FLOW	48	51	AND 4	58.0000	31.0000	72.7500
		1	AND 5	58.0000	31.0000	72.7500
FLUID FLOW	49	52	AND 6	29.0000	58.0000	20.5000
		1	AND 7	15.0000	30.0000	14.0000
FLUID FLOW	50	6	AND 52	72.7500	29.5000	35.0000
		1	AND 8	10.0000	20.0000	15.5000
FLUID FLOW		7	AND 21	10.0000	20.0000	20.0000
		1	AND 12	20.0000	18.0000	22.0000
FLUID FLOW		11	AND 13	25.0000	18.0000	2.5000
		1	AND 1	22.0000	5.0000	14.0000
FLUID FLOW		12	AND 17	36.0000	18.0000	2.5000
		1	AND 18	40.0000	18.0000	15.0000
FLUID FLOW		17	AND 19	43.5000	7.0000	62.0000
		1	AND 20	43.0000	6.0000	30.0000
FLUID FLOW		18	AND 17	36.0000	5.0000	27.0000

DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER
ALL LENGTHS ARE IN MILLIMETERS, A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

CONDUCTION	1	BETWEEN	19	AND	17	37.5000	5.0000	62.0000
CONDUCTION	1	BETWEEN	61	AND	18	47.0000	40.0000	70.0000
CONDUCTION	1	BETWEEN	61	AND	19	47.0000	106.0000	30.0000
CONDUCTION	1	BETWEEN	61	AND	20	60.0000	25.0000	80.0000
CONDUCTION	1	BETWEEN	20	AND	59	34.0000	8.0000	20.0000
CONDUCTION	1	BETWEEN	21	AND	22	20.0000	15.0000	21.5000
CONDUCTION	1	BETWEEN	22	AND	23	23.0000	15.0000	2.5000
CONDUCTION	1	BETWEEN	22	AND	8	21.5000	3.0000	12.5000
CONDUCTION	1	BETWEEN	26	AND	27	31.0000	15.0000	2.5000
CONDUCTION	1	BETWEEN	27	AND	28	34.0000	15.0000	18.5000
CONDUCTION	1	BETWEEN	28	AND	29	40.0000	16.5000	8.5000
CONDUCTION	1	BETWEEN	29	AND	27	31.5000	3.0000	50.0000
CONDUCTION	1	BETWEEN	62	AND	29	47.0000	106.0000	30.0000
CONDUCTION	1	BETWEEN	62	AND	28	47.0000	40.0000	70.0000
CONDUCTION	1	BETWEEN	28	AND	31	40.0000	13.0000	30.0000
CONDUCTION	1	BETWEEN	28	AND	31	31.5000	3.0000	34.0000
CONDUCTION	1	BETWEEN	62	AND	31	60.0000	25.0000	80.0000
CONDUCTION	1	BETWEEN	31	AND	30	34.0000	8.0000	25.0000
CONDUCTION	8	BETWEEN	43	AND	54	34.5000	25.0000	12.0000
CONDUCTION	8	BETWEEN	40	AND	57	34.5000	25.0000	12.0000
FORCED CONVECTION	21	BETWEEN	66	AND	19	36.0000	103.0000	
FORCED CONVECTION	21	BETWEEN	66	AND	61	2.3800	45.0000	
FORCED CONVECTION	22	BETWEEN	45	AND	11	15.0000	5.0000	
FORCED CONVECTION	-22	BETWEEN	45	AND	11	14.3640	20.0000	
FORCED CONVECTION	22	BETWEEN	45	AND	11	19.0000	2.3940	
FORCED CONVECTION	22	BETWEEN	45	AND	12	20.0000	1.5700	
FORCED CONVECTION	-22	BETWEEN	45	AND	11	14.3640	14.0000	
FORCED CONVECTION	22	BETWEEN	45	AND	12	9.1400	14.0000	
FORCED CONVECTION	22	BETWEEN	45	AND	11	19.5000	2.3940	
FORCED CONVECTION	22	BETWEEN	45	AND	12	20.0000	1.5200	
FORCED CONVECTION	-22	BETWEEN	45	AND	12	18.3500	3.6500	
FORCED CONVECTION	22	BETWEEN	44	AND	16	33.0000	18.0000	
FORCED CONVECTION	-22	BETWEEN	44	AND	15	541.0000	12.0000	
FORCED CONVECTION	-24	BETWEEN	44	AND	14	506.7000	12.0000	
FORCED CONVECTION	23	BETWEEN	44	AND	13	25.0000	18.0000	
FORCED CONVECTION	21	BETWEEN	43	AND	1	22.5000	20.0000	
FORCED CONVECTION	21	BETWEEN	43	AND	18	4.7600	12.0000	
FORCED CONVECTION	25	BETWEEN	43	AND	61	4.7600	40.0000	
FORCED CONVECTION	25	BETWEEN	64	AND	61	2.3800	145.0000	
FORCED CONVECTION	25	BETWEEN	64	AND	20	60.0000	25.0000	
FORCED CONVECTION	28	BETWEEN	38	AND	20	60.0000	30.0000	
FORCED CONVECTION	28	BETWEEN	46	AND	61	72.0000	15.0000	
FORCED CONVECTION	27	BETWEEN	38	AND	3	45.0000	30.0000	
FORCED CONVECTION	27	BETWEEN	38	AND	48	65.0000	15.0000	
FORCED CONVECTION	27	BETWEEN	46	AND	48	80.0000	10.0000	
FORCED CONVECTION	28	BETWEEN	47	AND	61	10.0000	145.0000	
FORCED CONVECTION	27	BETWEEN	36	AND	2	30.0000	20.0000	
FORCED CONVECTION	28	BETWEEN	36	AND	59	35.0000	15.0000	
FORCED CONVECTION	28	BETWEEN	53	AND	17	36.5000	3.0000	
FORCED CONVECTION	28	BETWEEN	53	AND	16	33.0000	7.5000	
FORCED CONVECTION	-28	BETWEEN	53	AND	15	239.6000	12.0000	
FORCED CONVECTION	-28	BETWEEN	53	AND	14	75.6000	12.0000	
FORCED CONVECTION	28	BETWEEN	35	AND	18	34.0000	13.0000	
FORCED CONVECTION	27	BETWEEN	53	AND	13	25.0000	7.5000	
FORCED CONVECTION	28	BETWEEN	54	AND	1	22.5000	20.0000	
FORCED CONVECTION	21	BETWEEN	65	AND	62	2.3800	45.0000	
FORCED CONVECTION	21	BETWEEN	65	AND	29	31.0000	30.0000	
FORCED CONVECTION	22	BETWEEN	42	AND	26	29.2000	15.0000	
FORCED CONVECTION	-22	BETWEEN	42	AND	25	345.0000	14.0000	
FORCED CONVECTION	-24	BETWEEN	41	AND	24	300.4000	14.0000	
FORCED CONVECTION	23	BETWEEN	42	AND	23	25.0000	15.0000	

FOR006

FORCED CONVECTION	23	BETWEEN	40	AND	8	22.5000	20.0000
FORCED CONVECTION	21	BETWEEN	40	AND	28	4.7600	12.0000
FORCED CONVECTION	21	BETWEEN	40	AND	62	4.7600	40.0000
FORCED CONVECTION	26	BETWEEN	63	AND	62	2.3800	145.0000
FORCED CONVECTION	26	BETWEEN	63	AND	31	55.0000	20.0000
FORCED CONVECTION	27	BETWEEN	37	AND	6	45.0000	30.0000
FORCED CONVECTION	27	BETWEEN	34	AND	7	30.0000	20.0000
FORCED CONVECTION	28	BETWEEN	34	AND	30	35.0000	15.0000
FORCED CONVECTION	28	BETWEEN	37	AND	31	60.0000	30.0000
FORCED CONVECTION	28	BETWEEN	49	AND	62	72.0000	15.0000
FORCED CONVECTION	27	BETWEEN	37	AND	51	65.0000	15.0000
FORCED CONVECTION	27	BETWEEN	49	AND	51	80.0000	10.0000
FORCED CONVECTION	27	BETWEEN	49	AND	60	105.0000	6.0000
FORCED CONVECTION	28	BETWEEN	50	AND	62	10.0000	145.0000
FORCED CONVECTION	28	BETWEEN	33	AND	28	31.0000	15.0000
FORCED CONVECTION	28	BETWEEN	56	AND	27	32.5000	3.0000
FORCED CONVECTION	28	BETWEEN	56	AND	26	31.0000	7.5000
FORCED CONVECTION	-28	BETWEEN	56	AND	25	50.0000	14.0000
FORCED CONVECTION	-28	BETWEEN	56	AND	24	11.0000	14.0000
FORCED CONVECTION	27	BETWEEN	56	AND	23	25.0000	7.5000
FORCED CONVECTION	28	BETWEEN	57	AND	8	22.5000	20.0000
FLUID FLOW	41	FROM	66	TO	45	(INDEX 41)	
FLUID FLOW	41	FROM	45	TO	44	(INDEX 41)	
FLUID FLOW	41	FROM	44	TO	43	(INDEX 41)	
FLUID FLOW	41	FROM	43	TO	39	(INDEX 41)	
FLUID FLOW	41	FROM	39	TO	66	(INDEX 41)	
FLUID FLOW	42	FROM	65	TO	42	(INDEX 42)	
FLUID FLOW	42	FROM	41	TO	41	(INDEX 42)	
FLUID FLOW	42	FROM	40	TO	32	(INDEX 42)	
FLUID FLOW	42	FROM	32	TO	65	(INDEX 42)	
FLUID FLOW	43	FROM	64	TO	38	(INDEX 43)	
FLUID FLOW	44	FROM	38	TO	36	(INDEX 44)	
FLUID FLOW	45	FROM	38	TO	46	(INDEX 45)	
FLUID FLOW	44	FROM	46	TO	47	(INDEX 44)	
FLUID FLOW	44	FROM	47	TO	64	(INDEX 43)	
FLUID FLOW	44	FROM	36	TO	35	(INDEX 44)	
FLUID FLOW	44	FROM	35	TO	54	(INDEX 44)	
FLUID FLOW	44	FROM	53	TO	55	(INDEX 44)	
FLUID FLOW	44	FROM	55	TO	64	(INDEX 43)	
FLUID FLOW	47	FROM	63	TO	37	(INDEX 47)	
FLUID FLOW	49	FROM	37	TO	49	(INDEX 49)	
FLUID FLOW	49	FROM	49	TO	50	(INDEX 49)	
FLUID FLOW	49	FROM	50	TO	63	(INDEX 47)	
FLUID FLOW	48	FROM	37	TO	34	(INDEX 48)	
FLUID FLOW	48	FROM	34	TO	33	(INDEX 48)	
FLUID FLOW	48	FROM	33	TO	56	(INDEX 48)	
FLUID FLOW	48	FROM	56	TO	57	(INDEX 48)	
FLUID FLOW	48	FROM	57	TO	58	(INDEX 48)	
FLUID FLOW	48	FROM	58	TO	63	(INDEX 47)	
BEARING CONDUCTION	51	BETWEEN	14	AND	13	1.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	14	AND	16	1.0000	-1.0000
BEARING CONDUCTION	51	BETWEEN	24	AND	23	2.0000	1.0000
BEARING CONDUCTION	51	BETWEEN	24	AND	26	2.0000	-1.0000

TEMPERATURES ARE IN DEGREES CELSIUS. THE FIRST 62 TEMPERATURES ARE CALCULATED, THE OTHERS ARE KNOWN

STEADY STATE TEMPERATURE CALCULATION, INITIAL TEMPERATURES

CALCULATED TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
1	70.000	2	70.000	3	70.000	4	70.000	5	70.000
6	70.000	7	70.000	8	70.000	9	70.000	10	70.000
11	70.000	12	110.000	13	110.000	14	127.000	15	65.000
16	103.000	17	100.000	18	84.000	19	70.000	20	70.000
21	70.000	22	63.000	23	60.000	24	97.000	25	66.000
26	110.000	27	108.000	28	79.000	29	66.000	30	70.000
31	70.000	32	70.000	33	70.000	34	70.000	35	70.000
36	70.000	37	70.000	38	70.000	39	70.000	40	70.000
41	70.000	42	70.000	43	70.000	44	70.000	45	70.000
46	70.000	47	70.000	48	70.000	49	70.000	50	70.000
51	70.000	52	70.000	53	70.000	54	70.000	55	70.000
56	70.000	57	70.000	58	70.000	59	70.000	60	70.000
61	70.000	62	70.000						

KNOWN BOUNDARY TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
63	21.000	64	21.000	65	21.000	66	21.000
						67	246.000

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD,

8 GEOMETRIC SECTIONS 3 LOAD SECTION(S), 2 BEARINGS. MODULUS OF ELASTICITY = 2.041E+05
 PLANE X - Y.

POS- TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT	POS.ERR DEFL/FOR	BEARING SEAT ANG.ERR	DEFL/MOM
1	.0	.0	.0	40.0							
2	40.0	.0	40.0	40.0	448.2	.0	.000	.000	.000	.000	0.00E+00
3	49.0	.0	40.0	50.0							
4	62.0	.0	50.0	60.0							
5	80.0	.0	60.0	202.5							
6	121.0	.0	202.5	202.5	111.2	.0	.000	.000	.000	1.36E-19	1.36E-19
7	162.0	.0	202.5	60.0							
8	182.0	.0	60.0	50.0							
9	195.0	.0	50.0	40.0							
10	202.0	.0	40.0	40.0	667.2	.0	.000	.000	.000	.000	0.00E+00
11	222.0	.0	40.0	.0							

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD,

8 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS. MODULUS OF ELASTICITY = 2.041E+05
 PLANE X - Z.

THRUST LOAD, NEWTONS = -5.501E+03

POS- TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT	POS.ERR DEFL/FOR	BEARING SEAT ANG.ERR	DEFL/MOM
1	.0	.0	.0	40.0							
2	40.0	.0	40.0	40.0	.0	.0	.000	.000	.000	.000	0.00E+00
3	49.0	.0	40.0	50.0							
4	62.0	.0	50.0	60.0							
5	80.0	.0	60.0	202.5							

FOR006

6	162.0	.0	.0	202.5	60.0	6
7	182.0	.0	.0	60.0	50.0	7
8	195.0	.0	.0	50.0	40.0	8
9	202.0	.0	.0	40.0	40.0	9
10	222.0	.0	.0	40.0	40.0	10

***** ERROR MESSAGE FROM THE EQUATION SOLVING ROUTINE, AT ITERATION LOOP 2 *****

THIS IS THE BEST WE CAN DO. IT
MAY BE USABLE.

REL. ACCURACY .001000, ITERATION LIMIT 200 NUMBER OF UNKNOWN 3
ABSOLUTE ACCURACIES

+00 -.3288870E+00 -.77852890E-08 -.20636620E-03
-.18621320E+01

*** P R O G R A M /P573D14 ** GEO. W. HOSANG A L O U E T T E TECHNOLOGY ** P R O G R A M /P573D14 ***

RAMGEN RIG 2 108 ROLL 208 SPLIT I/R BALL HYBRID BRGS REDESIGN 573RAMGENZHT22.DAT

Correct Alpha;Fits OIL 70F MOBIL DTE832 HT MODEL -2HT14.15X.DAT + U/race Lube #1

B E A R I N G S Y S T E M O U T P U T M E T R I C U N I T S

BRG.	LINEAR (MM)	ANGULAR (RADIANS)	DEFLECTIONS	FX	FZ	MY	MZ
1	-.139	1.065E-03	7.831E-07	5.409E-09	3.664E-05	5.994E+03	540.
2	-.139	7.629E-03	9.354E-08	5.412E-09	4.070E-05	.000	678.

REACTION FORCES (N) AND MOMENTS (MM-N)

FATIGUE LIFE (HOURS)

BRG.	O. RACE	I. RACE	BEARING	O. RACE	I. RACE	H/SIGMA
1	1.798E+03	2.864E+03	1.180E+03	2.87	2.50	2.60
2	1.261E+06	1.660E+06	7.730E+05	3.21	5.05	2.72

LUBE-LIFE FACTOR MATERIAL FACTOR

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CELSIUS)

BRG.	SHAFT	I. RING	I. RACE	I. FLNG.	ROL L.	EL.	O. FLNG.	O. RACE	O. RING	HSG.	BULK LUBE
1	69.4	95.5	99.9	68.8	94.1	122.	108.	122.	120.	92.7	28.9
2	80.7	70.7	68.8	68.8	102.	108.	108.	105.	105.	67.0	46.1

B E A R I N G S Y S T E M O U T P U T M E T R I C U N I T S

BRG.	O. RACE	O. FLNG.	I. RACE	I. FLNG.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	658.	.000	1.104E+03	.000	208.	313.	1.179E+03	3.461E+03	790.
2	1.073E+03	.000	356.	.000	259.	195.	640.	2.522E+03	576.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)	STARVATION FACTOR	THERMAL FACTOR	MENISCUS DIST. (MM)	CONDUCTIVITY (W/DEG.C)
1	.327	.401	.844	.406	.736
2	.407	.640	.850	.691	.779

FIT PRESSURES (N/MM2)

BRG.	SHAFT-COLD, OPER.	HSG.-COLD, OPER.	ORIGINAL CHANGE	OPERATING SHAFT-INNER RING (RPM)	SPEED GIVING ZERO FIT PRESSURE
1	34.6	11.0	4.02	5.526E-02	5.601E+04
2	23.0	9.94	.000	3.810E-03	5.941E-03

B E A R I N G S Y S T E M O U T P U T M E T R I C U N I T S

BRG.	LOCATION	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM**3)	KINEMATIC (CS)	DYNAMIC (CP)	PRESSURE COEFFICIENT (MM**2/N)	VISCOSITY
1	OUTER	122.047	.7833	3.634	2.847	.1125E-01	.342

INNER	99.870	.7993	FOR006	4.236	.1247E-01
BULK	28.913	.8504	5.300	36.75	.1875E-01
BRG. 2	108.101	.7934	43.21	3.620	.1199E-01
OUTER	68.844	.8216	10.68	8.772	.1463E-01
INNER	46.130	.8380	21.58	18.08	.1674E-01
BULK					

C A G E D A T A		METRIC UNITS		CAGE SPEED DATA		CALCULATED SPEED		CAGE/EPIC		CAGE/SHAFT	
CAGE RAIL - RING LAND DATA		SEP. FORCE		EPI CYCLIC SPEED		(RAD/SEC)		RATIO		RATIO	
BRG.	(MM-N)	HEAT RATE	(WATTS)	SEP. FORCE	(NEWTONS)	(RAD/SEC)	(RPM)	(RAD/SEC)	(RPM)	CALC/EPIC	CAGE/SHAFT
1	-665.	1.179E+03	2.03	.000	1.834E+03	1.751E+04	1.775E+03	1.695E+04	.968	.405	
2	-341.	640.	.000	.000	1.886E+03	1.801E+04	1.879E+03	1.794E+04	.996	.429	

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH ANGLE (DEG.)	ANGULAR SPEEDS (RAD/SECONDS)			SPEED VECTOR ANGLES (DEGREES)	
	WX	WY	TOTAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-9056.630	-4397.167	92.985	10668.080	1773.853
30.00	-9044.652	-4429.858	99.728	10071.710	1773.989
60.00	-9049.690	-4428.499	107.493	10075.720	1774.716
90.00	-9046.771	-4450.729	128.487	10083.130	1775.199
120.00	-9066.904	-4423.983	148.153	10089.710	1776.328
150.00	-9070.813	-4425.981	157.618	10094.250	1776.450
180.00	-9090.729	-4387.313	174.271	10095.560	1776.747
210.00	-9093.733	-4375.782	167.876	10093.150	1776.165
240.00	-9101.397	-4345.395	145.761	10086.580	1775.694
270.00	-9087.457	-4359.326	118.374	10079.660	1774.633
300.00	-9080.452	-4358.455	105.761	10072.830	1774.234
330.00	-9062.519	-4387.904	95.842	10069.370	1773.707

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 METRIC UNITS

AZIMUTH ANGLE (DEG.)	NORMAL FORCES (NEWTONS)		HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)		
	CAGE	OUTER	INNER	OUTER	INNER	OUTER	INNER		
.00	-1.623	1353.265	1064.342	2484.258	1992.582	.0010	.0002	-21.78	-28.99
30.00	-.891	1338.587	1049.727	2475.243	1983.419	.0010	.0002	-21.76	-29.04
60.00	-.425	1301.654	1012.897	2452.266	1959.946	.0010	.0002	-21.70	-29.25
90.00	-.316	1251.279	962.838	2420.213	1927.111	.0010	.0002	-21.62	-29.52
120.00	-.682	1204.324	916.053	2389.554	1895.378	.0010	.0002	-21.53	-29.85
150.00	-1.425	1170.642	882.853	2367.066	1872.198	.0009	.0002	-21.46	-30.09
180.00	-2.321	1158.436	870.557	2358.811	1863.466	.0009	.0002	-21.44	-30.18
210.00	-3.163	1170.515	882.775	2366.980	1872.144	.0009	.0002	-21.46	-30.09
240.00	-3.693	1205.129	917.152	2390.086	1896.136	.0010	.0002	-21.52	-29.88
270.00	-3.702	1253.002	964.925	2421.324	1928.503	.0010	.0002	-21.61	-29.58
300.00	-3.223	1302.450	1013.945	2452.765	1960.622	.0010	.0002	-21.69	-29.28
330.00	-2.463	1339.449	1050.721	2475.775	1984.045	.0010	.0002	-21.75	-29.07

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH ANGLE (DEG.)	ANGULAR SPEEDS (RAD/SECONDS)			SPEED VECTOR ANGLES (DEGREES)	
	WX	WY	TOTAL	TAN-1(WY/WX)	TAN-1(WZ/WX)
.00	-15468.950	-.310	.000	15468.950	1885.903
25.71	-15470.830	-.277	.000	15470.830	1885.570
51.43	-15224.670	.000	.000	15224.670	1853.816
77.14	-15409.170	.000	.000	15409.170	1890.607
102.86	-15884.020	.000	.000	15884.020	1928.480

AZIMUTH ANGLE (DEG.)	CAGE	NORMAL FORCES (NEWTONS)		HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
		OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
128.57	-15023.110	.000	.000	15023.110	1819.840	180.00	180.00	180.00	
154.29	-15283.870	.000	.000	15283.870	1872.745	180.00	180.00	180.00	
180.00	-15484.780	.000	.000	15484.780	1897.742	180.00	180.00	180.00	
205.71	-15825.090	.000	.000	15825.090	1918.241	180.00	180.00	180.00	
231.43	-15019.580	.000	.000	15019.580	1822.876	180.00	180.00	180.00	
257.14	-15402.180	.000	.000	15402.180	1888.349	180.00	180.00	180.00	
282.86	-15603.010	.000	.000	15603.010	1898.874	180.00	180.00	180.00	
308.57	-15267.770	.000	.000	15267.770	1857.368	180.00	180.00	180.00	
334.29	-15467.820	-.277	.000	15467.820	1885.536	-180.00	180.00	180.00	

FOR006

ROLLING ELEMENT OUTPUT FOR BEARING NUMBER 2 METRIC UNITS

AZIMUTH ANGLE (DEG.)	CAGE	NORMAL FORCES (NEWTONS)		HZ STRESS (N/MM**2)		LOAD RATIO QASP/QTOT		CONTACT ANGLES (DEG.)	
		OUTER	INNER	OUTER	INNER	OUTER	INNER	OUTER	INNER
.00	.422	434.887	313.589	782.724	768.148	.0010	.0000	.00	.00
25.71	-1.777	323.146	201.955	676.171	620.200	.0008	.0000	.00	.00
51.43	1.236	112.414	.000	392.824	.000	.0004	.0000	.00	.00
77.14	3.849	124.669	.000	413.682	.000	.0003	.0000	.00	.00
102.86	-12.342	129.331	.000	421.345	.000	.0003	.0000	.00	.00
128.57	-6.200	113.795	.000	395.228	.000	.0004	.0000	.00	.00
154.29	8.656	115.324	.000	397.872	.000	.0003	.0000	.00	.00
180.00	4.226	123.491	.000	411.719	.000	.0003	.0000	.00	.00
205.71	-8.861	120.961	.000	407.479	.000	.0003	.0000	.00	.00
231.43	-3.096	114.225	.000	395.975	.000	.0004	.0000	.00	.00
257.14	5.806	124.368	.000	413.182	.000	.0003	.0000	.00	.00
282.86	-.119	125.550	.000	415.141	.000	.0003	.0000	.00	.00
308.57	-.159	112.831	.000	393.552	.000	.0004	.0000	.00	.00
334.29	2.747	323.189	201.938	676.214	620.179	.0008	.0000	.00	.00

ROLLING ELEMENT CONTACT DIMENSIONS

MAJOR AND MINOR AXES FOR EACH BEARING, METRIC UNITS

BEARING NUMBER 1		#1 ROLLING ELEMENT AT 6 O'CLOCK			
ELEMENT NUMBER	AZIMUTH ANGLE	CONTACT		CONTACT	
		MAJOR	MINOR	MAJOR	MINOR
1	.00	1.005	.265	1.774	.148
2	30.00	.987	.260	1.734	.144
3	60.00	.962	.254	1.679	.140
4	90.00	.939	.248	1.624	.135
5	120.00	.922	.243	1.585	.132
6	150.00	.916	.241	1.570	.131
7	180.00	.922	.243	1.585	.132
8	210.00	.939	.248	1.624	.135
9	240.00	.962	.254	1.679	.140
10	270.00	.987	.260	1.734	.144
11	300.00	1.005	.265	1.774	.148
12	330.00	.000	.000	.000	.000

BEARING NUMBER 2		#1 ROLLING ELEMENT AT 6 O'CLOCK			
ELEMENT NUMBER	AZIMUTH ANGLE	CONTACT		CONTACT	
		MAJOR	MINOR	MAJOR	MINOR
1	.00	.989	.261	1.742	.145
2	25.71	.976	.258	1.711	.142
3	51.43	.963	.254	1.681	.140
4	77.14	.951	.251	1.652	.138
5	102.86	.941	.248	1.630	.136

6	128.57	.938	FOR006	1.622	.135
7	154.29	7.476	.247	.000	.000
8	180.00	.938	.247	1.622	.135
9	205.71	.941	.248	1.630	.136
10	231.43	.951	.251	1.652	.138
11	257.14	.963	.254	1.681	.140
12	282.86	7.476	.026	.000	.000
13	308.57	.986	.260	1.732	.144
14	334.29	.000	.000	.000	.000

TEMPERATURES ARE IN DEGREES CELSIUS. THE FIRST 62 TEMPERATURES ARE CALCULATED, THE OTHERS ARE KNOWN
 STEADY STATE TEMPERATURE CALCULATION, FINAL RESULT AFTER 149 ITERATIONS

CALCULATED TEMPERATURES					
NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
1	115.438	2	133.644	3	146.421
6	142.814	7	123.473	8	94.507
11	69.435	12	95.635	13	99.998
16	121.125	17	118.822	18	92.073
21	80.680	22	70.771	23	68.882
26	108.046	27	105.397	28	66.960
31	44.640	32	80.116	33	57.929
36	84.273	37	21.000	38	21.000
41	88.043	42	46.150	43	119.264
46	119.055	47	115.971	48	195.477
51	192.662	52	218.510	53	89.710
56	59.050	57	91.512	58	123.044
61	60.049	62	60.270		

KNOWN BOUNDARY TEMPERATURES					
NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
63	21.000	64	21.000	65	21.000
				66	21.000
				67	246.000

RUN COMPLETED ON APR-24-2005 AT 15:29:41.81

1.851	26 0.3	0.57	0.	6.35	E-3379.	.0261	FOR005	GAS (AIR) FLOW INSIDE TUBE @ 80F = 27C
	E-5 1.177	1004.	0.35	50.	E-3106.1	.0288		AIR CONVEC'N ROT SHAFT OD @ 27-66 AVG 46C
	27 0.086	0.7						
	1.105	1005.	27.	1.851	E-566.	2.033		AIR FLOW INSIDE TUBE @ 27-66 AVG 46C
	28 0.3	0.57	0.	52.5	E-351.9	.0288		
	1.105	1005.	27.	1.851	E-566.	2.033		AIR FLOW INSIDE TUBE @ 27-66 AVG 46C
	29 0.3	0.57	0.	7.94	E-3106.1	.0288		
	1.105	1005.	27.	1.851	E-566.	2.033		AIR FLOW INSIDE TUBE @ 27-66 AVG 46C
	41 29.54							
	42 25.25							
	43 18.4							
	44 5.1							
	45 13.3							
	46 0.							
	47 18.4							
	48 10.1							
	49 8.3							
	50 0.							

CONDUCTION PATHS

1	11	110.	20.	30.5
1	1	210.	20.	15.5
1	1	315.	30.	14.
1	2	429.	58.	20.5
1	48	5272.75	29.5	35.
1	1	529.	58.	31.
1	67	1097.	62.	3.
1	10	994.	62.	3.
1	9	6091.	62.	16.5
1	60	5287.5	62.	60.
1	48	358.	2.	60.
1	51	658.	2.	72.75
1	52	458.	31.	72.75
1	52	558.	31.	72.75
1	5	629.	58.	20.5
1	1	715.	30.	14.
1	6	5272.75	29.5	35.
1	51	810.	20.	15.5
1	7	2110.	20.	20.
1	1	1220.	18.	22.
1	11	1325.	18.	2.5
1	12	122.	5.	14.
1	12	1736.	18.	2.5
1	16	1840.	18.	15.
1	17	1943.5	7.	62.
1	18	2043.	6.	30.
1	18	1736.	5.	27.
1	19	1737.5	5.	62.
1	61	1847.	40.	70.
1	1	1947.	106.	30.
1	61	2060.	25.	80.
1	1	5934.	8.	20.
1	20	2220.	15.	21.5
1	21	2323.	15.	2.5
1	22	821.5	3.	12.5
1	22	2731.	15.	2.5
1	26	2834.	15.	18.5
1	1	2940.	16.5	8.5
1	28	2731.5	3.	50.
1	29	2947.	106.	30.
1	62	2847.	40.	70.

FOR005

1	28	3140.	13.	30.
1	28	3131.5	3.	34.
1	62	3160.	25.	80.
1	31	3034.	8.	25.
8	43	5434.5	25.	12.
8	40	5734.5	25.	12.
21	66	1936.	103.	
21	66	612.38	45.	
22	45	1115.	5.	
22	45	1114.364	20.	
22	45	1119.	2.394	
22	45	1220.	1.57	
22	45	1114.364	14.	
22	45	129.14	14.	
22	45	1119.5	2.394	
22	45	1220.	1.52	
22	45	1218.35	3.65	
22	44	1633.	18.	
22	44	1541.0	12.	
22	44	14506.7	12.	
23	44	1325.	18.	
23	43	122.5	20.	
21	43	184.76	12.	
21	43	614.76	40.	
25	64	612.38	145.	
25	64	2060.	25.	
28	38	2060.	30.	
28	46	6172.	15.	
27	38	345.	30.	
27	38	4865.	15.	
27	46	4880.	10.	
28	47	6110.	145.	
27	36	230.	20.	
28	36	5935.	15.	
28	53	1736.5	3.	
28	53	1633.	7.5	
28	53	15239.6	12.	
28	53	1475.6	12.	
28	53	1834.	13.	
27	53	1325.	7.5	
28	54	122.5	20.	
21	65	622.38	45.	
21	65	2931.	30.	
22	42	2629.2	15.	
22	42	25345.	14.	
22	41	24300.4	14.	
23	42	2325.	15.	
23	40	822.5	20.	
21	40	284.76	12.	
21	40	624.76	40.	
26	63	622.38	145.	
26	63	3155.	20.	
27	37	645.	30.	
27	34	730.	20.	
28	34	3035.	15.	
28	37	3160.	30.	
28	49	6272.	15.	
27	37	5165.	15.	
27	49	5180.	10.	
27	49	60105.	6.	
28	50	6210.	145.	

LUBE FLOW THRU BEARING CAVITY

On shaft under nut projection
In 6 shaft grooves .03" rad under nut
On shaft in annulus shaft-nut-brg I/R
On corner of brg I/R in annulus
In 6 shaft grooves .03" rad under brg I/R
On brg I/R along 6 shaft grooves
In circumf'l groove .03" rad brg I/R split
On brg I/R ID at circumf'l groove at split
In 10 rad grooves .023" rad I/R split face

- THRU BRG #1

- THRU BRG #2

?? COOLING AIRFLOW THRU BEARING CAVITY

- THRU BRG #1

FOR006

RUN ON APR-24-2005 AT 19:59:58.41

*** P R O G R A M /P573D14 ** GEO. W. HOSANG A L O U E T T E TECHNOLOGY ** P R O G R A M /P573D14 ***

RAMGEN RIG 2 108 ROLL 208 SPLIT I/R BALL HYBRID BRGS REDESIGN 573RAMGEN2HT22.DAT

Correct Alpha;Fits OIL 70F MOBIL DTE832 HT MODEL -2HT14,15X.DAT + U/race Lube #1

UNLESS OTHERWISE STATED, LINEAR DIMENSIONS ARE SPECIFIE IN MILLIMETERS, TEMPERATURES IN DEGREES CELSIUS, FORCES IN NEWTONS, WEIGHTS IN KILOGRAMS, PRESSURES AND ELASTIC MODULI IN NEWTONS PER SQUARE MILLIMETER, ANGLES AND SLOPES IN DEGREES, SURFACE ROUGHNESS IN MICRONS, SPEEDS IN REVOLUTIONS PER MINUTE, DENSITY IN GRAMS PER CUBIC CENTIMETER, KINEMATIC VISCOSITY IN CENTISTOKES AND THERMAL CONDUCTIVITY IN WATTS PER METER-DEGREE CELSIUS. REF AFAPL-TR-76-89 & -90, SKF AL79P007L, SOLAR SR84-M-2134-00.

THIS DATA SET CONTAINS 2 BEARINGS
BEARING NO. (1) -

BALL BEARING

BEARING NO. (2) - CYLINDRICAL ROLLER BEARING

THE MAXIMUM NUMBER OF MAIN LOOP ITERATIONS ALLOWED IS 61 AND THE RELATIVE ACCURACY REQUIRED IS .00010

MAX ITERATION NUMBER IN A CALL BEAR STATEMENT IN SHAPE =150

MAX ITERATION NUMBER IN A CALL BEAR STATEMENT IN SONR1 =200

SOLUTION LEVEL =2

SURFACE FINISH CONVERSION FACTOR (FMULT) = 1.111

Y.M. CORRECTION FLAG (EPFLAG) = .10E+01

B E A R I N G D A T A

BEARING NUMBER	NUMBER OF ROLLING ELEMENTS	AZIMUTH ANGLE ORIENTATION	PITCH DIAMETER	DIAMETRAL CLEARANCE	CONTACT ANGLE	INNER RING SPEED	OUTER RING SPEED
1	12	.000	60.000	.08503	-17.300	46200.	0.
2	14	.000	54.000	.00381	.000	46200.	0.

C A G E D A T A

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL-LAND WIDTH	RAIL-LAND DIAMETER	RAIL-LAND CLEARANCE	WEIGHT
1	OUTER RING LAND RIDING	.381000	3.0000	65.0000	.254	.004000
2	OUTER RING LAND RIDING	.317500	3.0000	60.0000	.203	.003000

S T E E L D A T A

* NOTE * FOR HEAT TRANSFER CASES, LIFE FACTOR(S) MAY NEED MANUAL ITERATION BASED ON COMPUTED TEMPERATURE(S)

BRG.NO.	INNER RING TYPE	LIFE FACTOR	OUTER RING TYPE	LIFE FACTOR
1	CRONIDUR 30	15.000	CRONIDUR 30	15.000
2	CRONIDUR 30	15.000	CRONIDUR 30	15.000

R O L L I N G E L E M E N T D A T A
BEARING NUMBER (1) TYPE - BALL BEARING

BALL DIAMETER OUTER RACEWAY CURVATURE INNER RACEWAY CURVATURE

12.7000
BEARING NUMBER (2) TYPE - CYLINDRICAL ROLLER BEARING .561
FOR006
.513

ROLLER DIAMETER	ROLLER LENGTH	ROLLER END SPHERE RADIUS	ROLLER INCL. ANGLE	AXIAL PLAY INNER RING	OUTER RING	FLANGE ANGLE INNER RING
7.5000	9.0000	.0000	.000	.0000	.0000	.000

EFF.LENGTH	OUTER RACEWAY FLAT LENGTH	CROWN RAD.	EFF.LENGTH	INNER RACEWAY FLAT LENGTH	CROWN RAD.	NO. OF AXIAL LAMINAE
7.4760	4.4450	254.000	7.4760	4.4450	254.000	2

S U R F A C E D A T A

BEARING NUMBER	OUTER RACEWAY FLAT LENGTH	CLA ROUGHNESS INNER	ROLL. ELM. .0127 .0508	OUTER RACEWAY FLAT LENGTH	RMS ASPERITY SLOPE INNER	ROLL. ELEM. 1.500 1.500
1	.1020	.1020	.0127	1.500	1.500	1.500
2	.1020	.1020	.0508	1.500	1.500	1.500

L U B R I C A N T D A T A

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MOBIL DTE 832	29.60	.8600	7.20E-04	.130
2	MOBIL DTE 832	29.60	.8600	7.20E-04	.130

BEARING NUMBER	DESIGNATION	WALTHER EQUATION A	WALTHER EQUATION B	AUX TEMP COEFF BETA
1	MOBIL DTE 832	10.015990	3.582929	.015952
2	MOBIL DTE 832	10.015990	3.582929	.015952

L U B R I C A T I O N A N D F R I C T I O N D A T A

BEARING NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELEM.+ RACEWAY) OUTER	INNER	ASPHERITY FRICTION COEFFICIENT
1	.50	.1000E-02	.9000E-03	.10
2	.50	.1000E-02	.9000E-03	.10

F I T D A T A A N D M A T E R I A L P R O P E R T I E S

BEARING NUMBER	COLD FITS SHAFT	(MM TIGHT) HOUSING	SHAFT	INNER RING	OUTER RING	HOUSING
1	.0356	.0089	36.0000	18.0000	18.0000	36.0000
2	.0279	-.0318	30.0000	15.0000	15.0000	30.0000

BEARING NUMBER	SHAFT I.D.	BEARING BORE	EFFECTIVE DIAMETERS INNER RING AVE. O.D.	OUTER RING AVE. I.D.	BEARING O.D.	HOUSING O.D.
1	.000	40.000	50.000	66.000	80.000	190.500
2	.000	40.000	48.000	64.000	68.000	190.500

BEARING NUMBER (1)	SHAFT	INNER RING	ROLL. ELEM.	OUTER RING	HOUSING
1	113763.0	213734.0	310275.0	213734.0	193050.0
2	.3200	.3000	.2600	.3000	.2700

WEIGHT DENSITY
COEFF. OF THERMAL EXP. 4.540 7.806 FOR006
.00000770 .00001053 3.190
.00000200 7.778
.00001080

BEARING NUMBER (2) SHAFT INNER RING ROLL. ELEM. OUTER RING HOUSING
MODULUS OF ELASTICITY 113763.0 213734.0 310275.0 213734.0 193050.0
POISSONS RATIO .3200 .3000 .2600 .3000 .2700
WEIGHT DENSITY 4.540 7.806 3.190 7.806 7.778
COEFF. OF THERMAL EXP. .00000770 .00001053 .00000200 .00001053 .00001080

STEADY STATE TEMPERATURE CALCULATION. ITERATION LIMIT 155, ABSOLUTE ACCURACY 2.00 DEGREES
UNLESS OTHERWISE STATED, INTERNATIONAL UNITS ARE USED

NODE POINTERS
BRG SHAFT I. RING I. RACE ROLL EL. O. RACE O. RING HOUSING BULK FLANGE
1 11 12 13 14 16 17 18 17
2 21 22 23 24 26 27 28 27
BRG INNER RACE OUTER RACE CAGE DRAG
1 13 14 16 15 16 45 14
2 23 24 26 25 26 50 24
FLANGE
17
27

NODES WHERE BEARING HEAT IS GENERATED

HEAT TRANSFER COEFFICIENTS
INDEX

CONDUCTION 1 46.7000
CONDUCTION 2 55.8000
CONDUCTION 3 27.7000
CONDUCTION 4 17.3000
CONDUCTION 5 26.0000
CONDUCTION 6 34.3000
CONDUCTION 7 2500.00
CONDUCTION 8 1500.00
CONDUCTION 9 500.000
CONDUCTION 10 17.4000

HEAT TRANSFER COEFFICIENTS
INDEX

FOR006

FORCED CONVECTION	21	.620000 .134000 .000000 .000000	.333000 .406000E-01 .000000 .000000	.333000 845.000 .000000 .000000	.476300E-02 1913.00 401.372	.212000 .000000 .000000
FORCED CONVECTION	22	.600000 .133100 .406000E-01 -3.55741	.500000 .000000 66.0000 .830000E-03	.310000 830.000 .982000E-02 -.190000	.540000E-01 1985.00 1403.03	51.9000 27.0000 .253808E+10
FORCED CONVECTION	23	.600000 .133100 .406000E-01 -3.55741	.500000 .000000 66.0000 .830000E-03	.310000 830.000 .982000E-02 -.190000	.540000E-01 1985.00 2006.04	106.100 27.0000 .253808E+10
FORCED CONVECTION	24	.600000 .133100 .406000E-01 -3.55741	.500000 .000000 66.0000 .830000E-03	.310000 830.000 .982000E-02 -.190000	.794000E-02 1985.00 5231.50	106.100 27.0000 .253808E+10
FORCED CONVECTION	25	.300000 .261000E-01 .000000 .000000	.570000 .185100E-04 .000000 .000000	.000000 1.17700 .000000 .000000	.635000E-02 1004.00 1054.50	345.000 .000000 .000000
FORCED CONVECTION	26	.300000 .261000E-01 .000000 .000000	.570000 .185100E-04 .000000 .000000	.000000 1.17700 .000000 .000000	.635000E-02 1004.00 1112.54	379.000 .000000 .000000
FORCED CONVECTION	27	.860000E-01 .288000E-01 .185100E-04 .254215	.700000 .000000 66.0000 .110500E-05	.350000 1.10500 .203300E-04 -.350000	.500000E-01 1005.00 6.64546	106.100 27.0000 .670648
FORCED CONVECTION	28	.300000 .288000E-01 .185100E-04 .254215	.570000 .000000 66.0000 .110500E-05	.000000 1.10500 .203300E-04 -.570000	.525000E-01 1005.00 .308466	51.9000 27.0000 .670648
FORCED CONVECTION	29	.300000 .288000E-01 .185100E-04 .254215	.570000 .000000 66.0000 .110500E-05	.000000 1.10500 .203300E-04 -.570000	.794000E-02 1005.00 1.04464	106.100 27.0000 .670648

H E A T T R A N S F E R C O E F F I C I E N T S
INDEX

FLUID FLOW 41

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FOR006

29.5400

FLUID FLOW

42

25.2500

FLUID FLOW

43

18.4000

FLUID FLOW

44

5.10000

FLUID FLOW

45

13.3000

FLUID FLOW

46

.000000

FLUID FLOW

47

18.4000

FLUID FLOW

48

10.1000

FLUID FLOW

49

8.30000

FLUID FLOW

50

.000000

3A-398

DESCRIPTION OF THE GEOMETRY AND INDICATION OF THE TYPES AND PATHS OF HEAT TRANSFER
ALL LENGTHS ARE IN MILLIMETERS, A NEGATIVE SIGN OF THE INDEX MEANS NO ROTATIONAL SYMMETRY

TYPE OF HEAT TR.	INDEX	NODE	NODE	1ST LENGTH	2ND LENGTH	3RD LENGTH
CONDUCTION	1	BETWEEN	11	10.0000	20.0000	30.5000
CONDUCTION	1	BETWEEN	1	10.0000	20.0000	15.5000
CONDUCTION	1	BETWEEN	2	15.0000	30.0000	14.0000
CONDUCTION	1	BETWEEN	3	29.0000	58.0000	20.5000
CONDUCTION	1	BETWEEN	48	72.7500	29.5000	35.0000
CONDUCTION	1	BETWEEN	4	29.0000	58.0000	31.0000
CONDUCTION	1	BETWEEN	67	97.0000	62.0000	3.0000
CONDUCTION	1	BETWEEN	10	94.0000	62.0000	3.0000
CONDUCTION	1	BETWEEN	9	91.0000	62.0000	3.0000
CONDUCTION	1	BETWEEN	60	87.5000	62.0000	16.5000
CONDUCTION	1	BETWEEN	48	58.0000	2.0000	60.0000
CONDUCTION	1	BETWEEN	51	58.0000	2.0000	60.0000
CONDUCTION	1	BETWEEN	52	58.0000	31.0000	72.7500
CONDUCTION	1	BETWEEN	52	58.0000	31.0000	72.7500
CONDUCTION	1	BETWEEN	5	29.0000	58.0000	20.5000
CONDUCTION	1	BETWEEN	6	15.0000	30.0000	14.0000
CONDUCTION	1	BETWEEN	51	72.7500	29.5000	35.0000
CONDUCTION	1	BETWEEN	7	10.0000	20.0000	15.5000
CONDUCTION	1	BETWEEN	8	10.0000	20.0000	20.0000
CONDUCTION	1	BETWEEN	11	20.0000	18.0000	22.0000
CONDUCTION	1	BETWEEN	12	25.0000	18.0000	2.5000
CONDUCTION	1	BETWEEN	12	22.0000	5.0000	14.0000
CONDUCTION	1	BETWEEN	16	36.0000	18.0000	2.5000
CONDUCTION	1	BETWEEN	17	40.0000	18.0000	15.0000
CONDUCTION	1	BETWEEN	18	43.5000	7.0000	62.0000
CONDUCTION	1	BETWEEN	18	43.0000	6.0000	30.0000
CONDUCTION	1	BETWEEN	18	36.0000	5.0000	27.0000

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CONDUCTION	1	BETWEEN	19	AND	17	37.5000	5.0000	62.0000
CONDUCTION	1	BETWEEN	61	AND	18	47.0000	40.0000	70.0000
CONDUCTION	1	BETWEEN	61	AND	19	47.0000	106.0000	30.0000
CONDUCTION	1	BETWEEN	61	AND	20	60.0000	25.0000	80.0000
CONDUCTION	1	BETWEEN	20	AND	59	34.0000	8.0000	20.0000
CONDUCTION	1	BETWEEN	21	AND	22	20.0000	15.0000	21.5000
CONDUCTION	1	BETWEEN	22	AND	23	23.0000	15.0000	12.5000
CONDUCTION	1	BETWEEN	22	AND	8	21.5000	3.0000	2.5000
CONDUCTION	1	BETWEEN	26	AND	27	31.0000	15.0000	18.5000
CONDUCTION	1	BETWEEN	27	AND	28	34.0000	15.0000	16.5000
CONDUCTION	1	BETWEEN	28	AND	29	40.0000	16.5000	8.5000
CONDUCTION	1	BETWEEN	29	AND	27	31.5000	3.0000	50.0000
CONDUCTION	1	BETWEEN	62	AND	29	47.0000	106.0000	30.0000
CONDUCTION	1	BETWEEN	62	AND	28	47.0000	40.0000	70.0000
CONDUCTION	1	BETWEEN	28	AND	31	40.0000	13.0000	30.0000
CONDUCTION	1	BETWEEN	28	AND	31	31.5000	3.0000	34.0000
CONDUCTION	1	BETWEEN	62	AND	31	60.0000	25.0000	80.0000
CONDUCTION	1	BETWEEN	31	AND	30	34.0000	8.0000	25.0000
CONDUCTION	1	BETWEEN	43	AND	54	34.5000	25.0000	12.0000
CONDUCTION	8	BETWEEN	40	AND	57	34.5000	25.0000	12.0000
CONDUCTION	8	BETWEEN	66	AND	19	36.0000	103.0000	12.0000
FORCED CONVECTION	21	BETWEEN	66	AND	61	2.3800	45.0000	
FORCED CONVECTION	21	BETWEEN	66	AND	61	15.0000	5.0000	
FORCED CONVECTION	22	BETWEEN	45	AND	11	14.3640	20.0000	
FORCED CONVECTION	-22	BETWEEN	45	AND	11	19.0000	2.3940	
FORCED CONVECTION	22	BETWEEN	45	AND	11	20.0000	1.5700	
FORCED CONVECTION	-22	BETWEEN	45	AND	12	14.3640	14.0000	
FORCED CONVECTION	-22	BETWEEN	45	AND	12	9.1400	14.0000	
FORCED CONVECTION	22	BETWEEN	45	AND	11	19.5000	2.3940	
FORCED CONVECTION	22	BETWEEN	45	AND	12	20.0000	1.5200	
FORCED CONVECTION	-22	BETWEEN	45	AND	12	18.3500	3.6500	
FORCED CONVECTION	22	BETWEEN	44	AND	16	33.0000	18.0000	
FORCED CONVECTION	-22	BETWEEN	44	AND	15	541.0000	12.0000	
FORCED CONVECTION	-24	BETWEEN	44	AND	14	506.7000	12.0000	
FORCED CONVECTION	23	BETWEEN	44	AND	13	25.0000	18.0000	
FORCED CONVECTION	23	BETWEEN	43	AND	1	22.5000	20.0000	
FORCED CONVECTION	21	BETWEEN	43	AND	18	4.7600	12.0000	
FORCED CONVECTION	21	BETWEEN	43	AND	61	4.7600	40.0000	
FORCED CONVECTION	25	BETWEEN	64	AND	61	2.3800	145.0000	
FORCED CONVECTION	25	BETWEEN	64	AND	20	60.0000	25.0000	
FORCED CONVECTION	28	BETWEEN	38	AND	20	60.0000	30.0000	
FORCED CONVECTION	27	BETWEEN	38	AND	3	45.0000	30.0000	
FORCED CONVECTION	27	BETWEEN	38	AND	48	65.0000	15.0000	
FORCED CONVECTION	27	BETWEEN	46	AND	48	80.0000	10.0000	
FORCED CONVECTION	28	BETWEEN	47	AND	61	10.0000	145.0000	
FORCED CONVECTION	27	BETWEEN	36	AND	2	30.0000	20.0000	
FORCED CONVECTION	28	BETWEEN	53	AND	17	35.0000	15.0000	
FORCED CONVECTION	28	BETWEEN	53	AND	16	36.5000	3.0000	
FORCED CONVECTION	-28	BETWEEN	53	AND	15	33.0000	7.5000	
FORCED CONVECTION	-28	BETWEEN	53	AND	15	239.6000	12.0000	
FORCED CONVECTION	28	BETWEEN	53	AND	14	75.6000	12.0000	
FORCED CONVECTION	28	BETWEEN	35	AND	18	34.0000	13.0000	
FORCED CONVECTION	27	BETWEEN	54	AND	13	25.0000	7.5000	
FORCED CONVECTION	21	BETWEEN	65	AND	1	22.5000	20.0000	
FORCED CONVECTION	21	BETWEEN	65	AND	62	2.3800	45.0000	
FORCED CONVECTION	22	BETWEEN	42	AND	26	31.0000	30.0000	
FORCED CONVECTION	-22	BETWEEN	42	AND	25	29.2000	15.0000	
FORCED CONVECTION	-24	BETWEEN	41	AND	24	345.0000	14.0000	
FORCED CONVECTION	23	BETWEEN	42	AND	23	300.4000	14.0000	
						25.0000	15.0000	

CALCULATED TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
1	70.000	2	70.000	3	70.000	4	70.000	5	70.000		
6	70.000	7	70.000	8	70.000	9	70.000	10	70.000		
11	70.000	12	110.000	13	110.000	14	127.000	15	65.000		
16	103.000	17	100.000	18	84.000	19	70.000	20	70.000		
21	70.000	22	63.000	23	60.000	24	97.000	25	66.000		
26	110.000	27	108.000	28	79.000	29	66.000	30	70.000		
31	70.000	32	70.000	33	70.000	34	70.000	35	70.000		
36	70.000	37	70.000	38	70.000	39	70.000	40	70.000		
41	70.000	42	70.000	43	70.000	44	70.000	45	70.000		
46	70.000	47	70.000	48	70.000	49	70.000	50	70.000		
51	70.000	52	70.000	53	70.000	54	70.000	55	70.000		
56	70.000	57	70.000	58	70.000	59	70.000	60	70.000		
61	70.000	62	70.000								

KNOWN BOUNDARY TEMPERATURES

NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE	NODE	TEMPERATURE
63	21.000	64	21.000	65	21.000	66	21.000	67	246.000

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD,

8 GEOMETRIC SECTIONS 3 LOAD SECTION(S), 2 BEARINGS. MODULUS OF ELASTICITY = 2.041E+05
 PLANE X - Y.

POS- TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT	BEARING SEAT DEFL/FOR	ANG.ERR	DEFL/MOM
1	.0	.0	.0	40.0	778.4	.0	.000	.000	.000	0.00E+00	.0000
2	40.0	.0	40.0	40.0							
3	49.0	.0	40.0	50.0							
4	62.0	.0	50.0	60.0							
5	80.0	.0	60.0	202.5							
6	121.0	.0	202.5	202.5	111.2	.0	.000	.000	.000	1.36E-19	.0000
7	162.0	.0	202.5	60.0							
8	182.0	.0	60.0	50.0							
9	195.0	.0	50.0	40.0							
10	202.0	.0	40.0	40.0	3114.0	.0	.000	.000	.000	0.00E+00	.0000
11	222.0	.0	40.0	.0							

SHAFT GEOMETRY, BEARING LOCATIONS AND SHAFT LOAD,

8 GEOMETRIC SECTIONS 1 LOAD SECTION(S), 2 BEARINGS. MODULUS OF ELASTICITY = 2.041E+05
 PLANE X - Z.

THRUST LOAD, NEWTONS = -1.140E+04

POS- TION	INNER DIAM. LEFT	INNER DIAM. RIGHT	OUTER DIAM. LEFT	OUTER DIAM. RIGHT	POINT FORCE	POINT MOMENT	LOAD INTENSITY LEFT	LOAD INTENSITY RIGHT	BEARING SEAT DEFL/FOR	ANG.ERR	DEFL/MOM
1	.0	.0	.0	40.0	.0	.0	.000	.000	.000	0.00E+00	.0000
2	40.0	.0	40.0	40.0							
3	49.0	.0	40.0	50.0							
4	62.0	.0	50.0	60.0							
5	80.0	.0	60.0	202.5							

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6	162.0	.0	.0	202.5	60.0	6
7	182.0	.0	.0	60.0	50.0	7
8	195.0	.0	.0	50.0	40.0	8
9	202.0	.0	.0	40.0	40.0	9
10	222.0	.0	.0	40.0	40.0	10

***** ERROR MESSAGE FROM THE EQUATION SOLVING ROUTINE, AT ITERATION LOOP 2 *****

THIS IS THE BEST WE CAN DO. IT
MAY BE USABLE.
90468260E+00 -.18111590E-07 .15121700E-03
-.30120580E+01

*** P R O G R A M /P573D14 ** GEO. W. HOSANG A L O U E T T E TECHNOLOGY ** P R O G R A M /P573D14 ***

RAMGEN RIG 2 108 ROLL 208 SPLIT I/R BALL HYBRID BRGS REDESIGN 573RAMGEN2HT22.DAT

Correct Alpha;Fits OIL 70F MOBIL DTE832 HT MODEL -2HT14,15X.DAT + U/race Lube #1

B E A R I N G S Y S T E M O U T P U T M E T R I C U N I T S

BRG.	DX	DY	DZ	GY	GZ	FX	FY	FZ	MY	MZ
1	-1.175	1.208E-03	7.967E-08	7.009E-10	5.553E-05	-1.201E+04	915.	-1.33	6.60	1.463E+04
2	-1.175	1.158E-02	3.438E-08	7.045E-10	6.486E-05	.000	3.077E+03	6.33	-1.73	126.

FATIGUE LIFE (HOURS)

BRG.	O. RACE	I. RACE	BEARING	H/SIGMA	LUBE-LIFE FACTOR	MATERIAL FACTOR
1	270.	368.	167.	2.03	2.52	2.39
2	1.486E+04	7.618E+03	5.404E+03	2.65	3.30	2.58

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES CELSIUS)

BRG.	SHAFT	I. RING	I. RACE	I. FLNG.	ROL L.	EL.	O. FLNG.	O. RACE	O. RING	HSG.	BULK LUBE
1	83.7	121.	127.	127.	122.	146.	146.	146.	142.	108.	32.1
2	92.4	83.8	82.1	82.1	115.	115.	115.	115.	112.	70.2	50.0

B E A R I N G S Y S T E M O U T P U T M E T R I C U N I T S

BRG.	O. RACE	O. FLNGS.	I. RACE	I. FLNGS.	R.E.DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	1.255E+03	.000	1.612E+03	.000	294.	281.	1.261E+03	4.704E+03	972.
2	1.194E+03	.000	587.	.000	351.	163.	684.	2.978E+03	616.

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

BRG.	FILM (MICRONS)	STARVATION FACTOR	THERMAL FACTOR	MENISCUS DIST. (MM)	CONDUCTIVITY (W/DEG.C)
1	.232	.288	.809	.412	.763
2	.336	.418	.828	.652	.811

FIT PRESSURES (N/MM2)

BRG.	SHAFT-COLD, OPER.	HSG.-COLD, OPER.	ORIGINAL CHANGE	OPERATING SHAFT-INNER RING	SPEED GIVING ZERO FIT PRESSURE
1	34.6	7.26	4.02	8.118E-02-5.079E-03-7.995E-02	5.520E+04
2	23.0	6.02	.000	5.430E-02 3.810E-03 4.360E-03	8.170E-03 5.571E+04

BEARING CLEARANCES (MM)

BRG.	TEMPERATURES (DEGREES C.)	DENSITY (GM/CM**3)	KINEMATIC (CS)	DYNAMIC (CP)	COEFFICIENT (MM**2/N)	PRESSURE VISCOSITY
1	146.386	.7658	2.606	1.996	.1011E-01	.1099E-01
2	127.244	.7796	3.364	2.622	.1099E-01	.1835E-01

C A G E D A T A M E T R I C U N I T S									
CAGE RAIL - RING LAND DATA									
BRG.	TORQUE (MM-N)	HEAT RATE (WATTS)	SEP.FORCE (NEWTONS)	ECCENTRICITY RATIO	CAGE SPEED DATA EPI CYCLIC SPEED (RPM)	CALCULATED SPEED (RAD/SEC)	CALC/EPIC RATIO	CAGE/SHAFT RATIO	
1	-639.	114.889	.000	.000	2.033E+03	1.941E+04	1.973E+03	1.884E+04	.408
2	-329.	82.100	.000	.000	2.083E+03	1.989E+04	2.083E+03	1.989E+04	.431
R O L L I N G E L E M E N T O U T P U T									
FOR BEARING NUMBER 1 METRIC UNITS									
AZIMUTH ANGLE (DEG.)									
ANGULAR SPEEDS (RAD/SEC)									
WZ TOTAL									
WY									
WX									
SPEED VECTOR ANGLES (DEGREES)									
TAN-1(WY/WX)									
TAN-1(WZ/WX)									
R O L L I N G E L E M E N T O U T P U T									
FOR BEARING NUMBER 1 METRIC UNITS									
AZIMUTH ANGLE (DEG.)									
ANGULAR SPEEDS (RAD/SEC)									
WZ TOTAL									
WY									
WX									
SPEED VECTOR ANGLES (DEGREES)									
TAN-1(WY/WX)									
TAN-1(WZ/WX)									
R O L L I N G E L E M E N T O U T P U T									
FOR BEARING NUMBER 1 METRIC UNITS									
AZIMUTH ANGLE (DEG.)									
NORMAL FORCES (NEWTONS)									
CAGE									
OUTER									
INNER									
HZ STRESS (N/MM**2)									
OUTER									
INNER									
LOAD RATIO QASP/QTOT									
OUTER									
INNER									
CONTACT ANGLES (DEG.)									
OUTER									
INNER									

FOR006			
ROLLING ELEMENT	OUTPUT	FOR BEARING NUMBER 2	METRIC UNITS
180.00	-17077.450	.000	17077.450
205.71	-17081.710	.000	2082.565
231.43	-17080.370	.000	180.00
257.14	-17084.360	.000	180.00
282.86	-17078.560	.000	180.00
308.57	-17081.020	.000	180.00
334.29	-17081.020	.000	180.00
			180.00

FOR006			
ROLLING ELEMENT	OUTPUT	FOR BEARING NUMBER 2	METRIC UNITS
AZIMUTH ANGLE (DEG.)	NORMAL FORCES (NEWTONS)	HZ STRESS (N/MM**2)	LOAD RATIO QASP/QTOT
	CAGE	OUTER	INNER
25.71	.000	1209.972	1061.617
51.43	-.010	1018.491	870.642
77.14	-.020	506.167	357.791
102.86	.033	147.456	.000
128.57	.048	145.317	.000
154.29	.001	152.719	.000
180.00	-.032	149.954	.000
205.71	.000	148.301	.000
231.43	.043	149.944	.000
257.14	.029	152.712	.000
282.86	-.019	145.325	.000
308.57	-.024	147.466	.000
334.29	.019	506.167	357.808
	.010	1018.491	870.633

FOR006			
ROLLING ELEMENT	CONTACT	DIMENSIONS	
AZIMUTH ANGLE	OUTER	MINOR	MAJOR
25.71	.000	1.203	1.184
51.43	30.00	1.184	1.158
77.14	60.00	1.158	1.133
102.86	90.00	1.133	1.114
128.57	120.00	1.114	1.107
154.29	150.00	1.107	1.114
180.00	180.00	1.114	1.133
205.71	210.00	1.133	1.158
231.43	240.00	1.158	1.184
257.14	270.00	1.184	1.203
282.86	300.00	1.203	.000
308.57	330.00	.000	.000

FOR006			
ROLLING ELEMENT	CONTACT	DIMENSIONS	
AZIMUTH ANGLE	OUTER	MINOR	MAJOR
25.71	.000	1.203	1.184
51.43	30.00	1.184	1.158
77.14	60.00	1.158	1.133
102.86	90.00	1.133	1.114
128.57	120.00	1.114	1.107
154.29	150.00	1.107	1.114
180.00	180.00	1.114	1.133
205.71	210.00	1.133	1.158
231.43	240.00	1.158	1.184
257.14	270.00	1.184	1.203
282.86	300.00	1.203	.000
308.57	330.00	.000	.000

FOR006			
ROLLING ELEMENT	CONTACT	DIMENSIONS	
AZIMUTH ANGLE	OUTER	MINOR	MAJOR
25.71	.000	1.203	1.184
51.43	30.00	1.184	1.158
77.14	60.00	1.158	1.133
102.86	90.00	1.133	1.114
128.57	120.00	1.114	1.107
154.29	150.00	1.107	1.114
180.00	180.00	1.114	1.133
205.71	210.00	1.133	1.158
231.43	240.00	1.158	1.184
257.14	270.00	1.184	1.203
282.86	300.00	1.203	.000
308.57	330.00	.000	.000

Introduction

Ramgen is currently designing a new generation Rampressor following a successful testing of the first unit. As part of the design process, BSi has been asked to analyze the rotor bearing system of the Rampressor 2 and the entire Test rig for testing the Rampressor 2. The final configuration of the rampressor and rig components as defined in an September 12, 2005 email from Ram Pudupatty of Ramgen, is specifically addressed in this report.

Rig Definition

The complete rig comprises the Rampressor, an Air Turbine drive supplied by BSi, and a Flex Coupling supplied by John Crane Flexibox. The Dyrobes model is illustrated in Figure 1 below.

Figure 1 - Dyrobes Model of Rampressor Test Rig

Rampressor Definition

In addition to the geometry and mass properties supplied in the Ramgen specification, imbalance levels of 0.22 gm-in were specified for both balance planes of the rampressor. For the coupling, imbalance levels of 0.166 gm-in were employed at both hubs.

A coupling stiffness of 130 in.lbf / deg at each joint was specified by Bill Kilty of Flexibox and this value was used in the analysis. A total of nine different Rampressor ball and roller bearing stiffness combinations were supplied by Ramgen for analyses. The matrix is illustrated below.

<u>Case</u>	<u>Ball Bearing Stiffness (lbf / in)</u>	<u>Roller Bearing Stiffness (lbf / in)</u>
1	1.10 E6	0.70 E6
2	1.10 E6	0.80 E6
3	1.10 E6	0.95 E6
4	1.10 E6	1.15 E6
5	1.30 E6	0.70 E6
6	1.30 E6	0.80 E6
7	1.30 E6	0.95 E6
8	1.30 E6	1.15 E6
9	2.20 E6	1.40 E6

Analysis Results

The geometry and mass properties of the rig generate two critical speeds in the rampressor rotor bearing system between 0 and 60,000 rpm. This is true for all cases except for Case No. 9 in which the second mode appears beyond 60,000 rpm. The modes show up on the Bode Plots for each case and the shapes are illustrated in Figures 2 and 3. The good news is that these modes are stable; i.e., the Log Decrement values are positive. In fact, no negative modes were found in any of the rig components for any of the nine cases.

The results of the nine analyses conducted are summarized in Table 1. Whirl Speed and Stability Maps together with Bode Plots and graphs of bearing forces vs. speed are presented for all 9 cases in Figures 4 through 57. Note that due to the large number of modes in the full rig model and limitations in the software, the whirl speeds of interest do not show up in the Whirl Speed Maps.

Figure 2 - Rampressor Mode 1

Figure 3 Rampressor Mode 2

Case		<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>	<u>5</u>	<u>6</u>	<u>7</u>	<u>8</u>	<u>9</u>
Ball Brg. Stiffness	lbf/in.	1.10E+06	1.10E+06	1.10E+06	1.10E+06	1.30E+06	1.30E+06	1.30E+06	1.30E+06	2.20E+06
Roller Brg. Stiffness	lbf/in.	7.00E+05	8.00E+05	9.50E+05	1.15E+06	7.00E+05	8.00E+05	9.50E+05	1.15E+06	1.40E+06
Rampressor Imbalance	gm-in.	0.22	0.22	0.22	0.22	0.22	0.22	0.22	0.22	0.22
Coupling Imbalance	gm-in.	0.166	0.166	0.166	0.166	0.166	0.166	0.166	0.166	0.166
Unstable Modes		No	No	No	No	No	No	No	No	No
Mode 1	(rpm)	29000	30000	31000	31500	29000	30000	31000	31500	32500
Rampressor CL	(in)	0.0012	0.0007	0.0005	0.0004	0.0013	0.0007	0.00055	0.0004	0.00045
Coupling Hub	(in)	0.011	0.0077	0.007	0.0069	0.0113	0.0078	0.0072	0.0066	0.0091
Ball Brg. Force	(lbf)	80	109	159	195	78	107	158	196	310
Roller Brg. Force	(lbf)	1872	1310	1224	1260	1878	1331	1270	1200	1726
Mode 2	(rpm)	49500	50000	51000	52500	51000	52000	53000	54500	> 60,000
Rampressor CL	(in)	0.0026	0.0038	0.0027	0.0047	0.0039	0.0047	0.0042	0.0054	
Coupling Hub	(in)	0.0023	0.0033	0.002	0.0027	0.0038	0.0039	0.0031	0.0032	
Ball Brg. Force	(lbf)	2460	3676	2599	4695	4027	4849	4444	5855	
Roller Brg. Force	(lbf)	1445	2322	1839	3793	2238	3013	3061	4556	

TABLE 1 – ROTORDYNAMIC ANALYSIS SUMMARY OF RESULTS

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case1.rot

Figure 4 – Case 1 Whirl Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case1.rot

Figure 5 – Case 1 Stability Map

Figure 6 – Case 1 Bode Plot – Rampressor Centerline

Figure 7 – Case 1 Bode Plot – Rampressor Coupling Hub

Figure 8 - Case 1 Ball Forces

Figure 9 – Case 1 Roller Bearing Forces

Figure 10 – Case 2 Whirl Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case2.rot

Figure 11 – Case 2 Stability Map

Figure 12 – Case 2 Bode Plot – Rampressor Centerline

Figure 13 – Case Bode Plot – Rampressor Coupling Hub

Figure 14 – Case 2 Ball Bearing Forces

Figure 15 – Case 2 Roller Bearing Forces

Figure 16 – Case 3 Whirl Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case3.rot

Figure 17 – Case 3 Stability Map

File:

Figure 18 – Case 3 Bode Plot – Rampressor Centerline

File:

Figure 19 – Case 3 Bode Plot – Rampressor Coupling Hub

Figure 20 – Case 3 Ball Bearing Forces

Figure 21 – Case 3 Roller Bearing Forces

Figure 22 – Case 4 Whirl Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case4.rot

Case 23 – Case 4 Stability Map

Figure 24 – Case 4 Bode Plot – Rampressor Centerline

Figure 25 – Case 4 Bode Plot – Rampressor Coupling Hub

Figure 26 – Ball Bearing Forces

Figure 27 – Case 4 Roller Bearing Forces

Figure 28 – Case 5 Whirl Speed Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case5.rot

Figure 29 – Case 5 Stability Map

Figure 30 - Figure 18 – Case 5 Bode Plot – Rampressor Centerline

Figure 31 – Case 5 Bode Plot – Rampressor Coupling Hub

Figure 32 – Case 5 Ball Bearing Forces

Figure 33 – Case 5 Roller Bearing Forces

Figure 34 – Case 6 Whirl Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case6.rot

Figure 35 – Case 6 Stability Map

Figure 36 – Case 6 Bode Plot – Rampressor Centerline

Figure 37 – Case 6 Bode Plot – Rampressor Coupling Hub

Figure 38 – Case 6 Ball Bearing Forces

Figure 39 – Case 6 Roller Bearing Forces

Figure 40 – Case 7 Whirl Speed Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case7.rot

Figure 41 – Case 7 Stability Map

Figure 42 – Case 7 Bode Plot – Rampressor Centerline

Figure 43 – Case 7 Bode Plot – Rampressor Coupling Hub

Figure 44 – Case 7 Ball Bearing Forces

Figure 45 – Case 7 Roller Bearing Forces

Figure 46 – Case 8 Whirl Speed Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case8.rot

Figure 47 – Case 8 Stability Map

Figure 48 – Case 8 Bode Plot – Rampressor Centerline

Figure 49 – Case 8 Bode Plot – Rampressor Coupling Hub

Figure 50 – Case 8 Ball Bearing Forces

Figure 51 – Case 8 Roller Bearing Forces

Figure 52 – Case 9 Whirl Speed Map

File: C:\Dyrobcs\10603-Ramgen\Rig 9-17-05\Case9.rot

Figure 53 – Case 9 Stability Map

Figure 54 – Case 9 Bode Plot – Rampressor Centerline

Figure 55 – Case 9 Bode Plot – Rampressor Coupling Hub

Figure 56 – Case 9 Ball Bearing Forces

Figure 57 – Case 9 Roller Bearing Forces

CHAPTER 4: COMPLETE THE CONCEPTUAL DESIGN OF THE RAMPRESSOR-TURBINE PRE-PROTOTYPE UNIT (TASK 14)

This report was previously submitted to the DOE in 2004 and is included in its entirety as an Appendix. Only the page numbers have been changed in the report to reflect its location within this BP4 Final Report document.

CHAPTER 4 APPENDIX

Rampressor Turbine Design Topical Report

March 1, 2003 to September 30, 2003

Principle Author: Ramgen Power Systems

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Submitted by:

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Abstract

The design of a unique gas turbine engine is presented. The first Rampressor Turbine engine rig will be a configuration where the Rampressor rotor is integrated into an existing industrial gas turbine engine. The Rampressor rotor compresses air which is burned in a traditional stationary combustion system in order to increase the enthalpy of the compressed air. The combustion products are then expanded through a conventional gas turbine which provides both compressor and electrical power. This in turn produces shaft torque, which drives a generator to provide electricity. The design and the associated design process of such an engine are discussed in this report.

This effort was funded in part by the Department of Energy under the cooperative agreement #DE-FC26-00N7 40915, awarded between the period of March 1, 2003 and September 30, 2003.

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1.0 Executive Summary

This report summarizes the results of Rampressor Turbine engine development design efforts. This effort was funded by Department of Energy cooperative agreement number DE-FC26-00N7 40915.

The Rampressor Turbine Rig represents a configuration where the Rampressor rotor is integrated into an existing or a new industrial gas turbine engine. The Rampressor rotor compresses the air which is burned in a traditional stationary combustion system in order to increase the enthalpy of the compressed air. The combustion products are then expanded through a conventional gas turbine which provides both compressor and electrical power. This in turn produces shaft torque, which drives a generator to provide electricity.

The high efficiency, low cost potential of the Rampressor Turbine engine product is key to the DOE vision of Distributed Energy for the future. Ramgen plans to advance the development of a 500KW class Rampressor Turbine to readiness for Field Demonstration. Based on improving knowledge of the physics of the engine, and limitations on funding it is believed that the proposed plan strikes the best balance between risk, cost and schedule.

This report describes how Ramgen evolved the supersonic shock inlet technology from the Ramgen Engine configuration to the Rampressor Turbine configuration. The goal of the Rampressor Turbine program is to eventually demonstrate a cost effective high efficiency industrial gas turbine utilizing a Rampressor compressor. The potential to generate higher efficiency is because of the inherently better performing Rampressor rotor. The successful development and commercialization of the Rampressor will lay the foundation for Ramgen to pursue development of the Rampressor Turbine product. To minimize the risks of the development of an engine, it was decided by Ramgen to stage the development of the engine by first pursuing the Rampressor Turbine design in stead of the Ramgen Engine.

An important consideration was how to proceed with the first Rampressor Turbine Rig. The rig could be designed and built

- 1) from scratch utilizing conventional combustor and expander (turbine) technology,
- 2) to incorporate advanced vortex combustor (AVC) technology with a conventional turbine,
- 3) or to integrate the Rampressor to an existing industrial gas turbine, thereby leveraging some of the known factors of the existing gas turbine engine

This report describes the analysis work completed on the AVC configuration and the insertion of a Rampressor on an existing gas turbine engine.

After due consideration, it was decided that in order to support a successful demonstration of the first Rampressor Turbine engine rig, it was prudent to integrate a

Rampressor to an existing gas turbine engine. Creating a complete Rampressor Turbine engine design from scratch would result in enormous costs to develop a one-off combustor and turbine. To design AVC into the engine from the start would be unnecessarily risky since the development of AVC is a complete program in itself. A delay or set-back in the development of either the Rampressor Turbine or AVC would delay both in a combined development scenario.

Other decisions that followed involve choosing a single spool gas turbine engine and a moderately sized engine at or below 1MW that had an overall pressure ratio at or below 10:1. These decisions were made methodically so as to give Ramgen the highest chance of success at demonstrating the Rampressor Turbine 1 technology. Once such a technology demonstration is made, an engine where performance becomes a key factor will be incrementally targeted in future Rampressor Turbine programs.

The design of the first Rampressor Turbine Rig, which combines a Rampressor with an off-the shelf combustor and turbine engine represents the integration of the Rampressor compression technology to an established industrial gas turbine engine design. Currently the Rampressor Turbine Rig configuration is designed to demonstrate the viability of this particular engine and does not particularly emphasize the performance metrics of the engine. At maturity, the production engine will have a thermal efficiency rivaling and surpassing conventional gas turbines, with highly competitive CO and NO_x emissions through its operation in a lean premixed mode.

Accomplishment of the completed design tasks has led to a greater understanding of what is required to adapt the Rampressor technology to an existing industrial gas turbine engine. Based on the efforts to date, it is clear that a great deal of work is yet to be done in order to match the various components of the Rampressor Turbine engine from an aero and mechanical perspective, for the successful operation of this engine. Significant design and analysis milestones towards answering the many challenges will be made in the following contract periods. Future efforts will build on the foundation of this research and development effort.

2.0 Introduction

This document presents the early stage design of a new kind of power generation device known as the Rampressor Turbine Engine. These efforts have been aided by the Department of Energy, as part of an initiative to bring viable and innovative energy technologies to market.

This document is split into two major sections. The first section details the planned evolution of this technology over a period of time to include an Advanced Vortex Combustor and how Ramgen's technology development in other programs feed into the Rampressor Turbine technology advancement. The second section of this document details the process by which one particular industrial gas turbine engine was selected as the best possible fit for demonstrating this configuration from many different existing

engines. A variety of factors including cost, design, resources and schedule were considered before the selection was made.

The Rampressor Turbine configuration was born from the challenge of creating a viable energy producing product based on superior Rampressor compression technology. The first power producing engine configuration was the Ramgen Engine. See Figure 2-1.

Figure 2-1. Ramgen Engine

Description of Ramgen's Technology Evolution

The design of the Ramgen engine represents a unique application of well-established ramjet technology to electric power generation. In both the Ramgen Engine and the Rampressor Turbine the ramjet thrust modules are mounted on the perimeter of an enclosed rotor, which revolves at supersonic rim speeds. Similar to the increased Mach number for ramjets used in aerospace applications, the faster the rotor spins, the more efficiently the compression process, which is a significant factor affecting engine efficiency. Ramgen projects that the technology can be scaled to produce electrical output ranging from 800 kilowatts ("kW") to at least 10 MW and arranged in configurations for larger outputs. Ramgen's development plan will target the 1MW class in engine size. The Ramgen Engine technology is at an early stage in its development evolution. Just as the gas turbine engine significantly increased its efficiency over its 60 years of development, it is realistic to expect that significant improvements will take place with the Ramgen engine and with the Rampressor Turbine engine during the ongoing process of technological improvement.

The first pre-prototype Ramgen Engine was assembled in 1998, at the Ramgen Company's test site in Tacoma, Washington. On February 2, 1999, the Ramgen Engine was ignited for the first time. Ramgen engineers continued to improve the pre-prototype Ramgen engine's combustor, fuel injection, ignition, and air film cooling systems in the Tacoma pre-prototype Ramgen engine. In the third and fourth quarter of 2000 the prototype engine was rotated to Mach 1.1.

Several improvements have already been conceptualized by Ramgen to increase efficiency, lower costs, and further reduce emissions. The extensive design work and

component testing that was required to achieve these advances in the next demonstration rig were executed during Budget Period 2 and 3. In addition, it is projected that the single most important factor to achieve higher efficiency in future generations of Ramgen engines will be advances in materials, which will permit higher rotational speeds. Advances in high strength/light weight materials have been occurring steadily and there is every reason to believe that such advances will continue.

In April, 2002, the DOE conducted a two day Design Review of the Company's technology at the NASA Glenn facility outside Cleveland, Ohio. This Design Review involved approximately 20 prominent scientists from the DOE, Department of Defense, and NASA, as well as a team of Ramgen scientists. At the time of this Design Review in Cleveland, Ramgen estimated that the development costs to demonstrate an advanced Ramgen Engine were in excess of \$30 million.

One of the most significant conclusions Ramgen brought back from the Review was that a deterministic component level validation approach was the best path to successfully develop the technology. Additionally, the National Energy Technology Lab (NETL) stated that the testing of the Tacoma engine should have been "more aggressive" with further subcomponent testing planned in the future. The new phase of subcomponent tests of the next generation Ramgen engine (i.e., inlet compression, combustor, rotor dynamics, nozzle expansion, control logic, etc.) will provide increased understanding of the mechanical challenges and aerodynamic flow-field characteristics.

As a result of that Design Review, Ramgen management concluded that most of the advantages and value of the technology could be realized at a fraction of this cost (and within probable DOE support levels), in a shorter time period and with less technical risk. Consequently, Ramgen has focused its resources on demonstrating the Rampressor and designing the Rampressor Turbine in the near-term. The technology required for the Rampressor and the Rampressor Turbine are also important components of the Ramgen Engine under development on a longer timeline.

Rampressor Turbine

In placing a Rampressor on the front end of a stationary combustor and turbine stage, Ramgen has developed what it calls the Rampressor Turbine. The conceptual rendering of Figure 2.2 represents a potential Rampressor Turbine product for commercial sale. As shown in the rendering a Rampressor compressor is configured to flow into a conventional silo style burner and then a conventional axial turbine.

Figure 2.2 Rampressor Turbine

The Rampressor Turbine simplifies the engine technology development by removing the hot combustor from the spinning Ramgen Engine rotor. By separating the combustor from the rotor the technical issues are significantly reduced. The Rampressor Turbine engine uses the ramjet technology to capture the projected advantages in efficiency, cost, and fuel flexibility of the original Ramgen engine, while significantly reducing the time, risk, and money required to complete the design and manufacture of the first commercial engine.

The design for the Rampressor turbine system as shown in Figure 2.2 was to use the high efficiency of the Rampressor compressor in a small simple cycle engine system that would have potential application into the distributed generation market.

The result was a conceptual system layout for a ~500 kW system. Two derivatives were considered at pressure ratios of 10 and 15:1. To achieve these pressure ratios, a centrifugal pre-compressor was coupled with the Rampressor wheel. The discharge from the Rampressor stage fed a reverse flow “can type” combustor that could, eventually utilize a trapped vortex flame holding scheme. The flow was then collected and routed into a centrifugal expansion stage to achieve a decrease in flow-path mean line radius prior to being completely expanded through two additional axial flow turbine stages.

The turbine and combustor stages of both designs were based on the Kawasaki S2A-01 which is a 700 kW system with a pressure ratio of ~ 9:1. The Kawasaki components were scaled to correct for mass flow and pressure ratio. Having estimated the size and speed of the turbine stages, the Rampressor wheels were sized to develop the appropriate inflow Mach numbers for the two different pressure ratio options.

The two systems considered are summarized below in Table 2.1:

Design Rating	10:1 Design	15:1 Design
Shaft Power	485 kW	485 kW
System Efficiency	30.8%	32.7%
Pressure Ratio	10:1	15:1
Air flow	4.1 pps	4.3 pps
Shaft Speed	44,800 rpm	54,870 rpm
TRIT	1700 F	1700 F
Exhaust Temp	885 F	885 F
Table 2.1 Rampressor Turbine Layout Options		

This conceptual design iteration highlighted the importance of the designing the turbine stages to provide good mass flow and speed matching with the compressor. In order to ensure proper turbine matching, subsequent Rampressor turbine conceptual design iterations involved the evaluation of multiple existing system candidates. These evaluations were based on many factors and ultimately led to the initial hybridized Rampressor turbine system that was proposed for further design.

3.0 The Rampressor Turbine Engine Design with Advanced Vortex Combustor and Matched Conventional Turbine

3.1 Integration of Rampressor Technology to a Gas Turbine

The approach to developing a Rampressor Turbine utilizing Advanced Vortex Combustion (AVC) technology would consist of several steps in the process of eventually combining the Rampressor compressor and AVC burner technologies into a Rampressor AVC Turbine. As shown in Fig. 3.1, the separate development of commercial applications of the Rampressor, Rampressor Turbine, and AVC technologies naturally leads to their eventual combination in a Rampressor AVC Turbine. The primary challenges to achieving this outcome are technology development coordination, timing, and funding levels. In addition, determining the specific application or set of related applications of the Rampressor AVC Turbine will be critical in assembling the appropriate technology development path.

The technology development path to reach a Rampressor AVC Turbine would consist of parallel development paths of a Rampressor Turbine and AVC technology that would eventually be combined after sufficient demonstration of each technology. This is shown schematically in Fig. 3.2.

The first stage in progressing towards a Rampressor AVC Turbine would be to combine for the first time a Rampressor, conventional combustor, and turbine for power production. The Rampressor compressor would leverage from the technology base produced by demonstrating a 10:1 Rampressor rotor test rig (Rampressor-II). The initial Rampressor Turbine application would require a lower pressure ratio than 10:1 and would likely be near 6:1, thus reducing the development risk.

The 1st prototype Rampressor Turbine would demonstrate the combination and operating capability of a Rampressor adapted to a conventional combustor (most likely a silo combustor) and power turbine. This prototype system would not likely be the correct size or configuration for an eventual Rampressor AVC Turbine application, but would serve to demonstrate the integrated operation of the components and provide many lessons learned.

The configuration for the 1st Rampressor Turbine scope of supply includes: Compressor, compressor motor, inverter, burner, turbine, turbine generator, inverter, rig skid, controls, rig oil supply, rig water, fuel, cooling air, vacuum, etc. The scope of supply of the entire project includes fuel cell, fuel cell support systems, heat exchanger, plumbing and valves between the Rampressor Turbine rig, fuel cell, and heat exchanger. See Figure 3.3; Notional Fuel Cell / Rampressor Turbine Rig Schematic.

Figure 3.3. Notional Fuel Cell / Rampressor Turbine Rig Schematic

The Rampressor Turbine concept currently envisioned is a 4:1 to 6:1 pressure ratio Rampressor compressor with an available industrial combustor and turbine. This assumes an Indirect System (un-pressurized) will be optimal. The test rig compressor and turbine will be electrically coupled to eliminate matching issues between the

components. This configuration will provide the lowest risk, the lowest cost and the quickest path to demonstrating the potential of an optimized Fuel Cell/Rampressor Turbine Hybrid system.

The second Rampressor Turbine prototype would be dedicated to a specific application and would look functionally like the early Rampressor Turbine design shown in Fig. 3.4 that has a Rampressor compressor, conventional silo combustor, and an expander turbine. For comparison, a Kawasaki small gas turbine which utilizes a similar design is also shown. This device would demonstrate full operation of a Rampressor compressor with a conventional combustor and expansion turbine and would not be restricted to limitations imposed by a hybrid fuel cell system.

In parallel, the AVC technology development path would consist of taking the ongoing NETL/RPS AVC program results and designing an AVC combustor to retrofit into an existing small gas turbine that utilizes a silo combustor. This approach would reduce risk by only modifying the combustor and not the compressor. This gas turbine test rig would then fully characterize the application of an AVC combustor in a proven gas turbine configuration. The results from the Rampressor Turbine and AVC silo combustor demonstrations would then allow integration of the Rampressor and AVC technologies into a single prototype Rampressor AVC Turbine unit.

Figure 3.4: Early conceptual rendering of a Rampressor Turbine and Kawasaki small gas turbine

4.0 The Rampressor Turbine Engine

4.1 Integration of Rampressor Technology to a Gas Turbine

The compressor part of the Rampressor Turbine Engine is based on aero propulsion supersonic flight inlet technology which can accomplish high compression with high efficiency in a single stage. This makes the Rampressor the simplest mechanical air-compression device to date through its minimal use of moving parts.

The compressed air then goes through a stationary combustor (single can, can annular etc.,) where it mixes with the fuel in the combustion chamber. The fuel-air mixture is burned in this chamber following which it is expanded in the turbine, which results in power generation. The generated power is converted into electric power using a generator, which is attached to the turbine shaft via a gear box.

The uniqueness of the Rampressor technology is that the compression is across a single stage. This leads to fewer moving mechanical parts, which reduces the cost of the product. In addition to these potential benefits, the Rampressor has a high compression ratio across an individual stage (i.e., 10:1 or greater). The overall efficiency of the Rampressor is expected to be equal or better than comparable centrifugal or axial compressors. The above characteristics lead to the big advantages of the Rampressor over centrifugal and axial compressors. The Rampressor is better than centrifugal compressors in that it maintains a relatively high efficiency even at high pressure ratios and it much more compact than an axial compressor. All of these characteristics lead to a much improved performance metric for a Rampressor integrated industrial gas turbine.

4.2 Ramgen's Technology Development Plan

This section deals with the integration of the Rampressor Turbine Program into Ramgen's technology development plan. As detailed in figure 4.1, the Rampressor Turbine program trails the Rampressor technology development in its various complementary phases. This allows sufficient lessons learned in the Rampressor technology to be incorporated into the Rampressor Turbine program. As shown in figure 4.1, the Rampressor 1 & Rampressor 2 programs are to be tested and designed, respectively, by end of FY 2004. FYs 2005 & 2006 are dedicated to development of Rampressor 3 and Rampressor 4 programs. Simultaneously, the Rampressor Turbine 1 program is scheduled for test in FY 2005, thereby allowing the incorporation of all the lessons learned in the Rampressor 1 & Rampressor 2 programs into this, for a better chance at success.

Future Rampressor Turbine Programs are planned based on additional information from the Rampressor Technology development. Incrementally, these future programs will demonstrate increased performance, both from an emissions standpoint and from an

aerodynamics perspective. The Rampressor Turbine 1 program, as indicated before, is principally concerned with demonstrating the concept of a Rampressor working in a power generation environment with no special emphasis on performance metrics.

The Advanced Vortex Combustion (AVC) program is being developed in fiscal years 2003 through 2004 and beyond, so that a combination of this technology along with the advancement of the Rampressor Turbine technology could be incorporated in the Ramgen Turbine program possibly as soon as fiscal year 2008 contingent upon funding and commercial interest.

4.3 Rampressor Turbine Applications

The Rampressor Turbine product is targeted for a variety of applications such as distributed generation (DG). The Rampressor Turbine product geared towards a distributed generation market can either be a high pressure ratio engine or a low pressure ratio engine. It can be operated in a simple cycle environment as well as in a recuperated cycle environment. Usually, the concentration of power outputs for this kind of system is below 1 MW.

In all applications, there is great pressure to reduce system costs, while maintaining high efficiencies. In order to satisfy these market demands and to make efficient use of the funds available, Ramgen decided to put more emphasis on engines that are smaller in size in terms of output power and lower in pressure ratio.

4.4 Methodology of Selecting Engines

The process of selecting the correct engine for the Rampressor Turbine 1 program was carefully considered and methodical. It was a balance between the resources available, cost and complexity of the program and the timeline available for demonstrating such an engine program. The flowchart of the overall process is shown in figure 4.2 and is briefly discussed in this section. The individual steps are later covered in more detail.

As shown in figure 4.2, the overall selection process included many steps. The first step was a study of the generic configurations and options available for the Rampressor turbine product. A decision about the configuration resulted in narrowing down the list of turbine engines that were suitable for the first test rig of Rampressor Turbine engine. The different gas turbines that were identified as being possible candidates were used for generating a list of the generic aerodynamic, mechanical and peripheral risk/issues. This general list was further refined, as shown in the third step, into a list of the important metrics by which different turbines were to be judged. The metrics were then weighted according to the importance that each of them carried to Ramgen's overall goals for the Rampressor Turbine 1 engine. This process helped in reducing the total number of identified engines to a more manageable list.

A further investigation of the list of engines that were left standing after the weighting process helped in reduction of the number of engines to 2. Finally, taking into

consideration Ramgen's resources and lessons learnt from the Rampressor 1 test rig, the list of 2 engines was further reduced to one engine. Once the engine was selected, the associated cross section layout was developed and an early design review of this layout was conducted. The next few sections detail each of the steps listed in the flowchart, shown in figure 4.2.

4.4.1 General Configuration Decisions

As mentioned earlier, the selection process for a suitable Rampressor Turbine 1 demonstration engine was started by trying to assimilate the different engine configurations within the constraints of the resources, cost, technical risk and time available. The gas turbine engines can be broadly classified as single spool or multi-spool engines. Also, since the Rampressor turbine product is ultimately targeted towards the distribution generation market an engine in the 400 KW class is a priority. The pressure ratio of the engine was a factor that was considered as well, because the higher the pressure ratio of the overall system, the greater the technical challenges for the Rampressor Turbine 1 engine.

Another important consideration was whether to build a new engine from scratch or integrate the Rampressor to an existing industrial gas turbine, thereby leveraging some of the known factors of the existing gas turbine engine. After due consideration for all of the above questions, it was decided that in order to support a successful demonstration of the Rampressor Turbine 1 engine, it was prudent to integrate a Rampressor to an existing gas turbine engine. Other decisions that followed involve choosing a single spool gas turbine engine and a moderately sized engine at or below 1MW that had an overall pressure ratio at or below 10:1. These decisions were made methodically so as to give Ramgen the highest chance of success at demonstrating the Rampressor Turbine 1 technology. Once such a technology demonstration is made, an engine where performance becomes a key factor will be incrementally targeted in future Rampressor Turbine programs.

4.4.2 Generic Aero/Mechanical/Peripheral Issues

This section deals with identifying the principal aero and mechanical issues as well as the major components of any Rampressor Turbine Engine. The idea behind identifying such issues is to ensure that during the gas turbine engine selection/ design process such issues are given due consideration. This helps in identifying the appropriate turbine for the Rampressor Turbine 1 program after considering the various risks involved with each of the issues.

4.4.3 Aerodynamic Issues

The major components that are required for this engine are the impeller in the front (to assist in self-aspiration), followed by a De-swirl vane (to remove the swirl from flow out of the impeller and align it axial for the Pre-swirl inlet) and a Pre-Swirl nozzle for the Rampressor rotor. All of the above mentioned components are upstream of the Rampressor Rotor. The components downstream of the Rampressor rotor are a De-Swirl

vane (required to take out the swirl coming out of the rotor), a Diffuser (to diffuse the flow from the rotor to a low mach number flow so that it could be fed to the downstream combustion components), a bleed port (to assist in the starting of the engine), a combustor (to allow for combustion) and a turbine (to expand the high pressure gas so that power is generated). Since Ramgen decided to integrate the Rampressor rotor to an existing gas turbine engine rather than build a completely new engine, the design effort concerning the combustion system and the turbine system design is minimal.

The other two other issues that need to be dealt with. First, the starter motor size for the Rampressor Turbine engine. Second, the requirement of turbine maps of the chosen industrial gas turbine engine to which the Rampressor is to be integrated into (helps determine the starting process of the Rampressor Turbine engine).

4.4.4 Mechanical Issues

All the issues discussed so far pertain to the aero aspects of the engine. On the mechanical and structural side of things, the requirements/issue are a low loss transition piece from the impeller De-Swirl vane to the Pre-Swirl nozzle, struts in the flow path to allow for oil feed lines, buffer air feed lines and bleed air lines. This has a potentially detrimental effect on the performance of the engine because of the blockage created and hence needs to be carefully designed. The diffuser and de-Swirl components, if variable geometry, could be challenging tasks. Since the Rampressor rotor is to be integrated with an existing gas turbine engine, access to the cross section of the engine is an important factor in the design of the Rampressor Turbine 1 engine. The other potential mechanical issues are the rotor dynamics of the overall system, the integration of the Rampressor shaft to the existing turbine shaft and tip speed limitations of the rotor design.

4.4.5 Peripheral Issues

Among the various peripheral issues for this engine is a requirement for vacuum system for the bleed air. Since the Rampressor rotor requires certain percentage of air to be bled off from the mainstream flow for performance and starting reasons, a higher mass flow rate engine will require a bigger vacuum system, which could prove to be a hindrance to the performance of the overall design. In addition, the Rampressor is in its early developmental stage and efforts have not been made to optimize the bleed requirement. Future programs of this design are expected to concentrate on such issues, which will be incorporated in the future Rampressor Turbine products, as was discussed in the overall technology development section of this document. The other peripheral system requirements are a tip clearance control system, a thrust balance system, a generator or a load bank for using the power generated, a gearbox between the generator and the turbine/compressor. The connection between the starter motor and repressor shaft is yet another peripheral issue that needs due consideration during the design phase of the project.

The issues discussed in this section help in defining the important metrics that are to be considered for choosing an appropriate gas turbine engine for the Rampressor Turbine 1 program. These metrics are discussed in the following section

4.4.6 Important Metrics & Their Weighting Factors

This section deals with specifically identifying the important metrics by which different engines are to be evaluated. Following the identification of these metrics, they are weighted according to how critical they are for the overall engine design. The various metrics that are considered for evaluating these different engines are:

- (i) Output Power of the system
- (ii) Design Pressure Ratio
- (iii) Mass Flow rate
- (iv) Whether the turbine is cooled or uncooled
- (v) Availability of the engine in the market
- (vi) Starter Motor Size/Requirement
- (vii) Compressor Tip Clearance Requirement
- (viii) Availability of Access to Engine Cross Section
- (ix) Access to Turbine Maps
- (x) Access to Compressor Maps
- (xi) Availability of Consultants
- (xii) Cost of the chosen engine
- (xiii) Industrial engine or an APU
- (xiv) Type of fuel used, natural gas, liquid or dual fuel
- (xv) Potential supplier of the same
- (xvi) Technical Challenge of tip speed
- (xvii) Rampressor centerline radius
- (xviii) Flow path from the Rampressor to the combustor inlet

The metrics are weighted on a scale of 10 to 100 in steps of 5, 10 being the least important and 100 being the most important factor, relative to each other. The Ramgen team discussed these various parameters and assigned the weighting factors, as was seen appropriate to the overall design of the Rampressor Turbine engine. These are captured in Table 4.0. From the table, it is clear that among all the metrics considered, the team picked the availability of the engine, the complexity of the engine (whether cooled or uncooled) and the availability of consultants that understand the various components of the engine as the most critical parameters for this design. Access to compressor maps of existing engines, starter motor size and the number of suppliers available for each of the different engines was considered least critical for the design. These decisions and the corresponding weighting factors are made/assigned after considering that this rig was the first of its kind.

Table 4.0 Important Engine Metrics Versus Weighting Factors

Important Engine Metrics	Weighting Factors

Output Power of the system	40
Design Pressure Ratio	80
Mass flow rate	60
Whether the turbine is Cooled or Uncooled	100
Availability of the engine in the market	100
Starter Motor Size/Requirement	30
Compressor Tip Clearance Requirement	75
Availability of Access to Engine Cross Section	80
Access to Turbine Maps	70
Access to Compressor Maps	20
Availability of Consultants	100
Cost of the chose engine	50
Industrial Engine or an APUS	40
Type of fuel used, natural gas, liquid or dual fuel	60
Potential supplier of the same	30
Technical challenge of tip speed	80
Rampressor Centerline radius	70
Flow path from the Rampressor to the combustor inlet	70

4.4.7 Ranking Process of Different Engines

In the previous section, the important parameters and the corresponding weighting factors were discussed. This section discusses how individual scores are assigned to each of those weighted parameters. In general, the scores are assigned on a scale from 1 to 5. A higher score is usually considered good and a lower score is considered bad. For the different parameters, the respective scores breakdown is discussed below

4.4.7.1 Output power of the system

The output power of the system is one of the many critical parameters that Ramgen identified. This is because it affects the size of rotor that Ramgen needs to design. Since this directly relates to the Ramgen's design experience, a careful analysis of the same was conducted and scores associated with the generic engine's power rating is detailed below.

Table 4.1 Assigned scores for the Output power of the system

Output power of the system (Horse Power)	Assigned scores
< 300, > 1000	1
700 to 1000	2
600 to 700	3
300 to 350, 500 to 600	4
350 to 500	5

4.4.7.2 Design Pressure Ratio

The overall pressure ratio of the system affects the aero and mechanical system challenges. A low pressure ratio machine has fewer design challenges compared to a high pressure ratio machine. Taking this into consideration, the generic engine's pressure ratio and its associated scores is detailed below

Table 4.2 Assigned scores for Design pressure ratio

Design pressure ratio	Assigned scores
> 8	1
7 to 8	2
6 to 7	3
5 to 6	4
< 5	5

4.4.7.3 Mass flow rate

The mass flow rate of the system affects the rotor design of the Rampressor. A higher mass flow rate requires a bigger flow path and a lower flow rate requires a smaller flow path. Since this directly corresponds to the actual design of the rotor, due consideration for this parameter was given and the generic engine's mass flow rate and its associated scores is detailed below

Table 4.3 Assigned scores for Design mass flow rate

Mass flow rate (lbm/s)	Assigned scores
> 10	2
6 to 10	3
< 6	4

4.4.7.4 Turbine cooling configuration

A turbine that is cooled is more complex than a turbine that is uncooled. Not only does the design have to consider the routing of the secondary flows required for cooling the turbine, but also the tap off points from the Rampressor rotor need to be roped into the design as well. Since this entails a more sophisticated design analysis, a turbine with no cooling would be a highly preferred way of designing the Rampressor 1 turbine. The generic engine's cooling configuration and its associated scores is detailed below

Table 4.4 Assigned scores for Turbine cooling configuration

Turbine cooling configuration	Assigned scores
Cooled	1
Potentially Cooled	3
Uncooled	5

4.4.7.5 Availability of engines in the second hand market

The simplicity of the design does not matter if such an engine is unavailable in the market relatively easily. So, an engine that is readily available in the market, even if its slightly more complicated in design, is preferable to one that is not available in the second hand market. The generic engine's availability and its associated scores is detailed below

Table 4.5 Assigned scores for Availability of engines in the second hand market

Availability of engines	Assigned scores
Unavailable	1
Very Low Chance of Availability	2
Engines are available. Time of availability is in question	3
Manufacturers/ Overhaul places have dealt with it in the past and if asked for they can locate such engines	4
Readily available	5

4.4.7.6 Starter motor size

For starter motor size determination, the Rampressor Turbine has been assumed to require a starter motor that will be large enough to take it up to 50% (which is pretty typical for gas turbines) of its design speed. A smaller starter motor is preferable because of its compactness and simplicity in design. Based on this assumption, the generic engine's starter motor size and its associated scores is detailed below

Table 4.6 Assigned scores for Starter motor size

Starter motor size (Horsepower)	Assigned scores
> 1000	1
750 to 1000	2
500 to 750	3
250 to 500	4
0 to 250	5

4.4.7.7 Availability of the turbine cross section

One of the essential elements required for integrating a new front end to an existing design, is the availability of an existing engine cross-section. The easier it is to get an engine cross-section, the higher the assigned scores. The availability of the generic engine's cross section and its associated scores is detailed below

Table 4.7 Assigned scores for Availability of turbine cross section

Availability of turbine cross section	Assigned scores
Reasonably good chance of getting it	4
Currently available	5

4.4.7.8 Availability of the consultants

Another critical element for designing a front end for an existing engine is the availability of designers with expertise and experience on the selected engines. The availability of such consultants results in a high score whereas the unavailability of such consultants results in a low score, in our weighting model. The associated scores of the same is detailed below

Table 4.8 Assigned Scores for Availability of consultants

Availability of consultants	Assigned Scores
None available	1
Low probability of identifying	2
If looked around, consultants maybe available	3
Pretty Confident of consultants being available	4
Consultants Currently available	5

4.4.7.9 Access to Turbine Maps

One of the considerations while designing a front end for an existing engine is the operating characteristic of the turbine. A turbine map provides information regarding the operation of the engine. Getting this information is invaluable and the associated scores of accessing the turbine maps is detailed below

Table 4.9 Assigned Scores for Access to turbine maps

Access to turbine maps	Assigned Scores
Unavailable	1
Decent chance of availability	3

Currently Available	5

4.4.7.10 Access to Compressor Maps

The other principal characteristic that is required for designing the front end is the operating characteristic of the existing compressor of the engine. This is important because designing the Rampressor to have the same characteristic as that of the existing compressor will make running the overall turbine engine a much simpler job. Getting access to the compressor maps of the generic engine, by way of reverse engineering, and its associated scores is detailed below

Table 4.10 Assigned Scores for Access to compressor maps

Access to compressor Maps	Assigned Scores
Unavailable	1
Decent chance of availability	3
Currently Available	5

4.4.7.11 Used / Overhauled gas turbine cost

This relates to the funds available at Ramgen for purchasing a generic gas turbine engine. The cost of the individual engine package and its associated scores are detailed below

Table 4.11 Assigned Scores for Used / Overhauled gas turbine cost

Used / Overhauled gas turbine cost (\$)	Assigned scores
> 200k	1
150k to 200k	2
100k to 150k	3
50k to 100k	4
< 50k	5

4.4.7.12 Type of engine

This relates to the whether the generic gas turbine engine is an industrial gas turbine or an APU or aero propulsion based. An industrial gas turbine is more rugged than a comparable aero engine. So, a natural preference for a first time demonstration is the industrial engine. This is reflected in the following table which matches the type of engine and its associated score as

Table 4.12 Assigned Scores for the Type of engine.

Type of engine	Assigned scores
Aero Propulsion	1
APU	3
Industrial	5

4.4.7.13 Type of fuel used

This relates to the type of fuel used in the generic gas turbine engine. The fuel type that the engine uses confines the testing location to a certain number of test cells. In order to give Ramgen the most flexibility in identifying the test cells, an engine that supports dual fuel is most preferred and is reflected in the table below. The type of engine and its associated scores are detailed below

Table 4.13 Assigned scores for the Type of fuel used

Type of fuel used	Assigned scores
Gas	1
Liquid / Dual	5

4.4.7.14 Number of overhaul engine suppliers

This relates to the number of suppliers available to supply the generic engine either in a used state or in an overhauled state. Having more than a single supplier gives Ramgen the flexibility to negotiate on prices as well as the engine package. So the engines that have more suppliers in the secondary market are preferable. This is reflected in the table below and the associated scores for the same is detailed below

Table 4.14 Assigned scores for the Number of overhaul engine suppliers

Number of Suppliers	Assigned Scores
No Suppliers currently identified	1
1 supplier currently identified	3
2 supplier currently identified	4
More than 3 supplier identified	5

4.4.7.15 Rampressor rotor tip speed

This relates to the technical challenge of the design of the Rampressor rotor for the generic gas turbine engine. The higher the tip speed the more challenging the design gets. Since Ramgen is committed to developing a Rampressor turbine 1 demonstration unit the risk is to be kept at a minimum. This indicates that lower tip speed engines would be preferable to higher tip speed engines and is reflected in the table below. The associated scores for the tip speeds is detailed below

Table 4.15 Assigned scores for the effect of Rampressor rotor tip speed

Tip Speed (ft/s)	Assigned Scores
> 2050	1
1950 to 2050	2
1850 to 1949	3
1750 to 1849	4
< 1750	5

4.4.7.16 Rampressor centerline radius

This also relates to the technical challenge of the design of the Rampressor rotor for the generic gas turbine engine. This parameter was evaluated against the existing design experience at Ramgen and a design similar to an existing design was given a higher score, largely because of familiarity. Having a smaller sized rotor makes for a challenge to get the secondary systems around the main rotor. The associated scores for the same is detailed below

Table 4.16 Assigned scores for the Rampressor centerline radius

Rampressor centerline radius (inches)	Assigned scores
< 3	1
3 to 3.5	2
3.5 to 4.0	3
4 to 4.5	4
4.5 to 7	5

The abovementioned tables and the scores allotted to each of the different parameters were done based on what Ramgen considered to be important design considerations. An assignment of scores for each of the turbines, based on the guidelines established by the abovementioned process along with the weighting factors assigned for the different engines (as was discussed in the previous section) provides a good indication of the engine's overall fit to the Rampressor Turbine 1 Program needs. The sum of the product of the individual engine parameter score and the assigned weighting factor provides each engine with an overall weighted score. The overall score of each of the several engines considered is shown in figure 4.3. From figure 4.3., it is clear that there are a few engines

that are clearly better suited for this application than others. The top five candidates are the Solar Spartan, Solar Saturn, Garrett Lycoming T53, Garrett 200 & Sundstrand GTLC engines. Of these five engines, the T53 engine is the only engine with multiple spools. Since dealing with the complexity of a multi spool engine was considered beyond the Rampressor Turbine 1 program requirements, this particular engine was eliminated from consideration for integration with a Rampressor rotor.

4.5 Comparison of the Top Engine Candidates

Based on the weighted scores method, four turbine engines were chosen: Solar Spartan, Solar Saturn, Sundstrand APS2000 and Garrett 36-200. The critical parameters for these four engines are shown in figure 4.4. All of these engines operate at about a pressure ratio of four, except the Saturn engine. The mass flow rates are vastly different, which directly impacts the bleed system requirement. As expected, the size of the bleed system requirement for each of these different engines is quite different. For most of these engines, the starter motor that is currently being used at Ramgen for Rampressor 1 can be made use of except for the Saturn engine. Among the four engines, a consultant for the Garrett engine is the toughest to find and clearly there are more suppliers for the Saturn and the Spartan engines. The Saturn engine is much bigger than the remaining engines and the Garret engine is really small. The centerline Mach number of the Saturn engine is much higher than the other engines. The Rampressor rig that is currently in operation at Ramgen operates at a centerline mach number of 1.6, which is consistent with all engines other than the Saturn engine. A table showing the pros & cons of each of the four designs is discussed below.

Since the top four engine candidates have their own advantages and disadvantages, further in-depth analysis was required to choose one final engine configuration for the Rampressor turbine 1 program. In order to do this, the Rampressor rotor design, which is described in the following section, was laid out with the four turbine engines.

Table 4.17 Pros & Cons of the four engines

Spartan		Saturn		Sundstrand APS 2000		Garrett 36-200	
Pros	Cons	Pros	Cons	Pros	Cons	Pros	Cons
Mean line point design relative Mach Number comparable to Rampressor 1 design	Potential overall efficiency improvements by having a Rampressor in the front end are modest	Consultants available to help Ramgen design a compressor	Pressure ratio much higher than the Rampressor 1 design	Pressure ratio comparable to the Rampressor 1 design	Minimally available in the secondary market	Pressure ratio comparable to the Rampressor 1 design	Impeller design for the design point does not meet established design criteria.
Design point mass flow rate/ pressure ratio comparable to the Rampressor 1 design		Widely available in the secondary market	Design point mass flow rate much higher than the Rampressor 1 design	Mean line design point relative Mach Number comparable to Rampressor 1 design	Design point mass flow rate more than twice that of the Rampressor 1 design	Design point mass flow rate comparable to the Rampressor 1 design	Relatively less number of consultants with expertise on this engine available to help Ramgen
Consultants available to help Ramgen design a compressor		An acceptable front end impeller design, if required, is available.	Requirements on test cells much more onerous than the other 3 engines		Relatively less number of consultants with expertise on this engine available to help Ramgen	Requirements on test cells similar not very burdensome.	
Widely available in the secondary market			Mean line design point relative Mach Number much higher than Rampressor 1 design		Potential overall efficiency improvements by having a Rampressor in the front end are modest	Mean line design point relative Mach Number comparable to Rampressor 1 design	Potential overall efficiency improvements by having a Rampressor in the front end are modest

4.6 Rampressor Rotor Layout Process

For laying out the design of the Rampressor rotor, the information that was required was the engine rpm, the overall compressor pressure ratio, the engine mass flow rate. Once this information was provided, the designer uses the usual design tools to layout the Rampressor rotor design. For all of the designs a variety of inlet conditions were tried out and only the design pertaining to an inflow total pressure of nearly 1 atmosphere and an inlet temperature of 59 F and 80 F are reported. The design layout process is by itself an iterative process starting with the incorporation of the lessons learned from the Rampressor 1 design process. The design process consisted of three steps

1. The Mean line radius is adjusted to get the rotor M_{rel} (relative Mach Number) at the gas turbine operating RPM.
2. The capture area is adjusted for the mass flow rate
3. If the tip speed is too high, the pre-swirl nozzle Mach number was increased along with the pre-swirl nozzle stage pressure ratio.

The rotor capture area and radius is scaled from the Rampressor 1 design to achieve the desired mass flow and the contraction ratio is adjusted to achieve a 1.3 Mach for the normal shock. The desired pressure ratio and inlet absolute mach number is checked to see if they are matched for highest performance and the process is repeated until all the design requirements are met. The parameters for the four turbines described earlier are captured in Figure 4.4.

From the above information, it is clear that the Saturn turbine engine design carries a slightly higher risk than any of the other engines, strictly from a Rampressor rotor design point of view. This is because of the higher M_{rel} for Saturn turbine than for other engines. However, the other factors discussed in the ranking and weighting section, make Saturn a highly viable candidate.

One of the principal components for the Rampressor Turbine engine is the impeller in front. This is currently shown in the design because it is assumed that the Rampressor in its current design state cannot self-aspirate. Depending on the data being collected with the Rampressor 1 design, this decision needs to be revisited. An early design analysis for the four engines for the impeller was carried out through a consultant and from his work, the only two turbines for which an impeller could be laid out for were the Solar Spartan and the Solar Saturn turbines. For the other two turbine engines, the impeller rotor designs became highly risky (meaning that design guidelines had to be violated to come up with reasonable impeller rotor designs). This led Ramgen with only two viable candidates namely Solar Saturn and the Solar Spartan gas turbine.

One more important consideration for designing the Rampressor Turbine is the availability of test data for the Rampressor part of the engine. Since Rampressor 1 testing has been completed, it is advantageous to design a system that has a rotor which is similar to the Rampressor 1 rotor. The Solar Spartan gas turbine Rampressor layout is

very similar while the Solar Saturn turbine engine Rampressor layout is vastly different, in their operating characteristics, to that of the Rampressor rotor. All these important factors, in addition to the resource, cost and schedule, make the Spartan gas turbine, the turbine engine of choice for the Rampressor turbine 1 program. The next few sections deal with the layout of the Spartan Rampressor Engine.

4.7 Aero-Mechanical Design of Spartan Based Rampressor Turbine

The layout of the Spartan gas turbine based Rampressor Turbine is shown in figure 4.5. The components that are within the dashed line box are the components that exist in the Spartan engine and components outside of the same are what need to be designed by Ramgen for an overall Rampressor Turbine engine. The part that Ramgen needs to concentrate and design to fit up with a Spartan gas turbine is shown in figure 4.6. As can be seen, the principal components include the : Impeller rotor, de-swirl vane, pre-swirl nozzle, Rampressor rotor, a diffuser, a de-swirl vane and the duct that takes the flow from the deswirl to the combustor section of the Spartan engine. The details of some of the individual components are discussed in the next few sections.

4.8 Impeller Design

The impeller layout is shown in figure 4.7. It has a 2.94 inch tip radius and spins at approximately 950 ft/s and needs to be designed for at least a 91% efficiency. The constraints with which this design has been evolved is the engine rpm of 37059 and a mass flow rate 5 lbm/s.

4.9 Rampressor Rotor Design

The Rampressor rotor for the Spartan engine based Rampressor turbine has the following preliminary details:

- Rotor size
- Tip Speed
- Relative Mach Number
- Pressure Ratio
- Outflow Mach Number
- Pre-Swirl Mach Number
  - Angle

4.10 Rampressor Turbine Stationary Diffuser

The current Spartan compressor discharge flow has no swirl component and is mostly axial flow discharging at a low mach number of 0.1. The Rampressor rotor design conditions for the Rampressor turbine engine has flow exhausting at a high swirl angle and a high mach number. These are design estimates at an early stage and will be verified and modified according to the test data generated from Rampressor 1 test data, which is currently operational at Boeing. Assuming that the estimates are correct, there is

significant diffusion and turning required in the flow exhausting out of the Rampressor rotor. In order to assist in that, a tandem vane cascade with a potential for a hybrid “Pre-Diffuser” volute section is shown in the layout shown in figure 4.8.

4.11 Potential Design Point Performance gains

A preliminary estimate of the efficiency of the individual components of the Spartan gas turbine based Rampressor Turbine was made based on certain assumptions. This section discusses those performance estimates. The impeller has 1.1 pressure ratio and is considered to be 91% efficient and the preswirl nozzles are expected to have a 2% pressure loss at design point. The Rampressor rotor is assumed to have a 92% efficiency without taking into account the tip clearance system loss and the mechanical losses. The existing Spartan compressor efficiency is approximately 75%. The de-swirl vanes and the diffusers are assumed to have a 4% pressure loss and the existing performance estimates for the combustor and the turbine are used for estimating the overall cycle efficiency. These component efficiencies cause an 18% overall simple cycle thermal efficiency which is 3-4% increase from the existing Spartan simple cycle efficiency.

This section discusses some of the off-design and starting/un-starting issues that the Spartan based Rampressor Turbine might face. Since the mean line Mach number is very similar to the Rampressor 1 hardware, the starting issues are expected to be very similar to Rampressor 1. So, a good understanding of this issue will be available after the completion of the Rampressor 1 testing. Then the detail design of the Rampressor turbine is ready to be carried out. Also, a better understanding of the unstart./surge characteristics of the Rampressor rotor will be available (from the test data of the Rampressor 1 rotor) before different components are matched to the Rampressor rotor.

The significant issue with the starting of the Rampressor turbine is the matching of various components during the startup process. There might be a need to bleed the flow between the impeller and the Rampressor rotor as well as bleed the flow between the Rampressor rotor and the combustor for starting purposes. More detailed information regarding this will be available during the detail design phase of this project. One of the ways that this particular issue can be better understood and resolved is by de-coupling the compressor part from the combustor and turbine. This intermediary step might be necessarily required to better understand the characteristics of different components before a final coupled Rampressor turbine is operated. Ramgen is seriously considering proceeding with such a design in the above mentioned fashion to give us the best chance of successful demonstration of the Rampressor turbine engine.

The starter motor required for the Rampressor turbine is estimated to be able to deliver atmost 448 hp at 37000 rpm. This is estimated based on assuming that the starter motor is required to take the compressor to full speed and full pressure ratio, before it can be dropped off from the starting sequence. A more realistic case would be where the power generated from the turbine is greater that that required by the compressor, which is expected to happen at around 18500 rpm and that which requires 112 horse power. The drive motor that is currently available with Ramgen can support worst case scenario

requirement, but its availability at the time of requirement is in question. Mechanically, the two significant issue are the tip speed of the Rampressor rotor and the rotor dynamics of the system.

4.12 Test Rig Cost

The components and the requirements identified, helped in estimating the costs for the Spartan Engine based Rampressor turbine test rig. The details of this are attached in the table captured in Figure 4.9. The table indicates that the initial cost estimate for the rig is \$706,000. This cost estimate has been arrived at based on Ramgen's prior experience with building the Rampressor 1 test rig. Since the size of the Rampressor part of the Rampressor turbine test rig is similar to the Rampressor 1 test rig, the cost estimates should be similar. The overhauled Spartan engine package has been estimated to be about \$40,000 and the drive, if a new one is required for this test rig, is expected to cost around \$50,000. These two component costs have the highest uncertainty in their estimates because of the availability factor. Ramgen has used the best available information, to date, to estimate these costs.

4.13 Test Cell Cost

The other part of the cost structure is the estimate for test cell cost. In order to obtain this, a number of test cells were approached with a set of initial requirements. The list of initial requirements that were specified for the test cells were

- i. A test cell large enough to accommodate our test rig with an approximate size of 150 inches long, 60 inches high and 40 inches wide.
- ii. Availability of liquid fuel like diesel or kerosene, as well as natural gas.
- iii. Test cell capable of dealing with approximately 6 lbm/s of exhaust air at 1200 F.
- iv. A test cell capable of handling a rig weighing approximately 5000 lbm.
- v. A vacuum system capable of bleeding approximately 1.5 lbm/s @ 5psia at 200F.
- vi. Secondary air requirement of approximately 6 lbm/s at 100 psig at 280 F for the drive turbine.
- vii. Tertiary air requirement of approximately 3.0 lbm/s at 100 psig at 70 f for secondary flow.
- viii. Water cooling for oil cooling purposes.
- ix. An oil system capable of delivering 8gpm at 65 psia that conforms to MIL-G-6529 Type III Spec.
- x. Facility should have appropriate inlet ducting to feed the air to the drive turbine.
- xi. Exhaust ducting to dump hot air at conditions mentioned before.
- xii. Load banks capable of using up to 200kW of power generated by the Rampressor turbine or a dynamometer of that size, for that same power requirements.
- xiii. A video taping of test data.

- xiv. Data Acquisition details regarding the pressure, temperature and mass flow rate measurements.
- xv. A build up area for assembling the engine together.
- xvi. Test cell availability from Jan 1, 2005 for 8 months including two months of build up time and 6 months of testing.

Based on the above mentioned requirements, the cost estimates provided by different test cells are shown in table of Figure 4.10. The test cell costs range from \$500,000 to \$2,000,000. As can be seen, the Boeing Test Cell at Seattle provides the most cost efficient choice amongst all the test cells.

Finally, the different test rig configurations based on availability of funds for this program was considered. There is interest from DOD on a fuel cell application of the Rampressor Turbine. Depending on what becomes available, it was decided that the Rampressor Turbine will be designed in a way so as to be easily adaptable to a hybrid fuel cell application, if possible. Such an application requires an external combustor to be integrated to the Rampressor based Spartan turbine, and not directly coupled with the engine as was previously shown in the figure 4.5.

4.0 Conclusion

The early stage design layout of the Spartan Engine based Rampressor Turbine is complete. From a pool of 22 different engines, the Spartan gas turbine was chosen as the most appropriate design for integrating the Rampressor to an industrial gas turbine. The reasons for this selection and the advantages of such a selection were discussed. The advantages of testing a de-coupled design of the same were also discussed and during the 2004-05 fiscal year, the final design of this design is expected to be complete.

Figure 4.1. Ramgen's Technology Development Plan

Methodology

Figure 4.2. Engine Selection Process Flow Chart

Results of Engine Selection Process

Figure 4.3. Engine Selection Process Scoring

Engine Layout Details

Engine	Engine RPM	Required Pressure Ratio	Mass Flow (lbm/s)	Tip Speed (ft/s)	Impeller PR	PSN Exit Mach No.	Rampressor Tip Radius (inches)	Rampressor Centerline Radius (inches)	Rotor relative Mach number	DeSwirl Vane Inlet Mach	DeSwirl Vane Inlet Angle (deg)
Garrett 36-200	62400	4.2	2.80	1774	1.2	0.39	3.26	2.92	1.54	0.7	-80
Solar Saturn	22516	6.27	13.45	1785	1.1	0.56	9.08	8.35	1.92	0.68	-83
Solar Spartan	37059	4.2	5.00	1756	1.1	0.33	5.43	4.96	1.56	0.71	-82
Sundstrand APS2000	45000	4.3	7.00	1795	1.2	0.46	4.57	4.03	1.58	0.7	-79

Figure 4.4. Critical Parameters for Four Engines

Rampressor- Turbine 1 X-Section

Figure 4.5. Rampressor Turbine 1 Layout

Spartan Aero Components

Figure 4.6. Spartan Rampressor Turbine Aero Components

Impeller Design

- 1.1 PR Design by A. Stone
  - 2.94" Tip Radius
  - 950 ft/sec tip speed (550 ft/sec outflow) -> Needs to be de-swirled?
  - Want ~91% efficiency

Figure 4.7. Impeller Design for a Spartan Rampressor Turbine

Rampressor Turbine Diffuser Design

Current Conceptual Layout

Duct Height Must Be Managed with Conservation of Angular Momentum to Avoid Increasing Swirl with Duct Radius

Figure 4.8. Rampressor Turbine Diffuser Design

Estimated Test Rig Cost Summary

Component	Description	Estimated cost	Estimated Cost Basis
Peripheral Equipment			
Drive	Electric Direct Drive	\$50,000	Probably use Bsi Drive again
Lube	Hydraulic components	\$4,000	Only needed for Rampressor Bearings
Controls		\$15,000	Upgrade/Modification to existing
Instrumentation		\$75,000	Puts total Instrumentation/Controls at ~\$100K
Rig Table	Mounting Stand	\$5,000	Engineering estimate
Containment		\$1,000	Engineering estimate
Peripheral Equipment		\$35,000	Load banks/gearbox
Analog to Digital Equipment		\$10,000	Engineering estimate
Misc.	Nut, Bolts, O-rings, Tubes, etc.	\$25,000	Engineering estimate
Impellor Rotor		\$15,000	Engineering estimate
Test Unit			
Rotor			Engineering estimate
Forgings	McWilliams	\$30,000	
Rotor machining	Sermatec	\$20,000	
Rotor Segments	3 sets of flow paths	\$168,000	
Unit (less Rotor)			
Static Hardware		\$150,000	Engineering estimate
Pre Swirl (2 sets)		\$16,000	
De-swirl (2 sets)		\$32,000	
Snail-shell Diffuser		\$10,000	
Bearings		\$5,000	
Turbine and Combustor	Commerical unit for purchase	\$40,000	Online quotes
Total Turbine Rig		\$706,000	

Figure 4.9. Estimated Test Rig Cost for a Spartan Rampressor Turbine

Ramprossor- Turbine Test Facility Costs

Test Cells	Boeing (WA)	Wyle Labs (LA, CA)	Alturdyne (SD, CA))	NRC (ON)	Standard Aero (Winnipeg)	MTSI (OH)	Woodgroup (LA)	Aero Systems Eng (MN)	China Lake (CA)	NASA Glenn
Setup Costs										
Cell Mods	\$114,000			\$115,000	\$500,000	\$480,000	\$485,000	\$522,000		\$1,200,000
Exhaust Duct	\$10,000									
LN2 system	\$5,000									
Load Banks	\$7,000									
Facility Usage	\$2,000									
DAS										
	\$0									
Testing (6 months)	\$300,000			\$450,000	\$490,000	\$322,500	\$244,000	\$244,000		\$1,000,000
Fuel Costs (6 months)	\$30,000			\$30,000	\$30,000					
Travel		\$80,000	\$80,000	\$80,000	\$80,000	\$80,000	\$80,000	\$80,000	\$80,000	\$80,000
Total Cell Cost	\$468,000	\$80,000	\$80,000	\$675,000	\$1,100,000	\$682,500	\$809,000	\$946,000	\$80,000	\$2,280,000

- Preliminary testing costs range from \$500,000 to \$2,000,000.
- Boeing is the preferred location.

Figure 4.10. Estimated Test Facility Costs for testing a Spartan Ramprossor Turbine

CHAPTER 5: VORTEX COMBUSTOR DESIGN, FABRICATION & TESTING (TASK 16)

This report has been completed and submitted to the DOE and CEC sponsor under separate cover. It included in its entirety as an Appendix; only the page numbers have been changed in the report to reflect its location within this BP4 Final Report document.

CHAPTER 5 APPENDIX

DEMONSTRATION OF A LOW EMISSION ADVANCED VORTEX COMBUSTOR

Final Report

Prepared For:

California Energy Commission
Public Interest Energy Research Program

Prepared By:

National Energy Technology Laboratory
(U.S. Department of Energy)

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CONSULTANT REPORT

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In addition to the previous list of people deserving acknowledgement for their role in this project, the following people deserve some special recognition, since they have been working "where the rubber meets the road." **Mark Tucker**, the lead technician on this project, has provided a judicious blend of technical expertise, sound operational judgment, and humorous anecdotes that have made this project both successful and enjoyable to all involved. **Mike Dera**, the lead technician from a different project, has stepped into this project to fulfill several critical roles, such as the main point of contact for all procurements actions, managing the utility resource schedule, and several other roles that have been filled with professional fervor and personal dedication. **Randy Barnes, Don Ferguson, Steve Beer, Scott Clingenpeel, and Paul Sellers** have been involved with ensuring that the gas analysis equipment is maintained, checked, calibrated, and re-calibrated. **Mike Bierer** has provided mechanical expertise for a wide variety of tasks, and in spite of being the target of the occasional practical joke; he maintains a very professional, results-oriented demeanor. The fabrication effort at NETL has been lead by **Brad Linton**, who is the type of person that always welcomes a challenge. **Mehrdad Shahnam** has been the key technical resource for the CFD simulations. His enthusiasm, professionalism, and attention to detail have made the CFD analyses an innovative and insightful part of this effort.

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Preface

The Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program, managed by the California Energy Commission (Energy Commission), annually awards up to \$62 million to conduct the most promising public interest energy research by partnering with Research, Development, and Demonstration (RD&D) organizations, including individuals, businesses, utilities, and public or private research institutions.

PIER funding efforts are focused on the following six RD&D program areas:

- Buildings End-Use Energy Efficiency
- Energy-Related Environmental Research
- Energy Systems Integration
- Environmentally-Preferred Advanced Generation
- Industrial/ Agricultural/ Water End-Use Energy Efficiency
- Renewable Energy Technologies

What follows is the final report for the “Demonstration of a Low Emission Advanced Vortex Combustor” Project, Contract No. 500-02-025, conducted by the National Energy Technology Laboratory (United States Department of Energy). The report is entitled “Demonstration of a Low Emission Advanced Vortex Combustor.” This project contributes to the PIER Environmentally Preferred Advanced Generation program.

For more information on the PIER Program, please visit the Energy Commission’s Web site at <http://www.energy.ca.gov/research/index.html> or contact the Energy Commission’s Publications Unit at (916) 654-5200.

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Abstract

New electric power generators must meet challenging regulatory emission requirements. Although gas turbines have historically been the preferred power generation approach, in some areas of California, even the cleanest and most efficient gas turbine combustion technologies are not capable of achieving emission requirements without costly after-treatment devices. This report describes an alternative gas turbine combustion technology that can achieve less than 3 parts-per-million NO_x emissions (at 15% oxygen) with combustion efficiencies in excess of 99 percent. Other important advantages of the Advanced Vortex Combustor concept have also been demonstrated in this effort. For example, the Advanced Vortex Combustor is dynamically stable over a wide range of operating conditions, which makes this approach significantly more attractive than a lean premixed swirl-stabilized combustion approach. In addition, the pressure drop associated with this combustion approach is significantly lower than a conventional gas turbine combustor, which translates into an improvement in the overall cycle efficiency. Although these results are encouraging, further improvements can be realized. Therefore, interested parties in both the gas turbine and industrial burner communities must be identified, and future commercial development opportunities must be pursued.

Executive Summary

In order to achieve regulatory emission requirements, many advanced gas turbine systems are required to install exhaust after-treatment devices that add considerable complexity to the system and increase the cost of electricity for the consumer. The primary pollutants that must be minimized are oxides of nitrogen (NO_x) and carbon monoxide (CO). The combustor design is critical for achieving low emissions in a gas turbine system, and this effort will focus on developing a novel combustor concept called the Advanced Vortex Combustor (AVC) that in previous testing has proven capable of controlling both of these pollutants.

The conventional swirl-stabilized combustor designs used in virtually all gas turbines today have been the standard for decades. Although recent advances in lean premixed combustors have resulted in significant improvements in performance, these combustors have not achieved NO_x emissions levels of less than 3 parts-per-million by volume (ppmv) without the use of costly exhaust after-treatment devices.

This project will investigate the use of an AVC to achieve NO_x emissions of less than 3 ppmv (at 15% oxygen) in a nominal 1 MW_{th} laboratory-scale test. This is a collaborative effort between Ramgen Power Systems (Bellevue, Washington) and the U.S. Department of Energy's National Energy Technology Laboratory (Morgantown, West Virginia). Both organizations have some operating and test experience with the AVC concept, but achieving NO_x levels of less than 3 ppmv (15% oxygen) at realistic gas turbine operating conditions is a challenging goal.

This effort started with the analyses of several preliminary concepts. The design, fabrication, and testing portions of this program have followed, and the entire project has been completed in about 17 months. In addition to completing this effort on-schedule, the project has been completed under-budget due in a large part to in-kind contributions by the NETL.

The results show that the AVC approach is indeed capable of achieving less than 3 ppmv NO_x levels at combustion efficiencies in excess of 99%. These results have been obtained at realistic industrial gas turbine operating conditions (i.e., 625°F air preheat and 147 psia). The lowest NO_x numbers measured in these tests are less than 2 ppmv ($\pm$ 1.2 ppmv) corrected to 15% oxygen basis. However, the combustion efficiency at this point is not certain, since the full-scale range (i.e., 500 ppmv) of the CO analyzer had been exceeded. Nevertheless, it is believed that significant improvements in combustor performance can be achieved.

In addition to the excellent emissions performance, several other important observations have been made throughout the course of this project. These observations are listed below and will be described briefly in the following paragraphs:

- The pressure loss across the AVC is lower than conventional combustors.
- The air-cooling approach is effective at maintaining adequate liner temperatures.

- The emission performance can be improved by using less cooling air than the original design.
- The cavity equivalence ratio can be used to significantly reduce the NO_x emissions without adversely affecting the CO emissions.
- No combustion oscillations have been observed in this effort.

The combustor pressure drop is significantly lower than conventional swirl-stabilized combustors. For swirl-stabilized combustors, a three percent combustor pressure drop is typically the accepted industry standard, and the average bulk gas velocity through the premixer is on the order of 150-200 feet-per-second. In this test program, the AVC combustor pressure drop varied from a minimum of 1.5% at 235 feet-per-second bulk velocity to a maximum of 2.8% at 265 feet-per-second. Therefore, the lower pressure drop characteristics of the AVC concept should translate into efficiency improvements in an overall gas turbine cycle.

The impingement/effusion air cooling design used in this effort has been very effective at maintaining liner temperatures below the design limit. In fact, the amount of cooling air has a dramatic effect on the emissions performance. As the cooling air pressure drop is reduced from the design point of 4% to 2%, the measured CO emissions drop by more than an order of magnitude. With a lower cooling air pressure drop, the combustor can operate at leaner conditions. The wider operating range allows lower NO_x emissions to be achieved.

The cavity equivalence ratio has a profound effect on the emissions performance of this AVC design. A 15% reduction in the cavity equivalence ratio reduces the NO_x emissions by approximately 50% (or about 3ppmv), without a significant change in the CO emissions. Further reductions in NO_x emissions may be possible by using an approach that reduces the cavity equivalence ratio further without causing flame extinction.

One of the most interesting aspects of this demonstration has been the lack of any significant combustion oscillations. Combustion oscillations are a significant problem for swirl-stabilized combustors, particularly for lean premixed combustors. The AVC design demonstrated in this effort is a lean premixed design, however, no oscillations have been observed. The reason for this observation is not clearly understood at this time.

In summary, this report describes the approach taken to achieve less than 3 ppmv NO_x emissions in an Advanced Vortex Combustor design. The test results show that in addition to achieving NO_x emissions that are less than 3 parts-per-million at realistic gas turbine combustor conditions, the Advanced Vortex Combustor concept offers other potential advantages. For example, this combustor design operates at a lower combustor pressure drop which translates into improvements in the overall gas turbine cycle efficiency. In addition to this, the Advanced Vortex Combustor demonstrated in this effort has not exhibited any significant combustion oscillations. Although the dynamic stability was not an original goal of this program, this may be one of the most significant achievements of this effort.

1.0 Introduction

Emission requirements for stationary power applications are low enough in some areas of the country to prevent even state-of-the-art gas turbine combustors from meeting the regulated levels without the added expense of exhaust after-treatment devices. These exhaust after-treatment devices are expensive and result in a higher cost of electricity for energy consumers. In addition, the after-treatment process must be closely monitored to minimize the release of harmful chemicals, such as ammonia, into the atmosphere, which has environmental consequences of its own.

Recognizing the problems associated with producing electrical power at ultra-low emission levels, researchers and scientists are developing some alternatives to swirl-stabilized lean-premixed combustors, such as catalytic combustion and surface-stabilized combustion systems. However, these alternatives are relatively immature for commercial acceptance by the gas turbine community.

AVC (AVC) is among the relatively immature concepts, and has not been widely accepted for stationary gas turbine applications. The AVC concept involves a different mechanism for stabilizing the primary fuel-air mixture entering the combustion chamber. The purpose of this effort is to demonstrate a low-emission AVC concept at realistic gas turbine conditions. It is believed that a successful demonstration of an ultra-low emission AVC concept will facilitate commercial acceptance in the gas turbine industry.

In a related effort, the U.S. Department of Energy's (DOE) Turbines Program has established an aggressive NO_x emissions goal of less than 3 ppmv NO_x emissions for future turbine power generation. These future turbine power plants may be required to operate on fuel compositions that range from hydrogen to coal syn-gas. The inherent sensitivity to fuel variations is one of the problems of existing state-of-the-art combustors. In order to achieve the goals of the DOE Turbines Program, improved methods may be required to stabilize the reaction zone in a gas turbine combustor. The AVC concept offers several potential advantages over more conventional swirl-stabilized combustor approaches for fuel-flexible combustor operation.

1.1 Background

The AVC concept was originally conceived at the Air Force Research Laboratory in the early 1990s. The phenomena of "locked" or "trapped" vortices has been known to reduce aerodynamic drag for years (Little and Whipkey 1979; Katta and Roquemore 1999), and the geometric features required to produce a "locked" or "trapped" vortex are the same features used to minimize drag. In the early 1990s, Hsu et al. (1998) proposed using this feature to stabilize reactions in gas turbine combustors for aero-propulsion applications. Since then, several papers (Burrus et al. 2001; Roquemore et al. 2001) and patents (Roquemore et al. 1999) have described the results from using this AVC concept to achieve stable and low emission combustor results. However, in the years prior to 2002, AVC efforts have focused primarily on aero-propulsion applications.

The AVC concept is fundamentally different from conventional swirl-stabilized combustors, since the flame stabilization mechanism is completely different. The swirling flow of a conventional gas turbine combustor produces a reverse flow along the axis—or core—of the main flow. This back-mixing of hot products ignites the incoming fuel-air mixture and sustains continuous combustion. It is believed that this method of flame stabilization may be more susceptible to process upsets and instabilities. Although swirl-stabilized combustors have been used for decades, achieving less than 3 ppmv NO_x emissions is complicated by the sensitivity to instabilities and sudden flame extinction near the lean blow-off limit.

In contrast, the AVC flame stabilization—the re-circulation of hot-products into the incoming fuel-air mixture—is accomplished by a stable vortex that is produced adjacent to the main fuel-air flow path. The vortex behavior is virtually independent of the main flow characteristics, which makes this method of flame stabilization so attractive. In order to work properly, the AVC concept must provide flame stabilization by lateral mixing from the vortex region into the main flow.

In an AVC concept, the re-circulation of hot products into the main fuel-air mixture is accomplished by incorporating two critical features. First, a stable recirculation zone must be generated adjacent to the main fuel-air flow. If the vortex region, or cavity region, is designed properly, the vortex will be stable and no vortex shedding will occur. This stable vortex is generally used as a source of heat, or hot products of combustion. The second critical design feature involves transporting and mixing the heat from the vortex, or cavity, region into the main flow. As described in previous work (Roquemore et al. 1999; Roquemore et al. 2001), this is generally accomplished by using wake regions generated by bodies, or struts, immersed in the main flow. This approach ignites the incoming fuel-air mixture by lateral mixing, instead of a back-mixing process. By using geometric features to ignite the incoming fuel-air mixture, instead of pure aerodynamic features, the AVC concept has the potential to be less sensitive to instabilities and process upsets. This is particularly important near the lean flame extinction limit, where small perturbations in the flow can lead to flame extinction.

In an effort prior to this project, Ramgen Power Systems utilized an AVC concept to achieve 9 ppmv NO_x emissions on a lean premixed AVC (Bucher et al. 2003). In an independent effort, the National Energy Technology Laboratory investigated a non-premixed Rich-burn, Quick-mix, Lean-burn (RQL) version of the AVC concept to investigate the ability to achieve low emissions (Straub et al. 2000; Straub et al. 2003) and reduce conversion of fuel-bound nitrogen to oxides of nitrogen. Therefore, both organizations have some operating and test experience with the AVC concept, but achieving NO_x levels of less than 3 ppmv (15% oxygen basis) at realistic gas turbine operating conditions is a challenging goal.

The current project is a joint effort between the National Energy Technology Laboratory (NETL) and a commercial partner, Ramgen Power Systems (RPS). Successful completion of this project will result in a better fundamental understanding of the AVC, which will be shared with the public. Although the AVC has the potential to be more compact,

have better flame stability, and have a lower cost than other alternatives, the following issues must be addressed to enable widespread use.

- Fundamental understanding of the key variables and the design tools is required to build a low emission AVC.
- Testing (laboratory and field), particularly for the stationary power market, must be completed to enable the acceptance of this technology by the industry.
- The perception that this concept is too risky must be overcome before gas turbine manufacturers will commit resources to develop an AVC prototype.

1.2 Project Objective

Although Ramgen Power Systems has already used the AVC concept to achieve 9 ppmv NO_x levels, this effort will focus on achieving less than 3 ppmv NO_x at realistic gas turbine operating conditions. Admittedly, this is an aggressive goal and will require a NO_x improvement that is nearly three times better than previous results.

2.0 Project Approach

In order to achieve the goals of this project, an AVC was designed, fabricated, and tested. These tests focused specifically on the combustor performance, and the testing occurred in an existing NETL test facility. The combustor design was based on the previous Ramgen AVC combustor that was sized to fit the Ramgen Engine concept. Portions of the previous combustor design were reused, or reworked, where appropriate. The test facility was also modified to accommodate the combustor design, as appropriate.

The following approach was implemented to achieve the goals of this program. A detailed description for each of these tasks is included in Appendix 1. A brief summary, key design features, and facility requirements of each technical task are described below.

2.1 Summary of Technical Tasks

The technical tasks performed under this contract are listed and briefly described below and shown in Figure 1. In summary, the approach centered around developing as much background information as possible using various engineering and computer models to supplement the conceptual design process. Since very few design rules exist for building an AVC to achieve less than 3 ppmv NO_x levels, the modeling, design, and fabrication tasks were critical for this effort.

Figure 1: Project schedule and task list

2.1.1 Computational Fluid Dynamic (CFD) Analyses

Most of this task was completed during the conceptual design stage to help guide the design and experimental approach. However, a portion of this effort was completed near the end of the program to actually model the combustor configurations that were fabricated.

2.1.2 Conceptual Design and Test Plan Development

This task was completed in November 2003.

2.1.3 Detailed Design

This task was completed in February 2004.

2.1.4 Hazard Analysis and Safety Review

This task was completed in October 2004, and the required operating permit was approved as a result of this task.

2.1.5 Complete Fabrication and Process Drawings

This task was completed in May 2004, and a letter was submitted to the Commission Contract Manager.

2.1.6 Complete Fabrication of Test Hardware

This task was completed on October 19, 2004 with the delivery of the last critical components required for combustor assembly and testing.

2.1.7 Installation and Checkout

This task was completed on November 9, 2004, at the conclusion of our third day of shakedown testing.

2.1.8 Advanced Vortex Combustor Test

This task was completed in December 2004.

2.1.9 Data Evaluation and Post-Test Analysis

This task was completed in December 2004.

2.1.10 Technology Transfer Activities

A technology transfer plan was developed and is included in this report. This task was completed in December 2004.

2.2 Test Facility Requirements

In order to meet the objectives of this project, the test facility must be able to conduct combustor tests at a pressure of 10 atmospheres with an air preheat of approximately 625 °F. The total mass flow of air required is on the order of 1.5 pounds per second. The existing NETL test facility met these requirements, but some modifications to the test section were required to achieve the overall goals of the program. These test facility modifications are described below.

2.2.1 Key External Design Features

The existing test facility was fabricated using 24-inch pipe sections and flanges. A cut-away view of the test rig is shown in Figure 2 and Figure 3. Although the support structure is not shown in the figure, the rig was suspended several feet above the floor. This allowed the combustor components to be fabricated outside the vessel and lifted into position for testing. In this project, an existing test section was utilized, which has four 7 x 12-inch optical access ports. This is the same test section that was used in the DOE Simulation and Validation Studies Project. In order to use the existing test rig and facility, some modifications were required. These modifications are described below in more detail.

Figure 2: Cutaway view and photo of NETL test section and top spacer

2.2.1.2 Bottom Spacer

In order to position the combustor at the same height as the viewports in the test section, a spacer section was added to the bottom of the test section. This spacer shown in Figure 5 has the same inside diameter as a 24-inch pipe, and the top and bottom surfaces incorporate standard ring joint seals. This spacer section was also used to insert most of the instrumentation into the vessel through 32 instrumentation ports located around the circumference. This spacer section was attached to the bottom flange, and the combustor was assembled inside this spacer section prior to raising the entire assembly into the test section.

Figure 5: Conceptual design for bottom spacer section

2.2.1.3 Test Section

As previously mentioned, an existing test section was utilized in this project. This test section is shown in Figure 2, but the test section was originally rated for 350 psig at 400 °F. Since cooling air to the combustor liner must contact the inside surface of this test section, this component was re-rated for a higher temperature. In order to re-rate this pressure vessel boundary, detailed thermal and stress analyses were completed. According to the ASME Boiler and Pressure Vessel Code, this test section is capable of operating safely at the higher temperature rating without exceeding the maximum allowable stress.

2.2.1.4 Top Spacer

Immediately above the existing test section, another spacer section was installed. This spacer provided a water-cooled surface for sealing some face-type o-rings on the combustor exhaust seal, as shown in Figure 6. In addition to providing an interface for the combustor exhaust seal, this top spacer also provided several access points for a gas

sampling probe, and exhaust gas thermocouples, as well as a smooth transition into the existing downstream pressure vessel components (i.e., the four-way cross shown in Figure 2).

Figure 6: Conceptual design for top spacer section

2.2.2 Key Internal Design Features

In addition to the modifications required to the external pressure vessel, the internal components were designed, fabricated, and tested as part of this project. These internal components are shown in Figure 7, and these components are described in more detail below.

Figure 7: Model of conceptual AVC design

2.2.2.1 Inlet-Air Plenum

The inlet plenum accepts air from the facility, conditions the flow to provide a steady, well-developed velocity profile, and directs the air toward the entrance of the premixer (see Figure 8). The plenum is approximately seven inches in diameter and roughly 15-inches long. It bolts directly to the pressure vessel bottom flange.

Figure 8: Photo of inlet-air plenum

In an attempt to verify the flow uniformity and turbulent intensities at the exit plane of the plenum, some cold flow measurements were made using a Pitot-tube and a hot-film anemometer. It should be noted that these measurements were made without the smooth bell-shaped inlet to the premixer. Figure 9 shows the results from Pitot-tube measurements across the exit plane of the plenum. In Figure 9a, some flow non-uniformity can be observed, particularly between opposite corners of the contour plot. After some investigation, some imperfections near the edge of the perforated plate were identified and corrected. After correcting these edge effects, the velocity distribution became much more uniform (see Figure 9b).

Figure 9: Velocity contours at the exit of the plenum: (a) initial as-fabricated results and (b) corrected edge effects

The turbulent intensity levels near the plenum outlet were also measured using a hot-film anemometer. The positioned of the perforated plate were selected to produce a turbulent intensity of less than 5 percent. The hot-film anemometer results are shown in Figure 10. Note that the turbulent intensity levels near the center of the exit are in the 2 to 4 percent range which is close to the design value. The measured intensity levels near the outside walls of the plenum exit were eliminated during testing by the smooth bell-shaped inlet that extends roughly 1-inch into the plenum. In summary, this cold flow data provides some confidence that the flow entering the premixer are uniform and the turbulent intensities are less than 5 percent.

Figure 10: Contour plot of turbulent intensities at the exit of the plenum

2.2.2.2 Premixer

Extending one-inch into the plenum, the premixer inlet collects air from the seven-inch diameter plenum, and transitions the flow smoothly to the rectangular cross-section of the combustor. A row of fuel posts injects fuel transversely into the main air flow in the premixing region. The fuel jets were designed to ensure complete mixing before the fuel-air mixture reaches the combustion area of the burner.

There are two rows of fuel posts at an axial separation of about one-inch. This allowed for ‘tuning’ the distance between the fuel injection plane and the combustion zone. This feature provided the opportunity to change the fuel transport time between the point of fuel injection and the flame. This time-scale becomes critical if an unstable operating condition is encountered. Additionally, the premixer is mounted between two half-inch spacer plates. If necessary, these plates can be moved to further change the fuel transport times.

2.2.2.3 Combustor

2.2.2.3.1 Vortex (Cavity) Section

As the flow approaches the combustion zone, a rounded centerbody, or bluff-body, divides the flow along the center. The air then passes over the vortex cavity and into the exhaust section. Combustion was sustained and protected in the vortex cavity. Mixing between the cavity and the main flow caused hot exhaust products from the cavity to ignite the main flow. Flameholders were used at each edge of the vortex cavity to 'pump' hot combustion products from the cavity into the main flow.

Fuel and air can be injected into the vortex cavity from either the bluff-body, or from the cavity side of the AVC module. In addition, two AVC modules were fabricated to inject fuel-air mixture into the cavity and produce different flow patterns in the cavity.

The first configuration tested was identical to that used in previous AVC testing, and provided a baseline comparison to the previous test and a benchmark for all changes tested in this experiment.

2.2.2.3.2 Combustor Liner

The combustor liner separates the combustion zone from the cooling air supply and consists of a thin metal wall with a ceramic thermal barrier coating. A combined impingement/effusion air cooling system keeps the liner temperature within limits. The liner was designed in a modular fashion to allow easy maintenance, replacement, and modification of the individual liner wall components.

The liner contains mounting bosses for the bluff body injection module, the AVC module, a spark igniter, and an optical window. The spark igniter is an off-the-shelf gas turbine igniter, and the optical window is a quartz plate mounted in the liner wall. The optical window allowed observation of the vortex region during operation, and provided access to visually inspect internal components.

2.2.2.3.3 Exhaust (CO Burnout) Section

After exiting the combustor liner, the hot exhaust products will enter the carbon monoxide burnout section to allow completion of the $\text{CO} + \text{O} \rightarrow \text{CO}_2$ reaction. The burnout section increases the combustor residence time to provide more time for this reaction to occur. In addition, the larger-diameter burnout section decreases the surface area to volume ratio, thereby decreasing the impact of wall quenching on the combustion kinetics. Table 1 shows the effect on surface area to volume ratio, velocity and burnout time of the burnout section, and compares it to a typical industrial gas turbine combustor.

Table 1: Surface-to-volume ratio for various combustor designs

Configuration	Area Change	Surface/ Volume	Velocity (fps)	Burnout Time (ms)
Industrial turbine	8:1	0.30	102	16
SimVal	10.2:1	0.58	41	20
GASL burner	1:1	1.65	250	8
New burner only	1:1	1.65	250	4
Burnout section only	7.8:1	0.58	32	32

The burnout section was water-cooled via a triple-wall, double-pass design nearly identical to the existing NETL Simulation Validation Combustor. At the burnout section exit, the cooling air plenum is sealed from the combustion products using a spring-loaded o-ring seal similar to the existing NETL approach.

2.2.3 Gas Sampling System

2.2.3.1 Sample Probe

The sample probe collects gas from the hot exhaust region and transports these gaseous products to the sampling system. Since the temperatures near the probe could be in excess of 2,000 °F, the probe must be cooled. The sample probe used in this effort was constructed using two different sizes of stainless steel tubing (i.e., ¼-inch and ½-inch). The gas sample is transported through the ¼-inch diameter tubing, and the cooling water flows through the annulus created by the concentric tubes (see Figure 11).

The gas sample is collected through five (0.038-inch diameter) holes that are located along the leading edge of the sample probe. These holes are spaced to provide an area-weighted gas sample at the exit of the CO burnout section (see Figure 12). Since the test rig operates at elevated pressure, the gas sample flows through the holes in the probe to the sampling system. Condensate is particularly important to avoid for single-digit NO_x measurement, due to the solubility of NO₂ in water. If NO₂ is dissolved in the water that is removed from the sample, it is possible to bias the NO_x measurements. Our attempts to minimize the loss of NO₂ are described in the next section.

Figure 11: Schematic of the sample probe design

Figure 12: 5-hole sample probe layout

2.2.3.2 Sample Conditioning and Pressure Control System

The sample line was electrically heat-traced once it left the water-cooled sample probe to prevent condensation of water vapor. A schematic flow diagram for the gas sampling system is shown in Figure 13. A small pressure control valve was used to vent excess flow through the sampling system and maintain a constant pressure in the gas analyzer manifold. Some of the gas analyzers were very sensitive to changes in flow or pressure, so this pressure control was finely tuned and closely monitored during testing. There were also a series of block valves that allowed the chiller and gas analyzers to be isolated, while high pressure nitrogen can be used to back-flush the sampling system.

In order to prevent loss of NO₂ in the chiller/dryer unit, a NO₂-NO converter was installed immediately upstream of the chiller/dryer. This device uses a heated catalyst bed to convert NO₂ to NO. Since NO is not soluble in water, this converter minimized the potential of NO₂ coming in contact with condensate in the chiller/dryer unit. The conversion efficiency for this converter is in the 90 to 95 percent range, which is satisfactory by industry standards. Furthermore, the dryer unit uses a peristaltic pump to continuously remove condensate as it forms. By continuously removing the condensate, the probability of losing NO₂ due to gas-liquid contact with condensate is further reduced.

Figure 13: Process flow schematic for the gas analyzers

2.2.3.3 Gas Analyzers

Table 2 shows the species that are measured, and the type of individual gas analyzers that were used in the test facility. Each of the gas analyzers were calibrated daily during testing. Each analyzer was checked with a zero gas and a span gas. Although not shown in Figure 13, a Thermo ONIX (Model Prima delta-B) mass spectrometer was also used to measure certain key gas components, like CO₂ and O₂. This redundancy allows a secondary check of the O₂ and CO₂ readings during operation.

Table 2: Table of gas analyzers and manufacturer model number

Gas Component	Manufacturer	Model	Range	Manufacturer Specified Accuracy
CO ₂	Horiba	PIR-2000	0-20%	+/- 0.2 %
O ₂	M&C	PMA-12	0-21%	+/- 0.21 %
CO	API Gas Filtration	Model 300	0-500 ppmv	+/- 5 ppmv
NO _x	Horiba	CLA-510 SS	0 – 20 ppmv	+/- 0.2 ppmv
Hydrocarbons	VIG FID	Model 10	0 – 100 ppmv	+/- 1 ppmv

In addition to the daily zero and span calibration procedure, the NO_x analyzer was periodically checked using a 10-point calibration curve. An example of the NO_x data from this 10-point calibration is shown in Figure 14. Note that all the NO_x data recorded and presented henceforth in this report were corrected using the linear fit, based on this 10-point calibration. The 10-point calibration data is also useful for determining the analyzer uncertainty. In short, the standard error associated with the NO_x analyzer was within the manufacturer specifications listed in Table 2. However, for a higher degree of confidence, the uncertainty of the measurements changes, as shown in Table 3.

Figure 14: Results from 10-point calibration of NO_x analyzer

Table 3: Uncertainty of NO_x analyzer based on 10-point calibration data

Confidence Interval (based on normal distribution of error)	NO_x Analyzer Uncertainty
68% (1-sigma)	+/- 0.16 ppmv
95% (2-sigma)	+/- 0.32 ppmv
99.7% (3-sigma)	+/- 0.48 ppmv

2.3 Test Plan

2.3.1 Test Objectives

The goal of the program was to demonstrate NO_x emissions of less than 3 ppmv at 15 percent oxygen using an AVC concept.

2.3.2 Revision History

Table 4: Table of revisions made to the test plan

Date of Revision	Revision Level	Description of Revision
9/24/2003	0	Draft released for CEC review
11/07/2003	1	Final testplan submitted to CEC
7/22/2004	2	Revised as per final fabrication plan
9/27/2004	3	Revised as per input from Ramgen
12/15/2004	4	Revised based on test observations

2.3.3 Controlled Variables

The following variables were monitored, but not varied independently during the experiment:

- Pressure
- Inlet-air temperature

2.3.4 Independent Variables

The following is a list of process variables that were controlled independently:

- A. Main fuel flow rate (premixed), FIC-800
- B. Cavity fuel flow rate (premixed), FIC-769
- C. Main air flow rate, FIC-178

D. Cavity air flow rate (aft-wall air), FIC-027

E. Cooling air pressure drop, FIC-181

In addition to the process variables listed above, different geometric aspects of the combustor were also investigated:

F. Location of air injection from the aft-cavity wall

As expected, the independent variables listed above changed slightly as this project progressed. For example, based on test facility capabilities, it was decided to control the total fuel and air flows to the cavity, instead of controlling the flows to the fore- and aft-walls of the cavity independently. Furthermore, the baseline testing was conducted at a single pressure (i.e., 10 atmospheres), instead of conducting tests at several different pressure levels. And finally, budgetary constraints limited the number of different geometric aspects that were investigated.

Variables A through D can be re-defined into variables that are more meaningful for combustion applications. The actual experimental variables are defined as the following:

$$PHI_{main} = \frac{[(FIC\_800 + FIC\_769)/(FIC\_178 + FIC\_027)]}{Q_{stoich}} \quad (1)$$

$$PHI_{cavity} = \frac{(FIC\_769 / FIC\_027)}{Q_{stoich}} \quad (2)$$

$$U_{dumpplane} = C_o \frac{(TI\_968 + 460)}{(PI\_950 + 14.7)} FIC\_178 \quad (3)$$

$$X_{cavity} = \frac{FIC\_027}{(FIC\_178 + FIC\_027)} \quad (4)$$

2.3.5 Dependent Variables

The variables that were measured as outputs from this experiment include:

- NO_x emissions at the combustor exit
- CO emissions at the combustor exit
- O₂, CO₂, N₂, and unburned hydrocarbons at the combustor exit

- RMS pressure levels in the combustor

2.3.6 Description of Data Analysis

As described above, there were at least 6 different independent variables that could be investigated. The experimental testplan included a portion of time that was used to screen these variables and determine an applicable range of operating conditions for closer investigation. Pending the results of this screening exercise, the experimental test plan incorporated statistical design of experiment techniques to facilitate optimization of the combustor performance characteristics.

2.4 Acoustic Model

Although the following paragraphs describe the initial acoustic model that was developed during the conceptual design stage, this model was revised in the detailed design stage. The revised model included the downstream acoustics of the test rig and the actual location of an inlet choke point (see Appendix 2 for a summary of these results).

2.4.1 Background and Motivation

One of the significant technical challenges for low emission gas turbine combustors, particularly those that utilize lean premixed technology, is controlling combustion-driven instabilities. Combustion instabilities occur when fluctuations in the heat-release rate couple with the acoustics of the combustion system to produce oscillations in the system pressure. This feedback process is depicted in Figure 15. The pressure oscillations produced by this constructive feedback mechanism can reach amplitudes of 10 to 20 psi and occur 100 to 1,000 times per second. Clearly, pressure oscillations of this magnitude and frequency can lead to failure of key components in the gas turbine system. To further complicate this problem, this issue is often addressed so late in the design process that many options to reduce these instabilities become intractable.

This section describes a simple one-dimensional model of the baseline AVC configuration. This model was used to estimate the response of the combustor to variations in the heat-release rate. Although this model was developed for the baseline combustor geometry, further refinements were incorporated as the combustor design evolved. The baseline combustor geometry is described below, followed by an illustration of how this model can be used to assess different geometric configurations.

Figure 15: Diagram showing the feedback process between the acoustic process and the combustion processes

2.4.2 Baseline Combustor Geometry

The baseline combustor configuration is shown in Figure 16. The cross-sectional areas and other model inputs are shown in Table 5. Note that the flame is assumed to be disk shaped, located at the interface between sections L45 and L67. The flame model follows the expression described by Schuermans and Polifke (1999).

Figure 16: Sketch of the baseline combustor geometry

Table 5: Inputs for the acoustic model

Combustor Segment	Length (in)	Area (in ²)	Gas Temperature (R)	Mach Number	Ratio of Specific Heats
L01	12.22	5.995	1219	0.04	1.4
L23	3.12	2.211	1219	0.11	1.4
L45	0.52	5.995	1219	0.04	1.4
L67	0.53	5.995	3177	0.07	1.3
L89	0.82	3.218	3177	0.13	1.3
L10_11	27.71	5.995	3177	0.07	1.3

The results from the baseline combustor configuration are shown in Figure 17. Note that

the peak response occurs in the range of 645 to 650 Hertz. This frequency corresponds to an acoustic mode that is one-half wavelength from the inlet of the premixer to the flame. Therefore, the flame is located at a pressure anti-node, or velocity node (see Figure 18). Furthermore, the overall length of the combustor is nearly the same as 1.5 wavelengths. This model was used to assess potential changes to the combustor geometry to alter this response during testing.

Figure 17: Frequency response of baseline combustor configuration to a normalized fluctuation in heat release rate of unity

Figure 18: Acoustic mode shape for the 650 Hz frequency mode

2.4.3 Alternative Combustor Geometries

This one dimensional model can be used to assess potential changes to the combustor geometry, as described below. Two examples will be evaluated. First, if the length of the premixer is approximately 4 inches longer than the baseline design, the resonant

frequency decreases to approximately 560 Hertz, and the gain increases, as shown in Figure 19.

However, if the premixer length is decreased by about 4 inches, the resonant frequency increases to approximately 700 Hertz and the gain decreases. However, a second peak at around 1,000 Hertz began to develop and the model was extended to higher frequencies without any problem.

Figure 19: Model results for the baseline combustor configuration and two alternative options

2.4.4 Summary of Acoustic Model Results

In closing, the model described was developed to investigate potential combustion dynamics issues. This model was developed for the baseline combustor configuration, and was refined throughout the design process. This model was used to quickly assess the effects of different geometric arrangements and the results were used to refine the combustor design and achieve more dynamically stable operation.

2.5 CFD Analyses

2.5.1 Background and Motivation

Designing an AVC to achieve less than 3 ppmv NO_x is not a trivial exercise. In addition to the fact that our goal of less than 3 ppmv NO_x is a factor of three lower than previous results, there are no established design rules or guidelines. Therefore, CFD models of several combustor configurations were instrumental in enhancing our understanding of the flow patterns and the qualitative effects of different configurations. In choosing

parameters for CFD analyses, particular emphasis was given to geometric variables that are difficult and costly to investigate experimentally.

The CFD modeling effort started very early in the program. After establishing a baseline geometry, boundary conditions, and combustion model, the focus of the CFD effort shifted to investigating geometric variables that were identified as potentially important to the combustor performance. The independent variables investigated using CFD are shown in Table 6. Two levels, or cases, for each variable were identified for detailed study.

Table 6: List of independent variables investigated using CFD models.

Variable Designation	Variable
A	Location of aft-cavity wall injection
B	Presence of fore-cavity wall injection
C	Width of AVC module
D	Number of flameholders
E	Shape of flameholders
F	Width of flameholders
G	Inlet shape

In addition to the independent variables listed in Table 6, the following variables have been kept constant for each simulation (see Table 7).

Table 7: Constant operating variables for CFD simulations

Variable	Constant Level
Degree of premixing	100%
Main equivalence ratio	0.60
Cavity equivalence ratio	0.60
Bulk exit velocity from premixer	250 ft/s
Cavity loading	4%
Cavity injection hole size and pattern	Same as baseline

Recognizing the fact that we did not have the resources or the time to investigate each of the variables listed in Table 6, a fractional factorial design of experiments approach was used to identify the most important variables with the fewest number of simulations. These results are described below.

2.5.2 Detailed Discussion of CFD Results

Using the hypothesis that shorter flames are better for minimizing the NO_x production from non-thermal pathways, the temperature profiles predicted from the CFD code were analyzed. For the purposes of this preliminary evaluation, the temperature profiles at three axial locations were investigated (see Figure 20). At each interrogation plane, the average and standard deviation of the temperature were calculated. In order to make some quantitative measure of the reaction completeness at each plane, the ratio of the average temperature to the standard deviation of the temperature was used. This measure produces a similar ranking of the simulation results relative to a qualitative ranking produced from looking at flame lengths, temperature profiles, and other factors.

Figure 20: Interrogation planes for CFD analyses

The hydrodynamics were simulated using the FLUENT^{TM} CFD package. Only one-half of the combustor was modeled, since the geometry and the flow are one-half symmetrical. The k- ϵ realizable turbulence model was used in this study. The combustion process was modeled with the aid of the partially premixed combustion model, using methane and air as the reactants. The combustion products were assumed to consist of 17 different gases, in addition to oxygen and nitrogen. In the partially premixed model, transport equations were solved for the mean reaction progress variable, mixture fraction and the variance of the mixture fraction. Mean scalars, such as mass fraction, temperature, and density were then calculated from the probability density function of mixture fraction and the reaction progress variable. An underlying assumption of the partially premixed combustion model is that the reaction time-scales are short relative to the mixing time-scales, so local chemical equilibrium is assumed.

Using the FLUENT CFD package, a series of eight different geometries were investigated. Each of these simulations is briefly described below.

2.5.2.1 Simulation 1

The geometry for this simulation was very similar to Ramgen's initial combustor design that achieved less than 10 ppmv NO_x. This combustor geometry was used as the baseline design for comparison to other simulations. The calculated temperature contours are shown in Figure 21. The colder (unburned reactants) are shown as blue. Note that the flame angles were relatively narrow and a significant amount of reaction occurred near the exit of the computational domain.

Figure 22 shows the average and standard deviation of the temperature at various axial locations along the combustor. Note that Stations 1, 2, and 3 in Figure 21 are represented by the three data points closest to the origin of the graph in Figure 22.

Figure 21: Temperature contours and flame shape for Simulation 1

Figure 22: Temperature profile for Simulation 1 showing average and standard deviation of temperature at various planes along the combustor axis

As previously mentioned, the results from this simulation were compared to the other simulation results. Results from each simulation are presented below, and include data analysis and discussion.

2.5.2.2 Simulation 2

The flame shape for Simulation 2 is shown in Figure 23. By comparing Figure 21 and Figure 23, one can see that the flame shape is dramatically different. The flame angle seems to be much wider, and the reactants are consumed more quickly. It is believed that this flame would produce lower amounts of NO_x via non-thermal mechanisms. However, this hypothesis needs to be tested in the laboratory. Figure 24 shows the temperature profile for Simulation 2 relative to the average results obtained from Simulation 1. Note that Figure 24 suggests that most of the reactants are consumed by the time the flow reaches Station 3.

Figure 23: Flame shape and length for Simulation 2

Figure 24: Temperature profile for Simulation 2 compared to the average temperature results from Simulation 1

2.5.2.3 Simulation 3

Figure 25 suggests that the flame shape for Simulation 3 has a very narrow angle of divergence. However, the temperature statistics at several planes along the combustor axis show that this geometry is considerably better than the baseline (see Figure 26). Further investigation of the CFD results showed good mass transfer between the cavity and the main flow for this geometry, and the flame is fairly short

above and below the plane. Therefore, this configuration appears to be a reasonably good design, in spite of the results shown in Figure 25.

Figure 25: Flame shape and temperature contours for Simulation 3

Figure 26: Temperature profile for Simulation 3

2.5.2.4 Simulation 4

The flame shape for Simulation 4 is shown in Figure 27. The flame angle seems to be wider in this combustor geometry, relative to the baseline design (see Figure 21). However, the flame appears to be reasonably longer than those predicted for Simulations 2 and 3. The temperature profile shown in Figure 28 shows quantitatively that this geometry is better than the baseline design, but probably not

as good as Simulations 2 and 3. Note that a significant amount of reaction continues downstream of Station 3.

Figure 27: Flame length and shape for Simulation 4

Figure 28: Temperature profile for Simulation 4

2.5.2.5 Simulation 5

The geometry modeled in Simulation 5 produces some surprising results. In all of the previous simulations, the cavity region of the combustor produced a recirculation of hot gases that ignited the incoming fuel-air mixture. In other words,

the cavity was used to stabilize the flame. In this geometry, however, the flow in the cavity is disturbed to the point that the flame is no longer stabilized by the cavity. Instead, the flame is stabilized behind the AVC cavity module. This produces a long flame, which is not our objective. The results from this simulation are shown in Figure 29 and Figure 30.

Figure 29: Flame shape and length for Simulation 5

Figure 30: Temperature profile for Simulation 5

2.5.2.6 Simulation 6

The combustor geometry modeled in Simulation 6 shows a flame that is attached somewhat downstream of the cavity region of the combustor. Figure 31 shows the flame shape is relatively compact near the core of the flow, but the flame does not spread to the walls of the combustor. This means that the fuel-air mixture near the walls does not react within the computational domain. As a result, Figure 32 shows that the average temperature statistics for this simulation are lower than the baseline geometry.

Figure 31: Flame shape and length for Simulation 6

Figure 32: Temperature profile for Simulation 6

2.5.2.7 Simulation 7

Figure 33 shows the computed temperature contours for the combustor geometry modeled in Simulation 7. There still seems to be a cold (un-reacted) region near the walls of the combustor, but the average temperature values are somewhat higher than Simulation 6. In fact, relative to our baseline geometry (i.e., Simulation 1), this combustor geometry appears to be an improvement at Station 1 and Station 3 (see Figure 34).

Figure 33: Flame length and shape for Simulation 7

Figure 34: Temperature profile for Simulation 7

2.5.2.8 Simulation 8

The geometry modeled in Simulation 8 produces some interesting results. Based on Figure 35, the flame shape appears to have a fairly narrow angle of divergence, but the flame downstream of the cavity fills the entire volume of the combustor, which is similar to Simulation 3. Further investigation shows that the region above and below the plane shown in Figure 35 is very hot, and the transport of hot products into the main flow is very efficient in this combustor geometry. The temperature statistics shown in Figure 36 also indicate that this geometry is producing a much shorter reaction zone than the baseline design.

Figure 35: Flame length and shape for Simulation 8

Figure 36: Temperature profile for Simulation 8

2.5.3 Summary of CFD Results

As mentioned previously, the combustor geometries that have been modeled using the FLUENT CFD code were carefully selected. One of our goals for these simulations was to screen the effects of the different variables listed in Table 3. Therefore, the combustor configurations were methodically selected to allow the results to be analyzed using statistical design of experiments techniques. However, in order to make comparisons, a parameter that measures the “goodness” of the results must be developed.

As can be seen from the temperature profiles, the standard deviations in temperature can be very large, depending on the effectiveness of mixing processes that are normal to the mean flow. For example, standard deviations on the order of 500 °F were observed for some of the combustor configurations (i.e., Simulations 1, 4, 5, and 6). If the model produced better stream-wise mixing, the standard deviation is reduced. Therefore, the measure of “goodness” should increase as the standard deviation is reduced.

Furthermore, it was suggested that the reaction zone should be as compact as possible to reduce production of NO_x from non-thermal mechanisms. Therefore, a higher average temperature indicated more complete combustion, and the measure of “goodness” should increase as the average temperature increases.

In this study, the parameter that we have chosen for our measure of “goodness” is the ratio of the average temperature at a specific axial location to the standard deviation of temperature at that location.

For Station 1 (see Figure 37), the ratio of average to standard deviation has been normalized with respect to the maximum value observed. Simulations 2 and 3 appear to

have the highest ratio of average to standard deviation of temperature. Simulation 7 and Simulation 8 rank third and fourth, respectively. It should be noted that the results for Simulation 5 have not been included in Figure 37, since no reactions occurred at this axial location. Therefore, the parameter that was selected for comparison does not make sense for Simulation 5 at Station 1.

Figure 37: (Normalized) ratio of average to standard deviation of temperature at axial Station 1

For Station 3, which is further downstream (see Figure 38), Simulation 2 still ranks the highest, which may be due to the fact that the upstream mixing is so efficient. Simulation 8 actually ranks second, and Simulation 3 ranks third. Simulation 7 drops to less than 70 percent of the value observed in Simulation 2, but it still ranks fourth of the eight configurations studied.

Figure 38: (Normalized) ratio of average to standard deviation of temperature at Station 3

The purpose of the previous discussion was to simply show that the selected parameter for our measure of “goodness” produces a quantitative ranking that is somewhat consistent with our subjective evaluation. However, this parameter can also be used to rank the relative importance of the different combustor design variables listed in Table 6. However, due to the relatively low resolution of the seven-factor/eight-run fraction factorial approach, the main effects of the variables investigated in the following analysis can be confounded (or aliased) with two-factor interactions. Furthermore, not enough data exists to discuss statistical significance, or to analyze the variance. Therefore, it is important to remember that these results should only be used to screen the variables of interest.

The results of this analysis for Station 1 are shown in Figure 39. Note that the shape of the combustor inlet has the most significant effect on the performance of the combustor at this particular axial location. Furthermore, the shape of the flameholder design has the second most significant effect on the results. Using this information, the experimental test plan should investigate these two design variations. Note that the other factors also affect the CFD results, but these variables are on the order of three times smaller than the effect of the combustor inlet shape.

Figure 39: Relative effects of key design variables on CFD results at axial Station 1

For Station 3 (i.e., further downstream), the results of this analysis are a bit different than the results from Station 1, as shown in Figure 40. The shape of the combustor inlet still ranks as the variable with the most significant effect. However, this effect is followed closely by the flameholder size, and the presence of air injection from the upstream face of the cavity. It is interesting to note that the shape of the flameholder has a very insignificant effect on the average-to-variance ratio at Station 3, whereas this variable was important further upstream (i.e., at Station 1).

Figure 40: Relative effects of key design variables on CFD results at axial Station 3

In summary, several design options for a low-emission AVC have been investigated using the FLUENT CFD code. The CFD results provided insight into the relative importance of these design options, and will also be used to guide the experimental portion of this project. As a result of this effort, the combustor design incorporated several modular design features. This modular design allowed different aspects of the design to be easily investigated in a systematic fashion. Furthermore, a considerable amount of information has been obtained regarding the fluid dynamics and vortex motion in the cavity. This experience was very helpful for operating and interpreting the experimental data.

3.0 Project Outcomes and Results

3.1 Measurement Error Discussion

3.1.1 Flow Uncertainties

All of the flow measuring devices that supply fuel and air to the test rig have been calibrated and compared against industry flow standards. An example of this flow verification is shown in Figure 41. The uncertainty in each of the flow measurement devices is based on this type of flow proving data. The uncertainties in each of the critical flows are shown in Table 8.

For the data that will be presented in this report, the measured flow values will be corrected based on the results of the flow verification testing, in order to more accurately represent the actual flows. Furthermore, assuming the errors are normally distributed, the two-sigma uncertainties shown in Table 8 represent a 95 percent confidence interval. These confidence levels will be used in all of the error discussions in this report.

Figure 41: Example of fuel flow verification test data

Table 8: Measured flow uncertainties

Description of Flow	1-Sigma Uncertainty (scfh)	2-Sigma Uncertainty (scfh)	Full-Scale Range (scfh)	2-Sigma Uncertainty (% Full Scale)
Main Fuel	7.1	14.2	4,000	0.36%
Main Air	174.7	349.4	70,000	0.50%
Cavity Fuel	0.81	1.62	1,000	0.16%
Cavity Air	11.9	23.8	20,000	0.12%
Cooling Air	154.1	308.2	40,000	0.77%

3.1.2 Equivalence Ratio Uncertainties

In this project, the fuel and air flows are used to vary the fuel-air ratio to the vortex region and the main flow. The equivalence ratio, however, actually depends on the stoichiometric fuel-air ratio, which changes depending on the composition of the fuel. Although the composition of the natural gas was not a variable in this experiment, a fuel sample was taken daily and analyzed. An example of this natural gas analysis is shown in Table 9.

Table 9: Natural gas composition

Analysis	December 2, 2004	December 3, 2004	December 6, 2004	December 7, 2004	December 8, 2004
CH ₄	0.879728	0.89350	0.890282	0.905246	0.89930
C ₂ H ₆	0.099756	0.07590	0.080453	0.064257	0.06600
C ₃ H ₈	0.004613	0.01062	0.009891	0.012066	0.01268
C ₄	0.001269	0.00414	0.003818	0.004872	0.00548
C ₅	0.000494	0.00169	0.001578	0.001971	0.00232
C ₆	0.000232	0.00077	0.000718	0.000881	0.00115
C ₇ + C ₈	0.000165	0.00047	0.000430	0.000489	0.00070
N ₂	0.011891	0.01126	0.011444	0.009088	0.01096
Average MW	17.845	17.93709	17.957	17.838	17.99197
Stoichiometric Ratio	0.0975	0.09705	0.09696	0.09713	0.09674
HHV (MBtu/scf)	1.08356	1.08877	1.08969	1.08762	1.09217

The measured stoichiometric ratio shown in Table 9 was used in the calculation of equivalence ratio. It was assumed that the stoichiometric ratio did not change significantly during the day in which the data was collected.

The equivalence ratio for the cavity and main fuel-air mixtures was calculated using the measured fuel and air flow rates to each of these regions. The uncertainty in these calculated values is shown in Table 10. In actual practice, the cooling air could play an important role in the local fuel air mixture, but the effect of the local cooling air flow is difficult to quantify in this experimental setup. Therefore, the cooling air flow is neglected in calculation of the main equivalence ratio and cavity equivalence ratio.

However, for the total equivalence ratio, the cooling air flow is included in the calculation and the uncertainty listed in Table 10.

In general, the equivalence ratio uncertainties are quite acceptable, but it is interesting to note that the uncertainty of the cavity equivalence ratio is larger than the uncertainty in the main equivalence ratio. This is due to the fact that the flow measuring devices that feed the cavity with fuel and air were significantly larger than the actual flows used during testing. Since the flow requirements were on the low side of the total range for the flow measurement device, the relative error increased. The flow measurement devices for the main fuel and air flows were more adequately matched to the test conditions; hence the uncertainty in the equivalence ratio was significantly lower. The overall equivalence ratio calculation included three high temperature air flow measurements and two natural gas flow measurements, so the uncertainty should have increased slightly.

Table 10: Uncertainties in equivalence ratio calculated based of flows

Independent Variable	Uncertainty (95% Confidence)
Main equivalence ratio	± 0.001
Cavity equivalence ratio	± 0.011
Overall equivalence ratio	± 0.006

It should be noted that the overall equivalence ratio at the exit of the burner was calculated based on the measured flows, but it can also be calculated based on the exhaust measurements made in the sample probe. When using the exhaust measurements to calculate an equivalence ratio in this report, complete combustion was assumed. It is a good practice to check whether these two measurements agree.

Figure 42 and Figure 43 show the comparison of the overall equivalence ratio, based on the flow calculation and the measured oxygen and carbon-dioxide levels in the exhaust, respectively. Included on these plots are error bars on the calculated overall equivalence ratio, based on the 95 percent confidence level uncertainties in the flow measurements (see Table 10). Note that the O₂ and CO₂ readings from the mass-spectrometer were used in these plots, and the error bars for the O₂ and CO₂ measurements were plotted in Figure 42 and Figure 43.

For the majority of the data shown in Figure 42 and Figure 43, the two methods of calculating the overall equivalence ratio agree within the expected error bands. However, some data fall outside the expected error bands, and one should be cautious about making conclusions about data when this occurs.

It should be noted that the data analysis shown in Figure 42 and Figure 43 have been conducted for the test conditions listed in Table 12. Figure 44 shows the CO and unburned hydrocarbon emissions as a function of the main equivalence ratio. Note that when the CO and unburned hydrocarbon levels start increasing, the exhaust

measurements of the overall equivalence ratio become biased toward a higher equivalence ratio, since some of the carbon and oxygen are lost to CO and complete combustion is no longer a valid assumption. This could explain the discrepancies observed at the very lean operating conditions in Figure 42 and Figure 43, but it does not explain some of the points that fall outside the error bands at higher equivalence ratio conditions. More analysis is required to understand the points that fall outside the expected error bands.

Figure 42: Comparison of overall equivalence ratio based on flows and O₂ measurements in the exhaust

Figure 43: Comparison of overall equivalence ratio based on flows and CO₂ measurements in the exhaust

3.1.3 Gas Sampling Uncertainties

For each of the gas analyzers used in this effort, the instrument accuracy according to the manufacturer is listed in Table 2. However, these values represent the best possible accuracy, since this involves the accuracy of the instrument only. In actual practice, there are several other factors that can affect the measurement error, including process variations, flow variations, and others. In an attempt to estimate the error in the gas analysis measurements, a specific operating point was replicated several times. These data have been collected as part of a statistically designed experiment, which will be discussed in the next section, but it is important to note that the replications were collected in a randomized order to achieve a better estimate of the error.

Table 11 shows the error associated with each of the gas analyzers. Note that the CO and NO_x error has been expressed as the error in the measurement that is corrected to 15 percent oxygen basis, which is the current industry standard for reporting these emission levels. The NO_x measurements are within ± 1.2 ppmv, however the CO measurements are higher than expected.

Table 11: Experimental errors associated with gas analyzers

Analyzer Description	Mean Square Error	Uncertainty (95% Confidence)
NO _x @ 15% O ₂	0.35 ppmv	± 1.2 ppmv
CO @ 15% O ₂	188.0 ppmv	± 28.3 ppmv
O ₂ Analyzer	0.0042 %	± 0.13 %
THC Analyzer	2.65 ppmv	± 3.4 ppmv
CO ₂ Analyzer	0.0016 %	± 0.08%
O ₂ – Mass Spectrometer	0.0001 %	± 0.02%
CO ₂ – Mass Spectrometer	0.0001 %	± 0.02%

An underlying assumption is that the uncertainty is representative of the remainder of the experimental domain. It should be noted that the operating point selected for replication in this study was near a boundary, in which the CO levels began to increase sharply. A set of data that is similar, but not identical to, the conditions that were used to estimate the error is shown in Figure 44. The differences between the data shown in this figure and the conditions used for estimating the error are listed in Table 12. The CO data in the region of the operating point is very sensitive, and slight variations in flow, temperature, or other factors can lead to significant changes in the CO measurements. Although these error estimates for the CO may not apply in other regions of the experimental domain, the data suggests that accurate CO measurements are difficult to obtain near the lean blow-off limit. The other gas species listed in Table 11 were well-behaved near this operating point, so the error estimates for all of the other gas analyzers are more representative for the entire experimental domain. Even the unburned hydrocarbon species were fairly well-behaved in the vicinity of this operating point (see Figure 44).

Figure 44: Replication of CO and UHC data near the operating point to estimate error

Table 12: Comparison between operating conditions used to estimate error and data shown in Figure 44

Experimental Variables	Data Shown In Figure 44	Data Used for Error Analysis
Main equivalence ratio	Varies as shown	0.625
Cavity equivalence ratio	1.35	1.23
Reference velocity	235 ft/s	235 ft/s
Cavity loading	6%	5.5%
Cooling air pressure drop	4%	3.5%

3.1.4 Analyzer Response Times

Another important aspect to consider in the data analysis is the amount of time required for the gas analyzers to respond to a change in the combustion process. Although it is possible to estimate the response time of the sampling system alone, the following paragraphs will describe an approach to assess the response time of the entire process from a change in the fuel-air ratio to a corresponding change in analyzer response.

Again, the data shown in Figure 44 has been used to assess the response time of the gas analyzers. This data has been collected by systematically varying the fuel over the same range of main equivalence ratio. By collecting data in opposite directions (i.e., decreasing fuel flow versus increasing fuel flow), the emissions measurements as a function of main equivalence ratio based can be compared to obtain an estimate of the response time for the entire process. Figure 45 shows the time series data for the CO

analyzer. If the raw data is used without considering the response time for the CO analyzer, the analyzer output is skewed relative to the main equivalence ratio. When the equivalence ratio is decreasing, the analyzer output is skewed toward lower CO measurements. When the equivalence ratio is increasing, the raw CO readings are skewed toward higher readings (see Figure 46).

Fortunately, the response times for each of the analyzers can be assessed by shifting both of the curves shown in Figure 46 by an equal amount until the agreement is acceptable. Note that the resolution of this approach is only as good as the sampling interval which is ten seconds for this data. However, since this data is collected over a fairly long time period (i.e. 15 minutes) and the changes in main equivalence ratio are relatively small, this approach seems reasonable. Figure 47 shows the results of shifting the CO and NO_x data collected from these two tests. This approach is used to estimate the response time for each of the analyzers, and the results are shown in Table 13.

Figure 45: Example of time-series data used to assess analyzer response times

Figure 46: Example of raw CO readings that are not corrected for response time

Figure 47: CO and NOx readings shifted to account for analyzer and sampling system response times

In summary, if neglected, the response times for the analyzers can significantly skew the measurements, particularly when time series data similar to that shown in this section are used. For the remainder of this report, the data will be adjusted to account for the time inherent time delay in the sampling system. The response times listed in Table 13 will be used throughout this report.

Table 13: Analyzer response times

Response Time	
Analyzer	(seconds)
NO _x analyzer	80
CO analyzer	140
O ₂ analyzer	30
CO ₂ analyzer	30
Hydrocarbon analyzer	30
O ₂ – mass spectrometer	40
CO ₂ – mass spectrometer	40

3.1.5 Summary of Error Analysis

The following is a summary of the error analysis that has been performed for this effort.

- The uncertainties in the flow measurements are within 1 percent full-scale of a secondary flow standard for all of the flow measurements (see Table 8).
- A specific operating point that was repeated several times was used to assess the error of the emissions measurements. The expected experimental error in the emissions measurements are shown in Table 11 with one exception. The experimental error for the CO is not representative of the entire experimental domain, since the operating point selected for error analysis is so close to a flame extinction boundary. However, it is interesting to note the amount of uncertainty that can occur in a region in which the CO is so sensitive to operating conditions.
- The expected uncertainty in the various equivalence ratio calculations is also discussed. Figure 42 and Figure 43 show the comparison between the equivalence ratio calculation, based on the measured flows, and the same value calculated based on the measured emissions at the sample probe. With a few exceptions, these two calculations agree within the expected error.
- The response time for each gas analyzer has been determined and is listed in Table 13. The data presented throughout the remainder of this report will be adjusted to account for these time delays. If these time delays are not included, the measurements could be skewed from the actual value.

3.2 Factorial Designed Experiment

3.2.1 Approach and Experimental Design

After extensively exploring the range over which the combustor would operate, a statistically designed experiment was conducted to investigate the effects of three primary independent variables. Due to constraints in the test schedule and an upper limitation on the amount of fuel that could be supplied to the cavity fuel-air premixer, only three of the four variables listed in Table 14 could be investigated. A two-level factorial with a replicated centerpoint design has been pursued. The main effects and two-way interactions of the first three variables listed in Table 14 have been investigated using this approach. It should be noted that the cooling air pressure drop had been maintained at 3.5 percent for the data collected in this experiment. This parameter has a significant effect on the combustor performance, and will be discussed in the following sections.

Table 14: List of independent variables investigated using a factorial design

Independent Variable	Low End of Experimental Domain	Mid-Point	High End of Experimental Domain
Main equivalence ratio	0.550	0.625	0.700
Cavity equivalence ratio	1.10	1.23	1.35
Reference velocity	205 ft/s	235 ft/s	265 ft/s
Cavity loading	5.5 %	5.5 %	5.5 %

3.2.2 Results from Statistically Designed Experiment

Figure 48 shows the NO_x levels that were observed for each operating point in the two-level factorial design. Note that each sub-graph shown in Figure 48 is actually a plot of measured NO_x emissions corrected to 15 percent oxygen as a function of the main equivalence ratio. The main variables are coded to cover a range from -1 to +1 over the experimental domain listed in Table 14. The NO_x levels clearly increase as the main equivalence ratio increases for each of the sub-plots. The rows of sub-plots represent a constant level of cavity equivalence ratio, and the columns of sub-plots represent a constant level of reference velocity. The centerpoint that was used for the error analysis discussion in the previous section is shown in the center sub-plot in Figure 48. Three data points have been taken at this centerpoint, and each of these points are shown in Figure 48.

A regression analysis was conducted to assess the relative importance of the three main effects (i.e., reference velocity, cavity equivalence ratio, and main equivalence ratio), and the two-way interactions between these variables on the observed NO_x levels. The results of this regression have indicated that the main equivalence ratio and the cavity equivalence ratio had statistically significant effects on the observed NO_x levels. However, the reference velocity and the two-way interaction effects are not statistically significant.

A similar analysis has been conducted for the CO measurements. Even though the flame extinguished at one operating condition before the CO analyzer had a chance to stabilize, the results suggests that the main equivalence ratio is the only factor that had a significant effect on CO emissions. This information is useful in later discussions, since it is possible to vary the cavity equivalence ratio to reduce the NO_x emissions without adversely affecting the CO emissions.

Figure 48: Categorized scatterplot of NO_x levels for two-level factorial design

3.2.3 Conclusions from Factorial Designed Experiment

This factorial designed experiment was critical in assessing errors associated with measurements. Furthermore, this experiment has been useful for identifying variables that are most important for achieving the goals of this program. For the variables that have been investigated, the conclusions are as follows.

- The NO_x emissions are significantly affected by the main equivalence ratio and the cavity equivalence ratio. For comparison, the relative magnitudes of each main effect are shown in Figure 49. Note that the dashed line represents a boundary for 95% statistical significance.
- The CO emissions are significantly affected by only the main equivalence ratio. All other main effects are statistically insignificant.
- No two-way interactions between the primary variables have been observed.

Dashed line represents boundary for 95% statistical significance

Figure 49: Comparison of main effects on NO_x for variables investigated in factorial designed experiment

3.3 Effect of Cooling Air Pressure Drop on Pollutant Emissions

Due to the fact that the liner temperatures were well below design limits, it was decided to reduce the cooling air pressure drop. The cooling air pressure drop, and subsequently the cooling air flow rate, had a dramatic effect on the CO emissions and the overall combustor performance (see Figure 50). For the data shown in this section, the cavity loading is maintained at 6 percent of the combined cavity and main air flows. The reference velocity is maintained at 235 feet per second, and the cavity equivalence ratio is maintained at a value of 1.20. It should be noted that the cavity equivalence ratio values throughout this report do not include the local dilution effect of the cooling air in the cavity region. Due to the effect of the cooling air, the actual equivalence ratio in the cavity is expected to be significantly lower than the values listed throughout this report.

As a result of decreasing the cooling air pressure drop from 3.5 percent to 2 percent, Figure 50 shows that the CO emissions were reduced significantly, and the point at which the CO began to increase suddenly shifted to a lower main equivalence ratio. This dramatically increased the operating range for the combustor and was critical for achieving the goals of this program.

Figure 51 shows that the NO_x emissions increase for the same main equivalence ratio as the cooling air pressure drop is decreased. However, due to the extended operating range, the same level of NO_x emissions can be achieved at a lower main equivalence ratio.

In order to better understand the effect of the cooling air on the emissions performance, Figure 52 shows the CO data plotted as a function of the total equivalence ratio. The total equivalence ratio is calculated based on all of the fuel and air flows, including the cooling air. Since the CO does not change significantly when plotted as a function of the total equivalence ratio, the cooling air seems to have a dilution effect on the reactants. In the end, the total equivalence ratio of the combustor is not changing much due to these changes in cooling air pressure drop.

It is interesting to note that the CO-NO_x curve does not shift significantly to the left as a result of reducing the cooling air flow (see Figure 53). In order to achieve less than 3 ppmv NO_x, the curve shown in Figure 53 must shift further to the left.

Figure 50: Effect of cooling air pressure drop on CO emissions

Figure 51: Effect of cooling air pressure drop on NOx emissions

Figure 52: Cooling air effect on CO shown as total equivalence ratio

Figure 53: CO vs NOx plot as a function of cooling air pressure drop

In order to investigate the repeatability of the experimental setup, the same conditions were repeated on a different day of operation. The repeatability of these results can be seen in Figure 54. Most of the data points fall within the calculated error bands for NO_x and CO.

Figure 54: Replication of data at the lower cooling air flow

3.4 Effect of Cavity Equivalence Ratio

In order to reduce the NO_x emissions and shift the CO - NO_x curve further to the left, recall that the cavity equivalence ratio affects the NO_x , but not the CO emissions. Therefore, using the two percent cooling air data shown in Figure 53 as a baseline, the cavity equivalence ratio is decreased. Again, it should be noted that the cavity equivalence ratio values represent the fuel-air ratio of the jets entering the cavity region. These values do not include the local effects of the cooling air, therefore, the actual equivalence ratio in the cavity is expected to be significantly lower than the values listed in this section.

Figure 55 shows the NO_x levels for the baseline cavity equivalence ratio of 1.20 and the NO_x levels for lower cavity equivalence ratio conditions. The NO_x reduction is on the order of 3 ppmv, or in excess of 50 percent of the baseline cavity equivalence ratio condition. Furthermore, at a main equivalence ratio of about 0.54, the NO_x emissions are below the 3 ppmv goal of the program.

Figure 55: Effect of cavity equivalence ratio on NO_x emissions

Further reductions in the cavity equivalence ratio resulted in flame extinction at these operating conditions, so for the present combustor configuration, it appears that these emission levels were optimized for this operating condition and combustor geometry. However, with some modifications to the combustor configuration, it may be possible to further reduce the cavity equivalence ratio in order to achieve even better performance.

The CO emissions for these operating conditions are shown in Figure 56. Note that the CO emissions began to increase sharply at a main equivalence ratio of about 0.56, and decreasing the cavity equivalence ratio seems to make the CO emissions more sensitive. The CO-NO_x data is shown in Figure 57. This figure shows that the curve has indeed been shifted to lower NO_x levels.

Figure 56: Effect of cavity equivalence ratio on CO emissions

Figure 57: Effect of cavity equivalence ratio on CO-NO_x curve

3.5 Effect of Premixer Reference Velocity

Although the previous results suggested that the reference velocity had no effect on the observed NO_x and CO emissions, some experience from the exploratory screening tests suggested that lower cavity equivalence ratio conditions could be achieved at a higher bulk reference velocity through the premixer. If a lower cavity equivalence ratio can be

achieved at the higher reference velocity, then it may be possible to shift the CO-NO_x curve shown in Figure 57 further to the left and achieve even better performance.

The effect of increasing the premixer reference velocity while maintaining a constant cavity equivalence ratio of 1.0 is shown in Figure 58. Note that the variation in NO_x levels at these two reference velocities was within the experimental uncertainty, but the point at which the CO began to increase suddenly (i.e., the “knee” of the curve) shifted to a lower main equivalence ratio. It should be noted that this effect is similar to the cooling air effect discussed previously.

In fact, when the same data from Figure 58 were plotted against the total equivalence ratio which includes the cooling air, the data collapsed into a single curve (see Figure 59). This behavior suggests that increasing the reference velocity through the premixer reduced the relative effect of the cooling air. In short, the total equivalence ratio shifted to higher values by increasing the reference velocity which should produce higher flame temperatures that are favorable for CO oxidation. It is also interesting to note that the CO-NO_x curve (see Figure 60) does not change significantly as a result of this increase in reference velocity. This could indicate that the flame extinction process is starting at roughly the same flame temperature.

Figure 58: Effect of reference velocity on NO_x and CO Emissions

Figure 59: Effect of reference velocity on CO and NOx emissions on a total equivalence ratio basis

Figure 60: Effect of reference velocity on the CO-NO_x curve

As previously mentioned, the effect of the reference velocity may not have a significant effect on the total NO_x emissions, but the higher reference velocity may allow the cavity equivalence ratio to be decreased to lower values without experiencing the rapid increase in CO emissions. In fact, it was possible to reduce the cavity equivalence ratio from 1.00 to 0.90 at the higher reference velocity conditions (see Figure 61). As a result,

the NO_x emissions are reduced by an order of 1 ppmv. Although one could argue the statistical significance of this change: the fact that a lower cavity equivalence ratio could be achieved without flame extinction at the higher reference velocity is certain.

By increasing the reference velocity and decreasing the lower range that could be obtained on the cavity equivalence ratio, the CO-NO_x curve shifted slightly to the left (see Figure 62). The NO_x levels are less than 3 ppmv at 15 percent oxygen basis. Therefore, it is believed that the data presented in Figure 62 demonstrates that the AVC concept is capable of achieving less than 3 ppmv NO_x at realistic gas turbine conditions.

Figure 61: Effect of cavity equivalence ratio at higher reference velocity conditions

Figure 62: Effect of cavity equivalence ratio on CO-NO_x curve at higher reference velocity conditions

In summary, the results shown in Figure 61 and Figure 62 represent the best performance observed for this combustor configuration. These results will be discussed in greater detail in the following sections.

3.6 Combustion Efficiency

There are a couple of methods available to calculate combustion efficiency (i.e., heat basis, or carbon conversion basis). For this report, the combustion efficiency is calculated based on a heat basis using the following equation (SAE Report 1982):

$$\eta = 100 \left[1 - 10109 \frac{EI_{CO}}{LHV} - \frac{EI_{UHC}}{1000} \right] \quad (5)$$

The terms EI_{CO} and EI_{UHC} represent the CO and unburned hydrocarbon emissions expressed in terms of grams-per-kilogram of fuel. Furthermore, the lower heating value of the fuel is expressed as LHV in the previous expression. The second term in brackets represents the energy lost in forming CO instead of CO₂, and the last term in brackets represents the energy lost by not burning all of the fuel.

For the data shown in Figure 61 and Figure 62, the combustion efficiency is calculated and plotted as a function of the NO_x emissions (see Figure 63). Note that the combustion efficiency is in excess of 99 percent for most of the data presented. In fact, for combustion efficiencies of about 99.5 percent, the NO_x levels are in the 2.5 ppmv

range. However, for the steep portion of the curve shown in Figure 63, the uncertainties in combustion efficiency can be significant due to the sensitivity of the CO and unburned hydrocarbon emissions in this region.

Figure 63 Combustion efficiency for the lowest cavity equivalence ratio achieved

3.7 Dynamic Stability (RMS Pressure Levels)

One of the significant technical challenges for low emission gas turbine combustors, particularly those that utilize lean premixed technology, is controlling combustion driven oscillations. Combustion instabilities occur when fluctuations in the heat-release rate couple with the acoustics of the combustion system to produce pressure oscillations. This constructive feedback mechanism can produce pressure waves that reach amplitudes of 10-20 psi and occur 100 to 1,000 times per second. Clearly, pressure oscillations of this magnitude and frequency can lead to failure of key components in the gas turbine system.

Since the AVC concept investigated in this effort was a lean premixed design, some combustion driven oscillations were expected. Two high speed pressure transducers (Kistler Model #206) were located in the premixer region of the combustor to detect pressure oscillations. Due to temperature limitations, these sensors were located outside the pressure vessel on an infinite coil, which is the typical approach used in many gas turbine test rigs. The signal from these pressure transducers could be collected on digital tapes at sampling rates of 24,000 samples per second. The output from one of these transducers was also analyzed using an RMS meter, and the output from this RMS meter was stored in data acquisition system.

The RMS pressure as a function of NOx emissions is shown in Figure 64. Note that these data were collected at the same time as the data presented in Figure 62 and Figure 63.

Furthermore, these RMS pressure levels are representative of the levels observed for all of the testing. In short, no significant pressure oscillations were observed during the testing of the AVC. Although not an original project goal, the dynamic stability of this design is arguably the most significant achievement of this project.

Figure 64: RMS pressure levels observed

3.8 Combustor Pressure Drop

One of the advantages of the AVC approach over a swirl-stabilized combustor is the fact that no swirl vanes are required. The process of generating swirl in the main air flow requires some energy that ultimately becomes an efficiency penalty for conventional gas turbines. With a conventional gas turbine, a typical combustor pressure drop is on the order of 3 percent at a nominal velocity of about 150 to 200 feet per second. Based on data collected during these tests, the AVC concept is capable of achieving significantly lower pressure drop operation at even higher velocities than a conventional swirl-stabilized combustor (see Figure 65). The lower pressure drop of the AVC is an attractive feature, not only for gas turbine applications, but also for other applications, such as industrial burners.

Figure 65 Combustor pressure drop measurements at two reference velocity conditions

In summary, although efficiency gains in gas turbine applications typically focus on increasing turbine inlet temperature and all of the materials and cooling issues associated with it, the approach of reducing the pressure drop is often over-looked. Admittedly, the efficiency gain due to the pressure effect is not as dramatic as the temperature effect, but the AVC approach may offer an alternative to achieve modest gains in efficiency without addressing problems that are limited by high temperature material development.

3.9 Technology Transfer, Publicity and Related Activities

Ramgen has focused the dissemination of the AVC results within the industrial gas turbine community since the funding from previous and present programs have been for low emissions ramjet and gas turbine combustor development. A greater understanding is required of potential applications in both the industrial boiler and process heating markets before the AVC technology can be successfully applied. It is Ramgen's position that the AVC technology will find applications in the industrial heating community and will make a valuable contribution towards greater flame stability and lower emissions.

3.9.1 Papers, Reports & Presentations

Several published and unpublished papers and reports have been compiled since the inception of the AVC development program in 2001. Test results from Phase 1 were published and presented at the 2003 ASME IGTI Turbo Expo in Atlanta, GA. The results have also been presented at several DOE technical conferences and seminars, including the annual Advanced Gas Turbine Systems Research meeting.

Ramgen and the DOE will pursue the publication of the current results in the ASME and AIAA technical journals, and in engineering magazines, such as *Gas Turbine World*, *Turbomachinery*, *Chemical and Mechanical Engineering*, *Distributed Energy*, and *Diesel & Gas Turbine Worldwide*.

Appropriate news releases from the DOE and Ramgen will occur at the end of the program.

3.9.2 Technical Organizations

The potential capabilities of the technology and the recent test results will be disseminated into the following U.S. organizations for additional exposure and possible collaboration.

- ASME International Gas Turbine Institute
- U.S. DOE EERE Office of Industrial Technologies
- American Flame Research Committee
- American Boiler Manufacturers Association
- Council of Industrial Boiler Owners
- Industrial Heating Equipment Association
- Energy Solutions Center
- Gas Technology Institute

3.9.3 Website Activities

Ramgen plans to devote a section of the company website to the AVC tests and results. They will provide a link to non-proprietary information in the form of Microsoft PowerPoint slides and video. Ramgen will work with the DOE NETL and the CEC to provide the necessary information for updating their government websites.

3.9.4 Combustor Manufacturers

The AVC technology has received limited exposure from gas turbine and industrial burner manufacturers, suppliers, and users. The completion of recent tests will lead to more interaction and potential joint-venture activities in the future. Ramgen has contacted several industrial gas turbine manufacturers to inform them of the AVC technology and its potential. The gas turbine manufacturers have recently been exposed to the technology through technical papers and presentations.

Ramgen and DOE NETL began disseminating the AVC technology to the industrial burner community by visiting the John Zink Company in Tulsa, OK, in February, 2004. The technical staff at the John Zink Co. was very interested in the AVC technology and its potential, and is supportive of testing an AVC prototype system in their development facility. Ramgen plans to meet with John Zink Co. and report the recent results.

In August, 2004, Ramgen met with the Gas Technology Institute (GTI) in Chicago to introduce the AVC technology and pursue potential collaboration. GTI has developed and/or co-developed industrial burners for over 40 years. A teaming agreement is in review between the Ramgen and GTI for future opportunities. It is anticipated that the GTI will assist Ramgen with identifying unique industrial applications for the AVC technology.

Ramgen has identified the following companies as potential organizations for collaboration and joint product development. These are not complete lists, and the majority of the companies are U.S.-based.

Industrial Combustion Manufacturers	Stationary Gas Turbine Manufactures
Pyronics	Alstom
Ipsen International	General Electric
Maxon Corp.	Solar Turbines**
Callidus	Capstone**
Coen Co.	Kawasaki
National Combustion Equip.	Rolls-Royce
Industrial Heating	Siemens
North American Manufacturer Co.	UTC PWPS
John Zink	Vericor
Alzeta**	Hitachi
	Ingersoll-Rand

(** -- California based)

3.9.5 California Activities

Ramgen intends to emphasize the introduction the AVC technology to private and government organizations in California. The organizations that have been identified are the California Energy Commission, Electric Power Research Institute, Southern California Gas, combustion hardware manufactures and users, universities, government laboratories, and electric power producers.

3.10 Production Readiness and Commercialization Strategy

Ramgen plans to develop different lines of combustion hardware for use in a wide variety of industrial power generation and process heating applications. Ramgen will continue to complete the development of the AVC technology that has tremendous promise for improved efficiency, lower emissions, greater flame stability, fuel flexibility, increased durability, and reduced manufacturing costs.

The market opportunities for the AVC technology are identified in Figure 66. This figure identifies application opportunities and distribution channels by dividing them into three categories; (1) gas turbines, (2) boilers, and (3) process heaters. The applications will depend on specific requirements, such as fuel type, load requirements,

emissions, and other unique features. The product will be distributed directly by Ramgen, turbine or burner manufacturers and/or overall system packagers.

The applications include the existing and aftermarket opportunities, as well as new applications that become available as a result of the unique features of the AVC technology. Each category requires market and technical evaluation in order to determine the viability of introducing the AVC technology. Ramgen is evaluating the potential for product insertion into a wide variety of industrial applications, including gas turbine power generation and mechanical drive, manufacturing processing, chemical process heating, steam boiler systems, and incineration. The immediate AVC application is for stationary gas turbines.

The Ramgen strategy is to offer its technology in the form of a component sale and/or license for incorporation into various industrial combustor designs. Ramgen will retain control over the proprietary AVC hardware manufacturing, supply and replacement.

Figure 66 AVC market opportunities

3.10.1 Industrial Process Heating & Boiler Opportunities

Industrial combustion technologies supply heat to nearly all manufacturing processes. The U.S. energy-intensive markets that rely on combustion processes, and use approximately 67 percent of the total energy used by all manufacturing industries are: steel, aluminum, glass, metal casting, forest products, refining, and chemicals. The major source of energy for these industrial processes comes from onsite combustion of petroleum products and natural gas. These often outdated and inefficient combustion systems consume over 15 percent of the total U.S. energy produced. The continued rise in fuel costs and further reduction in emission levels have caused the industry in partnership with the DOE and other government organizations to fund the development of high energy saving technologies that address improvements to existing combustion systems.

Industrial combustion systems are essential to the U.S. manufacturing sector. The total value of product shipments by the top U.S. combustion-based industries is estimated at \$890 billion. Every state—including California—with manufacturing facilities relies on combustion for everyday performance and virtually every industry requires some form of combustion for operation.

Combustion-based industries also represent an enormous source of employment. The top U.S. combustion-based industries employ more than 3.6 million people. These estimates must be expanded to include the world-wide manufacturing markets.

Installed combustion equipment can be broken down into two categories: boilers and process heating equipment, including furnaces, heaters, reactors, incinerators, dryers, ovens, and kilns.

The challenge is to balance competing priorities on energy efficiency, operating and capital costs, and compliance with emissions regulations. Large process heating furnaces and steam boilers are required to run for long periods of time with very few interruptions. Any disturbances and/or unscheduled shutdowns are extremely costly to a plant that can take several weeks to bring online.

The ability of the combustion system to respond to sudden fluctuations in fuel composition, large scale internal flow patterns, and plant load cycle is critical. The AVC technology has the potential to bring greater flame stability and lower emissions to industrial combustion systems with the promise of greater plant reliability and cost savings.

It is evident from the DOE FY2003 annual report that new industrial burner designs are constantly in development and field demonstration. Dr. Bob Gemmer, DOE's Energy Efficiency and Renewable Energy (EERE) Technology Manager – Industrial Combustion, has stated that presently there are 40 different DOE sponsored projects applied across the combustion-based industries. For example, the Forced Internal Recirculation Burner (FIR) combines several techniques to create a family of burners that promise to reduce NO_x and CO emissions without sacrificing efficiency or performance. Target NO_x emission for this project is 15 ppmv, representing a reduction of more than 90 percent from current uncontrolled levels. Co-funding is provided by GTI and GTI's Sustaining Membership Program. Another example is the Super Boiler project, which aims to create ultra-low emission, ultra-high efficiency steam-generation technologies that will be integrated in a package boiler. In addition to DOE support, the project is supported by the GTI Sustaining Membership Program and Southern California Gas Company.

It is also common to see combustion hardware manufacturers such as John Zink Co., Coen Co., Callidus, and Maxon Corp., as well as oil & gas companies, such as ExxonMobil, involved in these projects.

In summary, the development and introduction of new burner designs into the U.S. combustion-based industries is very active and will continue as the industries strive to meet the performance targets set in the roadmapping sessions for the years 2010 and 2020.

3.10.2 Stationary Gas Turbine Opportunities

Stationary gas turbines can be broadly classified into industrial and marine, aero-derivative, and frame designs, and in recent years, micro- and mini-turbines. All variations include combustors and fuel requirements of some description. Ramgen intends to offer its technology in the form of a component sale for incorporation into the various gas turbine designs.

The potential capability of the AVC technology to deliver 3 ppmv NO_x emissions can be an attractive option for the stationary gas turbine industry. The majority of simple cycle industrial & marine turbines are not able to achieve the 10 ppmv NO_x emissions level without the added expense of exhaust after-treatment, such selective catalytic reduction (SCR). The SCR technique requires the injection of highly toxic ammonia into the exhaust flow for NO_x abatement. This is a very costly and hazardous option when the storage and handling of ammonia in densely populated regions.

Although recuperated gas turbines such as the Solar Mercury 50, Ingersoll-Rand's PowerWorks, and Capstone's line of micro-turbines have demonstrated very low sub-10 ppmv NO_x emissions without after-treatment, Ramgen believes that the AVC technology has the potential to provide further flame stability and even lower NO_x emissions for the recuperated engines.

Another opportunity for the AVC technology is with fuel cell hybrid systems that offer the potential for exceptionally high efficiency and can be built in sizes small enough to make a difference in the development of distributed generation systems. These hybrid systems apply gas turbines as bottoming cycle for the hot molten carbonate and solid oxide fuel cell designs.

These hybrids can be configured as direct hybrids, where the flow paths of both fuel cell and turbine are cross coupled, or they can be configured as indirect hybrids where the exhaust heat from the fuel cell heats the working fluid of the turbine (air) through an external heat exchanger.

Fuel cell hybrids may offer unique applications for the AVC technology due to the fuel requirements and the system configurations, which typically require duct burners and/or unique turbine combustor requirements.

3.10.3 California Market Opportunities

The U.S. Department of Commerce (www.census.gov) reports the most recent California economic census (1997) that includes the kind of business, number of establishments, revenue, and number of employees. The tabulated information from the census is included in Appendix 5. The California industries that rely heavily on combustion-based processes are mining, electric power utilities, and a wide variety of manufacturers. The actual combustion-based devices or systems are not specifically called out in the census, but can be inferred for each kind of business.

The mining business relies heavily on steam for oil and gas extraction and removal of metal and nonmetallic ores. Reported are 910 establishments and 22,110 employees.

The reported utilities are electric generation, transmission and distribution, natural gas distribution, water, and sewage systems. There are 45 fossil fuel-based electric power generation establishments and 1,458 employees. Not included in these numbers are the small (<1 MW) distributed power generation units that are increasing in California.

There are 49,418 manufacturing establishments and 1,809,700 employees reported for a variety of businesses that depend heavily on steam generation and heating processes. Improved burner designs and heating systems will greatly enhance the efficiency of these manufacturing processes and reduce operating costs.

It is Ramgen's position that improved burner designs which deliver greater fuel flexibility, lower emissions, and higher combustion efficiency can lead to a stronger economic base for California.

3.10.4 AVC Development Program

The AVC technology is still in the pre-production level of development for an industrial burner or gas turbine application. Ramgen will continue to expand their knowledge base of the industrial burner industry and identify applications for the AVC technology. The immediate focus will be on the following two specific projects for improving stationary gas turbines.

3.10.4.1 Test Existing Hardware on Various Gaseous Fuels

On several occasions the Ramgen staff has discussed with NETL combustion scientists the requirement to burn hydrogen-enriched fuels in the gas turbines of the future. Gasified coal breaks down into a hydrogen-CO fuel that has very different flame characteristics from natural gas.

The flame speed of a hydrogen-air mixture is six times higher than a natural gas-air mixture. Conventional swirl-based systems need modifications in order to manage the potential for flashback and overheating of the hardware. The AVC approach shows tremendous promise to burn these hydrogen-based fuels at much higher throughput velocities, which provides compensation for the high flame speeds.

Another area to be investigated is burning a wider variety of medium and low-BTU sub-quality waste gases, including biomass, land-fill, and high percentage inert natural gases.

Ramgen will seek funding to continue testing the existing AVC hardware on a wide variety of fuels recommended by NETL and California Energy Commission. It is anticipated that relatively minor modifications will be required to prepare the existing hardware for these tests.

3.10.4.2 Design, Develop and Test AVC for Stationary Gas Turbine

The more extensive development path is to layout a program that will move the AVC technology towards a gas turbine retrofit application. Ramgen will pursue interest from

the Energy Commission and a cluster of California-based organizations that will assist with supporting the demonstration of the technology on stationary gas turbines in California.

Some of the organizations to be considered for involvement are: (1) gas and electric utilities, such as Southern California Gas and Southern California Edison; (2) other companies and organizations, such as Sempra Energy Solutions, EPRI, Lawrence Livermore and Sandia national laboratories; (3) combustion-based university research groups, such as University of California, Irvine; (4) and gas turbine manufacturers, such as Capstone Turbines in Chatsworth, and Solar Turbines in San Diego.

3.10.5 Project Objective

The overall technical objective of the project is to design, test and demonstrate an AVC-based combustion system on a stationary gas turbine in the state of California. In order to reduce program costs the gas turbine is expected to be below 5 MW. The emissions targets are 3 ppmv NO_x and 9 ppmv CO (15 percent O₂ corrected) without exhaust cleanup or internal catalysts. The project will be a multi-year program from the initial design, sub-component testing, and full engine demonstration.

3.10.6 Statement of Work

- Phase 1: Design AVC for specific industrial gas turbine
- Phase 2: Test AVC design in rig test at NETL or equivalent test facility in California
- Phase 3: Test AVC design at gas turbine OEM facility and prepare for engine test
- Phase 4: Conduct and support joint AVC gas turbine engine test in California

3.11 Summary of Project Outcomes and Results

The project outcomes and results can be summarized as follows:

- A detailed error analysis was completed for the data presented in this report. This information was used to identify 95% confidence error bands for the key results. As expected, these error bands were larger than the individual instrument uncertainties, because they represented the variations due to the entire process. In short, it is believed that the NO_x emissions presented in this report are accurate to within +/- 1.2 ppmv based on a 95% confidence interval.
- A factorial designed experiment was used to identify the key variables that affect the NO_x and CO emissions. The results showed that NO_x emissions could be reduced by reducing the main equivalence ratio or reducing the cavity equivalence ratio. The CO emissions could be reduced by increasing the main equivalence ratio. Note that the CO emissions were not significantly affected by the cavity equivalence ratio.

- The amount of cooling air used to cool the liner walls had a dramatic effect of the emission performance of the AVC combustor.
- The cavity equivalence ratio was varied to decrease the NO_x emissions without adversely affecting the CO emissions. At one condition, a 15 percent reduction in cavity equivalence ratio resulted in a 50 percent reduction in NO_x emissions (or about 3ppmv). The cavity equivalence ratio was used to shift the CO-NO_x curve to the left which allowed the goals of this program to be achieved.
- It should also be noted that the cavity equivalence ratio values presented throughout this report represent the fuel-air ratio entering the combustor as a high velocity jet. It is believed that these jets mix with cooling air in the cavity prior to being consumed, but the local amount of cooling air mixed with the fuel-air jet is not clear. Therefore, the values of cavity equivalence ratio entering the cavity is used as an experimental variable, but it should be noted that these fuel-air jets are diluted before reacting in the cavity.
- The reference velocity through the premixer was increased in order to achieve somewhat leaner cavity equivalence ratio conditions.
- NO_x emissions of less than 3 ppmv (at 15 percent oxygen) were achieved, and the combustion efficiency was in excess of 99 percent.
- No significant RMS pressure levels were observed during this testing. This is a significant finding, and will help generate commercial interest in the AVC concept.
- The combustor pressure drop for the AVC is significantly lower than conventional gas turbine combustors.
- A technology transfer plan has been presented in this report. In addition to this, a production readiness and commercialization strategy has also been presented.

4.0 Conclusions and Recommendations

- The project goal of demonstrating NO_x emissions of less than 3 ppmv (corrected to a 15 percent oxygen basis) was achieved in this effort.
  - NO_x emissions of less than 3 ppmv (at 15 percent oxygen) were achieved with combustion efficiencies in excess of 99 percent.
  - The lowest achievable NO_x emissions were around 2.0 ppmv (at 15% oxygen), but the CO levels were above the range of the CO analyzer (i.e., above 500 ppmv).
- No significant RMS pressure levels were observed during testing. This is a significant achievement for a lean premixed combustion approach.
- The pressure loss characteristics of the AVC concept are more favorable than conventional gas turbine combustion approaches. This could translate into efficiency improvements for the overall gas turbine cycle.

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6.0 Glossary

AVC	Advanced Vortex Combustor
CO	Carbon monoxide
CO ₂	Carbon dioxide
DOE	U.S. Department of Energy
DLN	Dry Low NO _x
HHV	High heating value
LHV	Low heating value
MW	megawatt
NETL	National Energy Technology Laboratory
NO _x	Oxides of nitrogen
OEM	Original Equipment Manufacturer
ppmv	Parts per million volume
RPS	Ramgen Power Systems
scfh	Standard cubic feet per hour
El _{co}	Carbon monoxide emissions based on the EI index (g CO per kg of fuel)
El _{UHC}	Unburned hydrocarbon emissions based on the EI index (g UHC per kg of fuel)
RMS	Root-Mean-Square

Appendix 1

Detailed Description of Tasks

Detailed Description of Tasks

1.0 Administrative Tasks

The following administrative tasks were completed as part of this project.

Task 1.1 – Kickoff Meeting

A Kickoff Meeting was held in Sacramento, California on August 5, 2003. Those in attendance at this meeting included the following.

- Doug Straub (NETL)
- Tom George (NETL)
- Chuck Alsup (NETL)
- Joe Williams (Ramgen Power Systems)
- Mike Batham (PIER)
- Avtar Bining (PIER)
- Judy Efhan (PIER)
- Frank Taniguchi (PIER)

Task 1.2 – Describe Synergistic Projects

Throughout the course of this project, the NETL shared results from other projects related to the AVC Project. These other projects addressed technical issues with achieving ultra-low emissions for gas turbine combustors, particularly directed at fuel variation effects. The status of these synergistic projects was updated in the monthly and quarterly progress reports.

Task 1.3 – Identify Required Permits

In order to accomplish the goals of this project, an existing test facility at the NETL was used. The permits required to complete testing in this facility were described in this task. A formal letter addressed to the CEC Contract Manager was submitted on August 20, 2003. This letter described the internal permits that would be required in order to complete the testing portion of this effort.

Task 1.4 – Obtain Required Permits

The internal operating permit for this activity was received on October 6, 2004. In order to receive this permit, a thorough review of the environmental, safety, and health issues had to be completed. This activity is described in Task 2.4 – Hazard Analysis and Safety Review.

2.0 Technical Tasks

Task 2.1 – Computational Fluid Dynamic (CFD) Analyses

The goal of this task was to use CFD codes in the following ways: 1) To help guide the conceptual combustor design, 2) To identify variables that should be investigated experimentally, 3) To improve the understanding of the experimental results. NETL lead this effort with analytical support from RPS.

The results from this task have been provided in the Quarterly Reports, Critical Project Review, and the Final Report.

Task 2.2 – Conceptual Design and Test Plan Development

The goal of this task was to identify promising approaches that could be used to achieve the goals of this program. Of the available alternatives, engineering models and analyses were used to select the best concepts. Both RPS and NETL were actively involved with this task.

An acoustic model of the conceptual geometry was developed and submitted to the Energy Commission on September 24, 2003. A draft test plan was also developed and submitted on September 24, 2003. In addition to this, the preliminary CFD model results were being generated, analyzed, and reported as a parallel effort to this task. On November 6, 2003, a conceptual design review meeting was held at NETL to resolve issues and define the final detailed design. On November 26, 2003, a final test plan was submitted to the Energy Commission.

Task 2.3 – Detailed Design

The goal of this task was to refine the conceptual combustor and test rig designs identified in the previous task to the point that fabrication drawings could be completed. Ramgen Power Systems lead the combustor design and NETL lead the design effort for the facility modifications.

A final design review meeting was held at NETL on February 10, 2004, but the main deliverables from this task were the Critical Project Review (CPR) Status Report and the CPR Meeting. The draft version of the AVC Design and Status Report was submitted to the Energy Commission on January 7, 2004. The Critical Project Review Meeting was held in Sacramento, California on February 27, 2004, and the Final AVC Design and Status Report was submitted to the Energy Commission on March 22, 2004.

Task 2.4 – Hazard Analysis and Safety Review

The goal of this task was to obtain internal DOE permits to modify the facility and test the Advanced Vortex Combustor concept. NETL lead this task.

The internal analysis of the ES&H aspects of this activity was completed on August 13, 2004. The results of the hazards analysis was documented on August 17, 2004, and all of the additional hazard mitigations were in place by September 10, 2004. The operating

permit was signed on October 6, 2004, and a copy of this permit was sent to the Energy Commission with the quarterly progress report for the quarter ending October 1, 2004.

Task 2.5 – Complete Fabrication and Process Drawings

The goal of this task was to develop the drawings and specifications necessary to fabricate the combustor and begin the modifications to the facility. Ramgen Power Systems lead the combustor drawing process and NETL lead the facility, process, and instrumentation drawing revisions.

Narrative summaries of this task were included in the progress reports, and a letter notifying the Energy Commission that this task had been completed was submitted on May 27, 2004.

Task 2.6 – Complete Fabrication of Test Hardware

The goal of this task was to complete the fabrication of the combustor components and the process/facility modifications. It was anticipated that most of the fabrication would occur at NETL with some articles being sent off-site, if capabilities such as laser drilling, or plasma-spraying were required.

The final combustor design included several processing capabilities not available at the NETL shop, so the budget was revised to increase the portion of the budget spent on fabrication, particularly for electric discharge machining, laser drilling, and electron beam welding capabilities. These processes were completed at external machine shops. Due to the business sensitivity of the combustor drawings, Ramgen was responsible for managing the fabrication effort at the external fabrication shops.

The fabrication effort at the NETL shop was completed on September 1, 2004, which was a couple of weeks behind our original schedule. The fabrication effort at the external machine shop was delayed due to late delivery of some high temperature alloys. The high temperature alloys that were supposed to be delivered in June were completely received on August 18, 2004. The essential components for testing were received at NETL on October 19, 2004. The spares and additional components for testing were received later.

Photos and narrative summaries of the results from this task were included in the quarterly, monthly, and even weekly progress reports to the Energy Commission.

Task 2.7 – Installation and Checkout

The goal of this task was to install the hardware and complete the necessary facility modifications. This task also involved calibration, checkout, and verification of all instrumentation. NETL was the lead on this task with onsite support from RPS.

Many of the calibrations and checkout activities were completed prior to receiving the combustor components. After receiving the components on October 19, 2004, the first shakedown tests were initiated on October 29, 2004. Narrative summaries of this

activity were included in the progress reports, and a letter notifying the Energy Commission that testing was about to begin was sent on November 4, 2004.

Task 2.8 – Advanced Vortex Combustor Test

The goal of this task was to demonstrate the performance of an Advanced Vortex Combustor at high pressure. NETL conducted the test on-site with NETL engineers and operators. RPS was present and actively participating in the investigation.

Following some early shakedown and low pressure testing, a Briefing Meeting was held in Morgantown, WV on November 18, 2004. Avtar Bining, the CEC contract manager, was on-site to attend this meeting. A briefing package of the project status and test results to date was presented at this meeting. The first tests at high pressure (i.e., 10 atm) were conducted on December 3, 2004 and the final tests on the Configuration 1 module were completed on December 8, 2004. Some additional tests were conducted on the Configuration 2 module in January 2005, and these results have been summarized and attached as an appendix to this report.

Task 2.9 – Data Evaluation and Post-Test Analysis

The goal of this task was to analyze the data and explain the test results. Both NETL and RPS were involved in this task.

A narrative summary of this task was included in the progress reports and a detailed discussion of the data evaluation was included in the final report that was submitted on January 6, 2005.

Task 2.10 – Technology Transfer Activities

The goal of this task is to make the knowledge gained, experimental results, and lesson learned available to key decision makers and the public. A draft technology transfer plan was included in the draft final report on January 6, 2005. The final technology transfer plan will be included as part of the final report.

Appendix 2

Acoustic Model of Final Combustor Design

Acoustic Model of Final Combustor Design

Analysis of Inlet Air Piping and Vessel Entrance

The Advanced Vortex Combustor (AVC) tests will require 0.65 – 1.75 pounds-per-second of air for the main air inlet. This air will be preheated to temperatures that range from 625 – 760°F using a non-vitiated air pre-heater. The operating pressure in the combustor will range from 150 – 270 psia. The air will be metered using a high-temperature flow control valve that is located approximately 30-feet away from the entrance to the pressure vessel. The main-air inlet to the vessel will be fabricated using 3-inch Sch80 stainless steel piping.

Based on a decision made at the conceptual design review meeting, the combustor will not incorporate a choke plate. Therefore, an acoustic model of the system should include the air delivery system. Due to the large pressure drop across the flow control valve, this valve can be used to define an acoustically closed boundary condition. That is to say, the acoustic velocity perturbations at the valve are considered negligible.

Using the acoustic Transfer Matrix Method described by Munjal (1987)¹, the acoustic impedance of the inlet air piping can be calculated. A simplified network of the acoustic transfer matrices for the inlet air piping (i.e., T1, T2, and T3) is shown in Figure 1. If the acoustic impedance of the main-air piping is calculated (i.e., $p1/u1$), the analysis shows a large number of resonant frequencies (see Figure 2). This is expected due to the existence of the long length of piping and several area changes. If these acoustic modes couple with the combustor, dynamics issues may be a concern for a wide range of frequencies.

Figure 1: Equivalent Acoustic Network of Inlet Air Piping and Core-buster

Acoustic Isolation Using “Core-Buster”

One of the features of the Ramgen AVC design is a “core-buster” at the inlet to the vessel. This core-buster design redistributes the axial jet of air entering the vessel into a

The initial goal in this analysis is to investigate whether the core-buster can be designed in such a way that the acoustics of the inlet piping are eliminated. Therefore an acoustic model of the core-buster behavior is required. A model of the core-buster is based on the same approach described by Richards et. al., (2001)².

In summary, the analysis shows that decreasing the hole size of the jets in the core-buster can help reduce the acoustic coupling between the main-air piping and the combustor, but the coupling cannot be reduced to acceptable levels over the entire frequency range of interest. Figure 4 shows the ratio of the upstream pressure (p1) to the downstream pressure (p2) for two different holes sizes (or Mach Numbers). The top graph in Figure 4 shows the baseline design in which the Mach Number is roughly 0.10. The bottom graph shows the results for a Mach Number of approximately 0.45. Note that some of the peaks are only reduced by a factor of 0.8, which is not good enough. Therefore, an alternative approach needs to be considered to isolate the acoustics of the upstream piping.

Figure 4: Acoustic Pressure Transfer Function Across the Core-Buster. The original (top) design with a low Mach Number and an alternative design (bottom) with a higher Mach Number are shown.

Acoustic Isolation Using A Perforated Choke Plate

Another alternative to isolate the acoustics of the upstream piping is to install a perforated plate that will introduce another choke point in the flow. This inlet-choke plate will effectively move the well-defined acoustic boundary from the valve to a location that is much closer to the inlet of the vessel, or core-buster. The new network model that is represented by introducing this inlet-choke plate is shown in Figure 5. The dramatically shorter length of pipe between the choke plate and the core-buster virtually eliminates all of the resonant frequencies over the range interest (see Figure 6). Figure 7 shows the response of the pressure upstream of the core-buster to a perturbation in the downstream

pressure. It is interesting to note that the gradual increase in this ratio at higher frequencies can be significantly reduced by decreasing the l/d ratio of the holes. The case shown in Figure 7 is for an l/d ratio of about 0.6. The case in which the l/d ratio is reduced to approximately 0.15 is shown in Figure 8.

Figure 5: Equivalent Acoustic Network for Inlet Choke Plate

Figure 6: Acoustic Impedance For System That Includes A Choke Plate 2-inches Upstream of the Core-Buster

$$\frac{d_{\text{hole}}}{0.0254} = 0.43 \quad (\text{in})$$

Figure 7: Acoustic Pressure Transfer Function Across the Core-Buster With Choke-Plate (l/d ratio for the core-buster is roughly 0.60)

$$\frac{d_{\text{hole}}}{0.0254} = 0.43 \quad (\text{in})$$

Figure 8: Acoustic Pressure Transfer Function Across the Core-Buster With Choke-Plate (l/d ratio for the core-buster is roughly 0.15)

In summary, by including an acoustic choke plate near the main-air inlet to the vessel the number of acoustic modes that can couple with the combustor are significantly reduced. Therefore, it is recommended that a choke plate be installed into the main-air piping.

Analysis of Combustor Acoustics Coupled With Test Rig

Description of the Acoustic Model

The acoustic model described here is somewhat different than the one that has been developed previously. One major difference is the fact that the acoustics of the upstream main-air system and the downstream vessel components have been incorporated into this model. In addition to including the geometry of the new combustor design, several other

features have been considered, including the core-buster design and the fact that the premixer actually extends into the plenum some distance. There are, however, some features that the current model has neglected. For example, a dynamic flame model has not been incorporated into this model, to date. In addition to this, the effusion cooling holes have not been included. It is believed that these cooling holes will introduce acoustic losses to the system such that the actual acoustic gain will be somewhat lower than what is calculated in this model.

The network of transfer matrices is considerably more complex for the entire system, but the network can be simplified into an equivalent network as shown in Figure 9. For the purposes of the acoustic model, a generic velocity source has been placed in the combustion region to simulate perturbations that may be generated at the flame front. For the purposes of this investigation, this acoustic source is expected to have a flat frequency response with a value of unity. It should be noted that most dynamic flame models are treated as first-order systems in which the gain rolls off as the frequency increases. Therefore, the choice of a zero-order system for the source will produce conservative estimates of the actual gain, particularly at high frequencies.

Figure 9: Equivalent Network For The Combustor/Test Rig Acoustic Model

Application of the Acoustic Model

The purpose of developing the acoustic model is to investigate options to improve the acoustic features of the system. The thermo-acoustic model that has been described in a previous report indicated that the acoustic response is very sensitive to changes to the length between the vessel inlet and the flame front. The results from this model also suggest that the main-air inlet design can play a significant role in determining the acoustic modes of the system (see Figure 10).

Figure 10: Diagram of Main-Air Inlet Details

In an attempt to optimize the design of the main-air inlet, the acoustic model is used to calculate the pressure in the combustion zone as a function of frequency. Depending on the location of the core-buster, the location of the choke plate, and the length of the inlet transition piping, significantly different pressure responses are calculated in the combustion zone. Since the size for most of the components has been finalized, these three parameters should not influence the combustor design. These parameters are shown schematically in Figure 11, and will be investigated using a factorial design of experiments methodology. Hopefully, this investigation will identify the key variables to control in order to reduce the acoustic response of the system. In order to measure the response of the system to changes in the lengths shown in Figure 11, the area under the combustor pressure response versus frequency curve will be calculated. These results will be described in the following paragraphs.

Figure 11: Schematic of Main-Air Inlet and the Design Parameters That Will Be Varied

The area under the combustor pressure frequency response curve is tabulated in Table 1. It should be noted that this area reflects the range of 0-1000 Hertz. Figure 12 shows the frequency spectra for each of the cases. It should be noted that although the case with the lowest area under the curve may appear to be the best configuration, the frequency spectra shows that this case does have some low amplitude, low frequency modes showing up in the combustor pressure response. These low frequency components are due to the distance between the inlet choke plate and the entrance to the vessel (i.e., L3). When L3 is considerably shorter, these low frequency modes do not occur.

Table 1: Geometric Combinations Investigated Using The AVC Acoustic Model

L1	L2	L3	Area Under the Curve (xe07)	
	12	5	12	1.97
	12	5	0.1	0.86
	12	0.1	12	4.34
	12	0.1	0.1	4.49
	0.1	5	12	2.38
	0.1	5	0.1	0.89
	0.1	0.1	12	5.08
	0.1	0.1	0.1	5.71

Figure 12: Frequency Response of Acoustic Pressure In Combustor Region

When the results listed in Table 1 are analyzed, the variable that has the most significant effect on the area under the pressure curve is the variable $L2$ (reference Figure 11). Fortunately, one of the design options involves attaching the core-buster to the inlet-air piping. This approach would allow this distance to be adjusted without lowering the entire combustor. This flexibility to alter the acoustics may be an important variable to consider during the test phase, but the question at this time is whether the acoustic response can be minimized using this length.

In a separate analysis, $L1$ and $L3$ are fixed to some reasonable values, and $L2$ is varied over a range of conditions. For this analysis, $L1$ is 8-inches, and $L3$ is 2-inches. As $L2$ is varied from a range of about 2.5-inches to 7-inches, the area under the pressure curve decreases as shown in Figure 13. Intuitively, inserting the core-buster more than 5-inches into the plenum may cause some problems with flow distribution, and the area under the curve appears to be approaching a point of diminishing returns. Therefore, the recommended value for $L2$ should be around 5-inches.

Figure 13: Area Under The Pressure Frequency Response Curve As A Function of L2.

Recommendations

Based on the analyses described above, the following recommendations for the main-air inlet design are presented:

1. A choke plate should be installed in the main-air inlet piping to isolate the acoustics from the facility piping.
2. This choke plate should be positioned closer than 12-inches from the entrance to the pressure vessel, or core-buster. Base on this study, it is recommended that the choke plate be positioned approximately 2-inches from the entrance to the core-buster.
3. The design option that allows the height of the core-buster to be adjusted without removing the bottom flange on the vessel will allow some flexibility in altering the acoustics of the system, if it is necessary to do so. It is recommended that this approach be pursued in the final design.
4. If the design option described in the previous recommendation is pursued, the initial design should target an L3 value of about 8-inches. This is the distance between the bottom flange of the vessel and the point at which the core-buster is attached to the inlet air piping (see Figure 11). Although the acoustic response is not very sensitive to this dimension, this is the dimension that has been used in calculating the results shown in Figure 14.
5. The core-buster should be located approximately 5-inches above the bottom flange of the vessel. This recommendation assumes that there are no issues with flow conditioning for the premixer at this location. Again, this is the value used in calculating the “new base case” results shown in Figure 14.

Figure 14: Pressure Response for L1=8", L2=5", and L3=2"

¹ Munjal, M. L., *Acoustic of Ducts and Mufflers*, John Wiley and Sons, NY, 1987.

² Richards, G. A., Straub, D.L., and Robey, E.H., "Dynamic Response of a Premix Fuel Injector," ASME Paper 2001-GT-036, June 2001.

Appendix 3

Summary of January 2005 Test Results

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Summary of January 2005 Test Results

1.0 Goals and Objectives of Test

The goal of this test was to evaluate the emissions characteristics of a new cavity configuration, denoted as Configuration 2. Configuration 1 was tested in December 2004 and the results were described in the draft final report submitted to the California Energy Commission on January 6, 2005.

2.0 Experimental Setup

The only difference in this experimental setup was the AVC module and the liner and frame that supported this module in the combustor assembly.

3.0 Discussion of Results

3.1 Comparison of AVC Modules

The Configuration 2 vortex module and the Configuration 1 vortex module results were compared at two different operating conditions (see Table 1). The cooling air pressure drop across the liner was maintained at 2.0% for both cases. Furthermore, the combustor back-pressure (132.3 psig) and inlet-air temperature (625F) were not varied.

Table 1: Table of Experimental Conditions Used For Comparison

Operating Condition	Main Air Flow (scfh)	Main Equiv. Ratio	Cavity Equiv. Ratio	Cavity Loading (%)
Condition 1	54,020	Varied	0.95	6.0
Condition 2	54,020	Varied	0.90	6.0

3.2 Summary of Results For Condition 1

Figure 1 shows the NO_x emissions for both AVC module configurations. The NO_x emissions are somewhat higher for Configuration 2. However, the difference is within the ± 1.2 ppm uncertainty in these measurements. The CO emissions are shown in Figure 2. The CO emissions for Configuration 2 start to increase sharply at a higher main equivalence ratio and a higher NO_x level. This has the effect of shifting the CO-NO_x curve to the right as shown in Figure 3.

Figure 1: NOx Emissions For Both Configurations

Figure 2: CO Emissions For Both Configurations

Figure 3: CO-NOx Curve For Condition 1

3.3 Summary of Results For Condition 2

The NOx emissions data for Condition 2 are shown in Figure 4. These results suggest that the NOx emissions were nearly the same until the lean extinction limit for Configuration 2 was reached (i.e., main equivalence ratio of about 0.53). The Configuration 1 module lean extinction limit was closer to 0.50.

Figure 5 shows the CO emissions for both AVC module configurations. Once again, the CO starts to increase at a higher main equivalence ratio for the Configuration 2 module. As a result, the CO-NOx curve was shifted to the right (see Figure 6).

Figure 4: NOx Emissions For Condition 2

Figure 5: CO Emissions For Condition 2

Figure 6: CO-NOx Curve For Condition 2

3.4 Summary of Other Configuration 2 Results

In previous testing with Configuration 1, the cavity equivalence ratio was found to be an important parameter to reduce the NO_x emissions without significantly impacting the CO emissions. Configuration 1 was capable of achieving significantly lower cavity equivalence ratio conditions, but achieving lower NO_x levels was prevented by sudden flame blow-off events.

Figure 7 shows the NO_x and CO emissions for several different cavity equivalence ratio conditions. All of these tests were conducted at 10 percent cavity loading and a

reference velocity of about 235 feet per second. The data suggests that the cavity equivalence ratio reaches a point of diminishing returns, since the lean extinction limit for the flame begins to shift to higher NO_x levels if the cavity equivalence ratio is too lean. This is an important finding for future combustor development efforts.

Figure 7: CO-NO_x Curves For Range of Cavity Equivalence Ratio Conditions

The effect of cavity loading is shown in Figure 8. Note that increasing the cavity loading from 6 percent to 10 percent seems to make the CO levels turn up more abruptly near the lean extinction limit. The cavity equivalence ratio was maintained at 0.90 for the data shown in Figure 8 and the reference velocity is approximately 235 feet per second for both cases.

Figure 8: CO-NO_x Emissions For Different Cavity Loadings

4.0 Conclusions and Recommendations

The emission characteristics of Configuration 2 were investigated in this test and some comparisons to Configuration 1 were made at select operating conditions. The results suggest that the emission characteristics for the Configuration 2 module are not as low as those observed with the Configuration 1 module. It appears that Configuration 2 shifts the CO-NO_x curve to the right (see Figure 3 and Figure 6) for the same operating conditions. However, it should be noted that the cooling air flow splits may have also changed with the introduction of the Configuration 2 module. These cooling air splits may also have a significant impact on the emission characteristics.

These results also suggest that lowering the cavity equivalence ratio to control NO_x emissions can result in sudden flame extinction at higher NO_x levels. Therefore, it appears that there may be some trade-off with achieving lower cavity equivalence ratio conditions in the cavity.

Appendix 4

Summary of February 2005 Test Results

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Summary of AVC Results From February Tests

1.0 Goals and Objectives of Test

The objectives of the tests conducted from 2/8 to 2/10/2005 were as follows:

- Determine if the curved inlet design provides better LBO and emissions performance relative to the previous straight inlet.
- Pursue a better understanding of the key operating variables associated with the AVC design.

2.0 Experimental Setup

The curved inlet incorporates curvature to the sidewall that turns the flow toward the cavity, or vortex region. A similar configuration was modeled in CFD and the CFD results were encouraging. The intent of this configuration is to create a shorter flame and to produce better mixing between the cavity and main flow. In addition to the curved inlet, this configuration incorporates a different flame-holder shape. This inlet configuration has been tested with the Configuration 2 AVC module, instead of the Configuration 1 module that was tested and reported in December 2004.

3.0 Independent Experimental Variables

The bulk average velocity at the dump plane is controlled by the amount of main air flow. The main equivalence ratio is independently controlled by the main fuel flow which is premixed. The cavity air loading is controlled by the amount of air being injected through the upstream wall of the AVC module. In these tests, the cavity air loading is varied from 5% to 7% of the total combustion air. The cavity fuel flow is used to independently control the equivalence ratio of the premixed fuel-air jets entering from the upstream face of the AVC module. In addition to these four independent variables, the cooling air flow through the liner also affects the total, or overall, equivalence ratio for the entire combustor. This will be discussed in more detail in the following paragraphs. Table 1 is a list of the independent process parameters with the corresponding independent experimental variables.

Table 1: Conversion of Independent Process Parameters To Experimental Variables

Process Parameter	Independent Experimental Variable
Main Air Flow	Reference Velocity, or bulk average velocity at the dump plane
Main Fuel Flow	Main Equivalence Ratio
Cavity Air Flow	Cavity Loading (%)
Cavity Fuel Flow	Equivalence Ratio of fuel-air jets injected into the cavity region
Cooling Air Flow	Total (overall) Equivalence Ratio

4.0 Discussion of Results

4.1 Comparison of Inlet Configurations

In relation to the straight inlet configuration, the curved inlet has been compared at two different operating conditions (see Table 2). The cooling air pressure drop across the liner has been maintained at 2.0% for both of these cases. Furthermore, the combustor back-pressure (132.3 psig) and inlet-air temperature (625F) is the same for all data presented in this report.

Table 2: Table of Experimental Conditions Used For Comparison

Case Number	Main Air Flow (scfh)	Main Equiv. Ratio	Cavity Equiv. Ratio	Cavity Loading (%)
Case 1	54,020	Varied	0.95	6.0
Case 2	47,905	0.65	Varied	6.0

4.1.1 Summary of Case 1 – Main Equivalence Ratio Effects

Figure 1 shows the NO_x performance for both inlet configurations. The NO_x emissions are very similar as the main equivalence ratio is varied. The CO performance for Case 1 is slightly improved with the curved inlet. However, the lean extinction limit is not improved significantly and the flame suddenly blows out of the cavity at a main equivalence ratio of approximately 0.53 (see Figure 2). Note that the emissions data at the lean extinction limit are not shown due to the time lags in the sampling system.

Figure 1: NO_x Emissions For Straight and Curved Inlet Configurations

Figure 2: CO Emissions For Straight and Curved Inlet Configurations

4.1.2 Summary of Case 2 – Cavity Equivalence Ratio Effects

Figure 3 shows the NOx emissions for both entrance configurations as a function of cavity equivalence ratio. The NOx emissions for the curved inlet are lower for the same cavity equivalence ratio, however the lowest achievable cavity equivalence ratio was slightly higher for the curved inlet. One possible explanation for this difference may be due to larger mixing rates from the main flow to the cavity region for the curved inlet. In any event, although the NOx is somewhat lower with the curved inlet, the emissions for this configuration are not as good as the Configuration 1 AVC module. The CO emissions from both the straight and the curved inlets were low for these operating conditions (see Figure 4).

Figure 3: NOx Emissions From Case 2

Figure 4: CO Emissions From Case 2

4.1.3 Summary of Inlet Configuration Results

The differences between the curved inlet configuration and the straight inlet have been presented in Figures 1 – 4. The observations are summarized below.

- With respect to the main equivalence ratio, the lean extinction limit is not significantly altered by the curved inlet configuration. Furthermore, the NO_x emissions are not significantly different with the curved inlet.
- With respect to the cavity equivalence ratio, the NO_x emissions are improved by 1-2 ppm (at 15% oxygen basis). However, the cavity equivalence ratio at which the flame is blown out of the cavity is somewhat higher for the curved inlet.
- The CO emissions are improved slightly with the curved inlet configuration.
- No significant dynamics has been observed with either configuration.

4.2 Investigation of Main Equivalence Ratio, Total Equivalence Ratio, and Residence Time Effects

One of the interesting findings from early AVC test results is the fact that the CO emissions correlate with the total equivalence ratio for a wide range of operating conditions. For example, when the cooling air pressure drop was decreased from 3.5% to 2.0%, the LBO limit shifted to the left when plotted as a function of main equivalence ratio (see Figure 5). However, when plotted as a function of the total equivalence ratio, a quantity that includes the cooling air, the data falls along the same curve (see Figure 6). The same behavior is observed when the reference velocity is increased (see Figure 7). In these early experiments, the cooling air pressure drop was maintained at a constant level and the main equivalence ratio was varied using the main fuel flow. Using this approach, changes to the main fuel flow actually affect both the main equivalence ratio and the total (or overall) equivalence ratio. In other words, these two variables are confounded and were not varied independently. So one unanswered question is which fuel air ratio dominates the combustor performance (i.e., the main equivalence ratio or the total equivalence ratio).

The test facility has been designed such that the cooling air flow can be varied independently from the other air flows, a situation that is not typically available in an actual engine application. The experiment described in the following paragraphs investigates the effects of the cooling air flow by varying this parameter independently.

Figure 5: CO Emissions As A Function Of Main Equivalence Ratio For Different Cooling Air Flows. (Data From December 8, 2004 With Configuration 1 AVC Module).

Figure 6: CO Emissions For Different Cooling Air Pressure Drop. (Data From December 8, 2004 With Configuration 1 AVC Module).

Figure 7: CO Emissions As A Function Of Reference Velocity. (Data From December 8, 2004 With Configuration 1 AVC Module).

4.2.1 Test plan

In order to separate the effects of the cooling air flow, the main air flow, and the main fuel flow, a statistically designed experiment was performed with the goal of understanding the CO emissions. The basic experimental variables that were investigated in this test plan included the following.

- Reference Velocity
- Main Equivalence Ratio, and
- Total Equivalence Ratio

The reference velocity was chosen to essentially change the residence time through the combustor. The main equivalence ratio and the total equivalence ratio are included to investigate which of these variables has the largest impact on the pollutant emissions.

The test plan is shown in Table 3. Note that the changes in equivalence ratio are relatively small changes. These ranges have been chosen based on limitations in the cooling air flow. The test matrix is a two-level factorial design with four replications at the center-point of the domain. The measured values of CO and NO_x will be regressed with a multi-variable linear regression to assess the effects of each of the variables listed in Table 3. The two-way interactions will also be investigated.

Table 3: Test Matrix For February 10, 2005

Trial	Reference Velocity (ft/s)	Total Equivalence Ratio	Main Equivalence Ratio
1	235	0.43	0.60
2	235	0.43	0.57
3	235	0.40	0.60
4	235	0.40	0.57
5	200	0.43	0.60
6	200	0.43	0.57
7	200	0.40	0.60
8	200	0.40	0.57
CP-1	217.5	0.415	0.585
CP-2	217.5	0.415	0.585
CP-3	217.5	0.415	0.585
CP-4	217.5	0.415	0.585

It should be noted that the cavity loading and the cavity equivalence ratio have been maintained at a constant value for these tests. Furthermore, the pressure and inlet air temperature have also been maintained at constant values that are shown in **Table 4**

Table 4: List of Combustor Variables Kept Constant For Statistical Experimental Design

Variable	Target Value For Factorial DoE
Cavity Equivalence Ratio	1.10
Cavity Loading	6.0%
Combustor Pressure	10 atm
Inlet-air Temperature	625 F

4.2.2 Summary of NO_x Results

The NO_x data from this experiment are shown below. Note that the emissions data are corrected to 15% oxygen basis. The effect of the total equivalence ratio is shown by the red lines, and the NO_x emissions tend to increase as the total equivalence ratio increases. The effect of the main equivalence ratio can be observed by comparing corresponding points in each row of sub-plots (e.g. the green arrow). The main equivalence ratio does not seem to play a significant role on the NO_x emissions. The effect of reference velocity can be seen by comparing corresponding points in each column of sub-plots (e.g. the blue arrow). These data suggest that the total equivalence ratio has the largest effect on NO_x emissions. However, a multi-variable regression analysis will be used to assess the statistical significance of these effects.

Figure 8: Scatterplot of NOx Results From Statistical DoE

The results from the multi-variable regression are shown in Figure 9. It should be noted that the regression coefficient for the reference velocity is actually negative which indicates that increasing the reference velocity will actually decrease the NOx emissions. In other words, decreasing the residence time will decrease the NOx emissions, this qualitative trend agrees with theoretical predictions. The fact that the main equivalence ratio has no statistical significance in this experiment suggests that the NOx emissions are not dominated by the fuel-air ratio in the main inlet stream. Instead, the cooling air is playing a dominant role in the NOx emissions over the range of operating conditions investigated in this experiment. This is an important finding for future combustor development efforts.

Figure 9: Graphical Interpretation Of Main Effects On NO_x Emissions. (Note The Dotted Line Represents Statistical Significance At 95% Confidence)

4.2.3 Summary of CO Results

The CO emissions for each of the conditions in the test matrix are shown in Figure 10. In spite of the relatively narrow operating range that has been chosen for this experiment, the analysis of the CO data is complicated by non-linear behavior. Note that the CO emissions at the center-point are much lower than the overall average value. As a result, a linear model like the one used in the previous NO_x analysis does not provide a good fit to the experimental data. A non-linear regression model would be an option, but the test matrix does not include enough data points for a non-linear regression. Another option is to transform the CO data to logarithmic coordinates and then use a linear model for the regression analysis. This approach will be discussed in more detail in the following paragraphs.

Figure 10: Scatterplot of CO Emissions From Statistical DoE

The transformation of the CO data into a logarithmic scale is shown in Figure 11. Note that the center-point is now much closer to the overall average and it appears that a linear model will fit the transformed data much better. Indeed, the residuals from the model that is fitted to the logarithm of the CO emissions are much closer to a normal distribution than a linear fit of the raw CO emissions (see Figure 12). Therefore, the regression analysis using the logarithm of the CO data is preferred.

Figure 11: Scatterplot of the Log(CO) Data From The Statistical DoE

Figure 12: Distribution of Residuals For (a) Linear Regression of Raw CO Data, and (b) Linear Regression of Log(CO) Data

The results from the regression analysis are shown graphically in Figure 13. The dashed lines in Figure 13 indicate 95% confidence limits for each main effect. It should also be noted that none of the two-way interactions were significant in this test. The total equivalence ratio dominates the CO emissions. The fact that the total equivalence ratio effect is negative means that increasing the total equivalence ratio will decrease the CO emissions, which is plausible for the range of operating conditions investigated in this test. The reference velocity effect is statistically significant, but the confidence level is about 96% for this effect. The fact that the velocity effect is positive implies that increasing the velocity, or decreasing the residence time, will result in an increase in CO. Again, this qualitative trend seems plausible. An interesting finding from this experiment is the fact that the main equivalence ratio does not play a significant role in the CO emissions over the range of operating conditions that have been investigated.

Figure 13: Graphical Interpretation of Main Effects on CO Emissions

5.0 Conclusions and Recommendations

In this test, the inlet shape to the combustor was modified based on results from some preliminary computational fluid dynamic (CFD) simulations. The CFD simulations indicated that this inlet shape should produce a shorter flame zone and improve the CO oxidation. Although the CO emissions were improved slightly, the LBO margin was not significantly improved with the curved inlet shape. Some improvement in NO_x emissions was observed at some operating conditions relative to the straight inlet. However, at other conditions there was no significant improvement in NO_x emissions.

In an attempt to better understand the effects of the cooling air flow in the combustor performance, a statistically designed experiment was performed in which the cooling air flow was varied independently. Although the range of operating conditions was fairly limited in this experiment, the results suggest that the cooling air flow, or total equivalence ratio, dominates the production of pollutant emissions (both NO_x and CO). In previous tests, the total equivalence ratio was confounded with the main equivalence ratio by the requirement of maintaining a constant cooling air pressure drop. Based on these results, future efforts to improve the emissions performance of this AVC design should focus on improving the cooling air distribution, or investigating a back-side cooled liner design.

CHAPTER 6: GREENHOUSE GAS AND ALTERNATIVE FUELS STUDY (TASK 15) & GREENHOUSE GAS REDUCTION MARKETING OPPORTUNITIES (TASK 19)

This report has been prepared by an independent consultant to address Tasks 15 and 19 and is included in its entirety as an Appendix. Only the page numbers have been changed in the report to reflect its location within this BP4 Final Report document.

CHAPTER 6 APPENDIX

Report on Reduction of Greenhouse Gases Resulting from Applications of Ramgen Technology

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EXECUTIVE SUMMARY

The Ramgen technology is projected to be a breakthrough in economically viable products for compression and for energy generation. The higher efficiencies and lower emissions of the Ramgen products under development offer the strong prospect that their widespread use would reduce greenhouse gases.

Since the sound barrier was broken in the late 1940's, ramjet engines have been widely used as a means to propel aerospace vehicles at supersonic speeds. The technology is very well understood and fully characterized. Ramgen's primary technical innovation has been to apply ramjet engine concepts in a stationary compressor application.

Specific opportunities for Ramgen products to qualify for credits in the developing carbon credits markets have been identified in order to consider whether carbon emission credits can be an economic incentive in achieving the widespread use of the Ramgen products required for significant greenhouse gas reduction.

The Report addresses four types of Ramgen products. The Rampressor, to compress air and other gases; the Rampressor Turbine, to generate electricity; the Ramgen Engine, to generate electricity that can use waste gases as fuel; and, a hybrid that uses the Rampressor Turbine with a fuel cell. There is an assessment of the priority applications of the products to achieve the greatest reductions in emissions of greenhouse gases and maximize the economic incentives from efficiency savings and from carbon credits.

A) The Rampressor-- Importance of Breakthrough Compression Technology

The GHG emission reductions from the Rampressor are achieved by improving the efficiency of compressing air and other gases; utilizing the heat generated by compression; and, using a gas-fired drive (rather than electrical powered) in some applications. The Rampressor will be the first commercial Ramgen product.

DOE has cited higher efficiency in compression as a national priority. The air and gas compression industry is referred to as the Fourth Utility. Compressed air is one of the most important utility requirements of industrial manufacturing. In addition, efficient compression of CO₂ is pivotal to effectively dealing with green house gas emissions.

The four principal advantages of shock compression compared to existing compressor technologies are the Rampressor can achieve exceptionally high compression efficiency; is capable of producing very high single-stage compression ratios; has a simplicity and size making it less expensive to manufacture; and, offers the opportunity for significant waste heat recovery. The combined heat and compression (CHC) system efficiency for the Rampressor is approximately 60% versus less than 20% system efficiency for conventional designs.

Eligibility of the Rampressor Products to Earn Carbon Credits

The Prototype Carbon Fund identifies five Energy Efficiency Improvements Project types that qualify to earn carbon emission reductions. The Rampressor would qualify for four

of the five.

Rampressor Priority – Applications Using Both Compression and Heat

The maximum greenhouse gas reductions can be achieved with the Rampressor in applications using both compression and heat. In order to identify the most economically viable applications in this Report the analysis has been to identify industries with a need for compression and for heat.

Review of Industrial Compressor Applications

One DOE study found that electricity used to drive air compressors is heavily concentrated in three manufacturing groups: chemicals, primary metals, and petroleum manufacturing.

The total compressed air use and percentage of compressed air to the total electrical demand for the industry are two factors in identifying priority applications for the Rampressor for greatest possible economic viability and impact on greenhouse gases.

Use of Waste Heat

The use of waste heat is the enormous advantage that the Rampressor has over other options to reduce greenhouse gases by bringing higher efficiency to the compression process. The 625 to 640 degree F waste heat generated by the Rampressor can be recovered for a number of different processes. The processes where the recovered heat can be used include: Direct Contact Water Heater; Indirect Liquid Heating; Direct Heating & Drying; Indirect Air/Gas Heating; Refrigeration and Freezing; Dehumidification; Exhaust Gas Oxidant (non-boiler); and, Steam.

Rampressor Savings

The combined value of the electricity saved and carbon credits earned by Rampressor technology operating in the U.S. in only three size ranges as a stand alone, combined heat and compression, and a gas-fired oil-free air compressor is estimated, after twenty years, to result in an annual value of \$2 billion dollars in the United States.

B) Rampressor Turbine

The Rampressor Turbine, the first Ramgen product to produce electricity, reduces emissions of greenhouse gases by improving the efficiency of electrical generation and by operating in venues where the use of exhaust heat (combined heat and power) is economically viable with this technology. Like the Rampressor, the Ramgen Turbine will qualify for carbon credits in four of the five categories of the Prototype Carbon Fund.

The Rampressor Turbine is a small engine (250 kW to 5 MW) for electricity generation that combines the compression efficiency of the Rampressor with advances in small-recuperated turbines. The Recuperated Ramgen Turbine can approach 40% without the development challenge posed by the need for advanced ceramic components. It is projected in this report that the combination of this high efficiency, low capital cost, and reliability will stimulate large-scale commercial Distributed Generation and Combined Heat and Power. This is of national importance at a time when the electrical grid use is at

maximum, and there is opposition to large-scale new power plants near urban centers.

The Importance of Breakthrough Ramgen Technology to Distributed Generation

There is general agreement in many DOE supported studies that Distributed Generation, with Combined Heat and Power, would: 1) dramatically reduce fossil fuel use and air pollutant emissions; 2) improve power quality, efficiency, reliability and return on investment; and 3) enhance energy security.

Despite these projected benefits, the growth of DG, beyond the considerable onsite generation that already exists in U.S. industries, has been slowed by many factors. One critical factor in the assessment of some experts is that in order for DG to come into widespread use a highly efficient, low cost and reliable small engine with low emissions has to be commercially available.

When a reliable small engine can compete with grid power on cost per kW without considering the heat then in the assessment of some experts DG will be used much more extensively and this will result in the use of the heat as well. In this report, the assumption is made that large scale additional DG has to be driven by competitively priced electricity and that as DG begins to be applied widely then the use of the heat will follow naturally. The only small engine with this potential that is now approaching a commercial model of which this Report is aware is the Ramgen Turbine.

Priority Application of Rampressor Turbine:

Industrial Process Heating Systems

The priority marketing opportunity for reducing greenhouse gases with the Rampressor Turbine is in industrial processes that can use the heat from small engines. A DOE supported study outlines how industrial CHP potential tends to be strongest in areas with relatively high electricity prices and concentrations of industrial sites. Current CHP potential is concentrated in the Pacific (led by Southern California), East North Central, Mid Atlantic, and South Atlantic regions.

The “Steam System Opportunity Assessment for the Pulp and Paper, Chemical, Manufacturing, and Petroleum Refining Industries” produced by the DOE focuses on the CHP potential led by Chemicals, Primary Metals, Paper, and Food. Together, these four industries represent over half of the market potential. Another DOE study has analyzed the potential for on-site power generation in the U.S. industrial sector with a focus on what is termed DOE Industries of the Future (IOFs). Total purchases for electricity and steam for these IOFs are in excess of \$27 billion annually.

The CHP Cooling Market

The CHP cooling market potential is led by the Electronics industry, followed by Chemicals, Petroleum, and Food. As shown below, these four industries account for over two-thirds of the current cooling capacity as well as the potential for CHP-driven absorbers.

Remaining CHP Potential in the Industrial Sector

Existing on-site generation capacity in the industrial sector, exclusive of emergency generator sets, is in excess of 45,000 MW with a vast majority (42,000 MW) being combined heat and power. The remaining CHP potential in the industrial sector is estimated at 88,000 MW. Remaining potential for the Industries of the Future industry groups is estimated at 61,000 MW or 69% of the total CHP opportunity. The CHP potential includes systems that export power to the grid meaning that the estimated potentials are in excess of site (in-fence) demand requirements.

CHP Economics and Greenhouse Gases

The economics of CHP with current technology indicated that a 5 MW CHP system is competitive with delivered electricity prices for 37 percent of industrial customers (current gas prices are a critical variable). A 1 MW reciprocating engine system is competitive in 20 percent of the industrial sector. The Rampressor Turbine will be more price competitive than current small engine options used in these calculations.

CHP's competitiveness is sensitive to fuel price and declines as natural gas prices increase for each CHP system. The environmental value of CHP is most important for its potential to reduce global climate change emissions. One study estimates 130 GW being generated from CHP, which would produce a 70 million metric ton reduction in carbon equivalent emissions.

The estimated remaining CHP technical potential for the industrial sector with plant sizes 1 MW or less is approximately 88,000 MW. The paper, chemical, food, primary metals and refining sectors account for approximately 61,000 MW or 61 percent of the remaining CHP potential. Paper has the largest remaining CHP potential of any one sector, accounting for 26,000 MW. However, significant remaining potential exists in other industries such as textiles, rubber and plastics, metals fabrication and equipment, industries that have not aggressively implemented CHP.

Combined Heat and Power in the Building Industry

One study concludes that by combining on-site power with technologies to provide cooling, heating, humidity control, energy storage and/or other process functions using normally wasted heat there would be a reduction of almost 18 million metric tons of carbon dioxide emissions (based on carbon equivalent) annually by the year 2020.

CHP Potential at Federal Sites

A study considering CHP at Federal building concluded the Federal building types with greatest CHP potential are hospitals, industrial, and R&D facilities. Agencies with most potential are the military, Department of Veterans Affairs (VA), and Department of Energy (DOE).

Conclusions for Rampressor Turbine

The following conclusions were made as to the priority applications of the Rampressor Turbine to reduce greenhouse gases: 1) The food processing industries as there is high electrical use and need for heat; experience with CHP; and, need for small engine; 2) The

chemicals; forest products (includes paper), fabricated metal products; industrial machinery and equipment; transportation equipment; textiles; and rubber; 3) Cooling with CHP; The building industry particularly with Hospitals and Colleges with the 24 hour usage of these facilities and the experience they have with CHP; and, 4) Selected DoD and DOE facilities and VA hospitals.

C) Ramgen Engine

Eligibility of the Ramgen Engine to Earn Carbon Credits

The unique capability of the Ramgen Engine to burn methane in waste gases, which currently are a major contributor to greenhouse gases, results in the engine qualifying for carbon credits in four of the Prototype Carbon Fund's general project category and in two of the Fund's "Other Project Activities" category.

Applications of Ramgen Engine to Reduce Emissions of Greenhouse Gases

Ramgen Engine, with the patented AVC system (advanced vortex combustion), is projected to further increase efficiency and provide the capability for use of waste gases as fuel. The Ramgen engine can be sized from 800 kW to 10 MW. The work underway on the Rampressor and Ramgen Turbine engine set the technology base for the Ramgen engine. Methane is rated as the second largest anthropogenic¹ greenhouse gases contributor, after carbon dioxide, to global warming as it traps, per volume, 21 times more heat in the atmosphere than carbon dioxide. Coalmine and landfill methane are selected as a priority application of the Ramgen engine because they both emit significant amounts of methane into the atmosphere adding to greenhouse gases, offers the potential to capture methane economically.

Coal Mine Methane

Officials at the Jim Walter Resources mining complex in Alabama, working with Ramgen consultants, gave leadership to the strategy developed for this report for coal mine methane. The highly innovative approach focuses on draining much larger amounts of methane from bleeder and gob wells, where the methane is at a higher concentration than in the main ventilation shaft. This strategy results in 75% of the total methane released from coal mines worldwide being used as a fuel by the Ramgen engine.

The Ramgen Engine is projected to have compelling advantages using coalmine methane emissions as fuel to generate electricity as the Ramgen Engine: can efficiently use the coalmine methane at low concentrations as fuel; is projected to have low emissions of NOx; does not require pre-compression of fuel; can operate at partial loads without losing significant efficiency; is less susceptible to damage from impurities in the fuel stream than other engines; and, has fuel flexibility, high efficiency, low emissions, low maintenance costs and low capital costs.

An example of the significance of this approach is that if the proposed ventilation systems and the Ramgen Engine had been in place at coal mines in the 2000, an estimated 450 MMTCO₂E would have been secured. For coal mine operators the

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¹Source of greenhouse gas resulting from human activity

economic incentive to install this technology would have equaled \$4,831,280,418 per year in 2000 from the combined sales of electricity² and carbon credits earned³ by utilizing and preventing the methane in bleeder vents, gob well, and sealed areas of active coalmines from entering the atmosphere.

Landfill Methane

Applying the assumptions made in this Report the use of the Ramgen Engine could, for purpose of example, have reduced methane worldwide by 423 MMTCO₂E in 2000. In developing this scenario it was calculated that landfill operators would have financial incentives from the sale of electricity generated and methane destroyed of \$4,412,264,081 in 2000.

D) Rampressor Turbine in Hybrid Configuration with Fuel Cell

A project is now underway to use the Rampressor Turbine in a hybrid configuration with a fuel cell. An analysis on projected greenhouse gas reductions from this hybrid has not been done but, based on computer modeling, efficiencies of 70% are possible in a fuel cell/Rampressor Turbine the benefits for greenhouse gas reduction would be significant.

The Rampressor Turbine Engine is particularly well suited for use in hybrid configuration with fuel cell and the interface designed for this project. The Rampressor Turbine engine utilizes super-sonic inlet technology to provide a compressor that can reach desired pressure ratio in one stage with superior simple cycle efficiency. Ramgen will optimize the sub-components of the gas turbine engine, namely, the compressor, combustor, and turbine to contribute directly to the improvement of the overall system efficiency.

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² Assumes \$0.02 profit per kWh

³ \$5.52 per MTCO₂e

REPORT OVERVIEW

The Ramgen technology is projected to be a breakthrough in economically viable products for compression and for energy generation. The Ramgen products will feature higher efficiencies, with lower costs and emissions. In addition, there is a strong prospect that the widespread use of the Ramgen products being developed will reduce greenhouse gases. This Report includes:

- 1) A short review of the Ramgen technology;
- 2) An update from an earlier Report on the status of carbon credits for greenhouse reduction;
- 3) Analysis of how specific Ramgen products are projected to reduce greenhouse gases and of how carbon credits may be an economic factor in driving the widespread use of these products. The objective of this analysis is to identify the priority applications of the Ramgen products being developed.

INTRODUCTION - RAMGEN TECHNOLOGY

Since the sound barrier was broken in the late 1940's, ramjet engines have been widely used as a means to propel aerospace vehicles at supersonic speeds. The technology is very well understood and fully characterized.

All conventional, subsonic jet engines feature discrete compression, combustion and turbine/expansion sections to create the thrust used to propel an aircraft. In operation, hot pressurized exhaust gas expands through the turbine to drive the compressor, and then further expands through a nozzle, creating forward thrust.

Ramjet engines feature these same discrete compression, combustion and expansion sections in a much simpler format. At supersonic velocities, air is ingested into the engine and flows around a fixed obstructing body in the center of the engine duct, "ramming" the air flow into channels between the center-body and the engine's sidewall. Inside these channels, the airflow is almost instantaneously slowed to subsonic speeds, creating "shock waves." These shock waves are associated with a dramatic increase in pressure, or, in other words, "shock compression." As with conventional subsonic turbine engines, fuel is then added and the hot, pressurized exhaust gas expands through a nozzle to create forward thrust.

Ramjets are simple, with no moving parts, but the aircraft has to be moving at supersonic speeds to initiate the shock necessary for effective operation. As a result, all ramjet experience has been in the context of supersonic planes and missiles.

Ramgen's Technology Breakthrough

Ramgen's primary technical innovation has been to apply ramjet engine concepts in a stationary compressor application.

Rampressor Rotor

Ramgen's core design, the Rampressor, is a relatively simple device. It features a rotating

disk, which operates at the high peripheral speeds necessary to achieve supersonic effect in a stationary environment. The rim of the disk has three raised sections and cavities that mimic the effect of the center-body and channels of a conventional ramjet inlet. Air entering through the front inlet is ingested into a space between the supersonically spinning disk and the outer edge of the disk casing. When air enters this space, the raised sections of the disk rim create a “ramming” effect, generating shock waves and air compression in a manner completely analogous to ramjet inlets on supersonic vehicles. The efficiency of this compression process is very high because the compressor has very few aerodynamic leading edges, and minimal drag.

Ramgen Air Compressor Rotor Disk

The disk chambers or “strakes” are angled, so the compressed air is “augured” via rotation into a collector and then on to the compressed air system. The compression process is inherently oil-free, requiring no oil for lubrication and/or sealing.

The shock effect begins at the speed of sound. The higher Mach achieved, the greater the resulting shock wave. The strength of the shock is thus dependent on the disc speed and the properties of the particular gas ingested into the Rampressor.

The strength of the shock wave, hence the amount of compression, increases exponentially with the Mach number. For example, in air at Mach 1.6 a compression ratio of 3.5:1 is achieved, while at Mach 2.4 it is approximately 15:1. The higher Mach number is achieved by spinning the disc faster.

Ramgen Compression Technology Opportunity Matrix

The Ramgen compression process has significant advantages with the large number of applications that require high compression ratios. These applications include industrial air compressors, turbochargers and gas turbines of all sizes and designs. Low compression ratio applications can benefit from improved efficiency as well as those with high compression ratios. This range of potential applications that Ramgen technology can benefit has lead to initiating work on a simplified, “normal shock” inlet system that can be used as a “drop in” efficiency upgrade in conventional designs and platforms. HVAC chillers are included due to the relatively high molecular weights and lower inlet temperatures associated with both the fluids and the application.

To the best of Ramgen’s knowledge, no other organization is currently undertaking significant research and development using supersonic shock wave technology in a stationary platform.

Development of Two-Stage Compressor and Additional Products

With commercial success on the single-stage product line, a two-stage, intercooled design will be offered in sizes from 600 to 2000hp. These packages will feature two Ramgen compression stages driven directly by a double ended high speed drive, also equipped with magnetic bearings. These two-stage packages will be sold below prices for comparable offerings and, depending upon size, are expected to have efficiencies 3-5% better than conventional three-stage designs without factoring in the much larger

efficiency gains unique to the Ramgen compressor from using excess heat.

After successfully addressing the air compressor markets, Ramgen anticipates that modifications could quickly be made to offer single-stage compressors suitable for gas turbine power generation and propulsion systems and for process gas applications. The strategy will be similar to that of the air compressor initiatives.

There are several gases that offer substantial opportunity, these include CO₂ compression for sequestration applications; natural gas compression, to support the higher compression ratio CCGT engines currently being installed and to serve as high efficiency pipeline boosters; and, a wide range of refrigerants to support commercial and industrial space cooling and refrigeration.

INTRODUCTION – CARBON MARKETS

In a Report submitted to DOE by Ramgen in October of 2003, Ramgen outlined the developing carbon market and the potential for Ramgen technology to qualify for credits. In this current Report, the specific opportunities for Ramgen products have been identified in detail. The objective in considering the carbon market is to consider the prospects for carbon emission credits being an economic factor in motivating the widespread use of the Ramgen products. The development of the carbon markets has continued to evolve and has changed significantly over the last year.

Governments and companies have started to take steps to reduce their greenhouse gases (GHG) emissions either voluntarily, or, increasingly, because of current or expected regulations. Since GHGs mix uniformly in the atmosphere, from an environmental standpoint a reduction in one place is equivalent to the same reduction anywhere in the world. Most of the regulations constraining GHG emissions and most of the voluntary actions take advantage of this substitutability and allow for the purchase of emission credits both within and outside of the regulated area, thereby laying the ground for the projected “carbon market.”

The emerging carbon market is one of the few markets for environmental services currently in operation; and the only one with a worldwide reach. Abatement costs are projected to be lower in transition economies and developing countries and therefore, the carbon market is expected to be an occasion to reduce greenhouse gases at lower costs, by making new public and private investment in clean technologies to economies in transition and developing countries.

The goals, and the full potential of the carbon market, however, are far from being realized and many obstacles remain. This Report draws from multi sources, including the World Banks Carbon Surveys, to review the status of the carbon market in July of 2004 and identify emerging trends.

The term “carbon market” encompasses a wide range of disparate transactions, involving various underlying assets, contractual structures, and governing regulations not all of which are relevant to Ramgen technology.

MARKET STRUCTURE

Two Types of Carbon Transactions

Carbon transactions can be grouped in two main categories:

1. *Trades of emission allowances*, for example, Assigned Amount Units (AAUs) under the Kyoto Protocol, or allowances under the EU Trading Scheme (EUAs). These allowances are created and allocated by a regulator, usually under, a cap-and-trade regime. (Not Applicable to Ramgen products)
2. *Project-based transactions*, (this is the category in which Ramgen products qualify for credits). These are projects, which reduce GHG emissions compared

which what would have happened otherwise, and get “emission credits” in return. Unlike allowance trading, a project-based transaction can occur even in the absence of a regulatory regime: an agreement between a buyer and a seller is sufficient.

Category 1 emissions allowances account for only a small fraction of the total volume of assets exchanged on the carbon market (only 3 percent in 2004), although they represent the majority of transactions (more than two-thirds over the same period). The main reason for this imbalance is that allowance markets are still largely in their infancy, with current high-volumes of activity only in the UK allowance market. This situation is likely to change rapidly as details of the National Allocation Plans (NAPs) for the European Trading Schemes (EUETS) are released, and as allowance markets become operational in Canada and elsewhere.⁴

Most of the assets traded are from two types of project-based transactions; 1) projects intended for compliance with the Kyoto Protocol, i.e. under either Joint Implementation (JI) or the Clean Development Mechanism (CDM) programs being developed as part of that system; and, 2) projects not intended for Kyoto Compliance. The non Kyoto group covers a wide range of transactions involving buyers with different objectives who are operating in different regulatory situations.

Segments of the Carbon Market

Buyers primarily engage in carbon transactions because of regulatory pressure (present or anticipated). These regulations, at an international, national or sub-national level, define various segments of the carbon market.

Globally, the most prominent of these anticipated regulation systems is the Kyoto Protocol (1997), which calls for industrialized countries and economies in transition not to exceed certain GHG emission targets during the period 2008-2012. In addition to domestic policies and measures, countries that have ratified Kyoto termed “Annex B Parties” can purchase Assigned Amount Units (AAUs), and implement emissions-reducing projects. These projects are under the categories of “Joint Implementation” (JI), and in non-Annex B countries “Clean Development Mechanism” (CDM).

The EU Emissions Trading program is similarly structured for trading credits generated by large-scale projects reducing greenhouse gases within the European Union. A linking directive governs the relationships between the EU-ETS and the Kyoto Protocol. It allows entities under the EU-ETS to use emission reduction credits from either JI or CDM projects against their targets under the EU-ETS under certain conditions. When the EU Trading program starts operating on January 1, 2005, the earlier U.K. trading system will merge into the wider trading system.

Other Annex B countries, including Canada and Norway, are developing similar domestic cap-and-trade schemes as part of their strategies to meet their Kyoto targets. It is our understanding these will allow entities subject to emissions quotas to use project-

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⁴State and Trends of the Carbon Market 2004”, World Bank, June 2004

based mechanisms to offset part or all of their obligations. These trading systems, however, are not yet operational. Mitigation plans are less advanced in many nations such as Japan, where it is not clear yet whether industrial sectors and electric utilities will face mandatory emission reduction requirements, and whether a domestic emissions trading program will be created.

All the international or national regulations described above are related to, and driven by, the Kyoto Protocol (although most are formally independent of it). The Kyoto Protocol is not yet in effect and may never come into effect if Russia does not ratify it.

Possible Systems for Carbon Trading at Local Levels

Regulations constraining carbon emissions have been developed in the U.S. and Australia, even though the two countries have announced that they would not ratify the Kyoto Protocol. For example, the state of New South Wales in Australia imposes benchmarks on the emissions of utilities and other entities, and allows for the purchase of GHG Emissions Certificates to meet these targets. In the U.S., many states are adopting policies to reduce GHG emissions.

States in the U.S. are taking a range of approaches, from comprehensive, cross-cutting programs to those more narrowly focused on issues such as energy, air pollution, agriculture, transportation, natural resources, and education. While some state programs are expressly designed to confront the challenge of global warming, others are designed to achieve different policy goals but have the additional effect of reducing GHG emissions.

The issue with the state actions in this Report is whether some may influence carbon credit trading programs that would affect the economic viability of the Ramgen products. Twenty-eight states and Puerto Rico have developed, or are developing either strategies or action plans, to reduce net GHG emissions. Several states have set numeric goals for reducing emissions to mitigate climate change.

- New Jersey has committed to reduce GHG emissions to 3.5 percent below 1990 levels by 2005.
- Massachusetts capped CO emissions along with SO and NO from its six highest-emitting power plants, with a goal of reducing CO emissions from those plants by 10 percent over the next several years. Plants that fail to make adequate on-site reductions can still meet the standard by purchasing emissions credits.
- New Hampshire passed a law to reduce power plant emissions of CO₂, SO₂, NO_x, and mercury. The program requires a reduction of CO₂ emissions to 1990 levels by 2006.
- Oregon enacted a law in 1997 establishing CO₂ standards for new power plants; the required reduction of 17 percent can be achieved through offset projects coordinated by the non-profit Climate Trust.
- Washington is in the process of adopting rules similar to Oregon.

At a regional level, the New England governors signed an agreement in August 2001 with the Eastern Canadian Premiers for a comprehensive regional Climate Change Action Plan, including regional emissions reduction targets. In June 2003, Maine became the first state to codify this agreement in state law. In April 2003, New York's Governor Pataki sent out letters to the governors of ten other Northeast states inviting them to participate in the development of a regional CO₂ cap-and-trade program for power plants. The governors from Maine, New Hampshire, Vermont, Massachusetts, Rhode Island, Connecticut, Delaware, and New Jersey responded with interest in working with New York. Maryland and Pennsylvania are participating under an "observer" status. The group, called the Regional Greenhouse Gas Initiative (RGGI), has begun to work on the program design, which it aims to complete by April 2005. In September 2003, the governors of Oregon, California, and Washington directed their staffs to cooperate during the next year to arrive at policy recommendations for reducing GHG emissions.

These 'non-Kyoto' systems in countries such as the U.S. are different from and sometimes less stringent than the proposed 'Kyoto' ones, in terms of the constraints they impose. It is unclear at this stage if some of these systems will ultimately be linked with the 'Kyoto' trading system.

Business Activity

The extent to which there is strong interest from many corporations in reducing greenhouse gases, regardless of the reasons, enhance the prospects for the widespread use of the Ramgen products. In addition to responding to mandatory regulations, some corporations are already engaged voluntarily in the carbon market for various strategic reasons. Their participation often takes the form of project-based transactions.

There is also trading activity in the retail market. This market consists of companies and individuals without significant emissions (and therefore unlikely to be regulated under domestic regulations) who wish to be climate-neutral in order to demonstrate their social responsibility or promote a particular brand. These entities purchase emission reduction credits in small quantities. Several "traders" serve this small but growing market, by implementing larger emission reduction projects, and then retiring slices of the emission reductions for their customers.

A growing number of major companies are undertaking significant efforts to address climate change. These efforts include setting GHG reduction targets, investing in the installation of more efficient energy technology; and participating in emissions trading.

Examples of Cogeneration Projects

- AEP has constructed a 900 MW state-of-the-art natural gas cogeneration facility for Dow Chemical Company to provide energy and steam to its Plaquemine, LA site.
- ABB has built approximately 1,500 small combined heat and power plants in Europe, reducing GHG emissions by 60 percent compared to coal-fired power plants and by about 30 percent compared to natural gas-fired plants.
- Air Products and Chemicals' larger hydrogen plants function as cogeneration

facilities. Steam and power are often produced in addition to hydrogen and exported to a nearby user. The energy efficiency of these hydrogen plants is over 85 percent of what is theoretically achievable. A cogeneration unit was also installed to provide electricity, heating, and cooling at the Air Products Hersham UK European headquarters, which reduced CO₂ emissions by 2,700 metric tons per year. Two Air Products European operations have contracted to purchase electric power from cogeneration units operated by customers, resulting in lower GHG emissions than would buy power off the public utility grid.

- Georgia-Pacific and Weyerhaeuser employ cogeneration in their integrated pulp and paper mills.
- SC Johnson broke ground for its Landfill Gas Green Energies initiative in April 2003, installing a turbine that produces electricity and steam through cogeneration for its largest manufacturing plant, Waxdale, in southeastern Wisconsin.

Promoting Energy Efficiency Examples of Energy Efficiency Projects

Companies are reducing energy use and GHG emissions by providing energy-saving products. The extensive activity in this area indicates that there will be strong interests in the savings that will be available from the Ramgen compressor products.

- AEP plans to investment over \$43 million over the next three years to achieve over 247 million kWh of energy efficiency gains.
- Air Products and Chemicals' energy efficiency projects resulting in an estimated 26 MW of power savings, equivalent to avoiding 174,000 tons of CO₂ emissions.
- Cummins Inc. has committed to cutting electricity consumption by 6 MW on peak demand days.
- PG&E Corporation's utility, Pacific Gas and Electric Company, reduced overall energy use in 2002 at 88 of its California facilities by almost 24 percent compared with 1998 baseline energy-use levels. This resulted in savings of almost 28 gig watt-hours of electricity, and prevented approximately 7,000 tons of CO₂ from being emitted to the atmosphere.
- Sunoco's energy consumption at refineries in 2002 on a per-barrel basis was 25.3 percent lower than in 1990. At its chemical plants, energy usage was 11.7 percent lower in 2002 than in 1990.
- Between 1999 and 2002, United Technologies' energy-efficiency program, covering more than 200 facilities and more than 30 countries, reduced energy consumption by 27 percent and enabling a 15 percent reduction of GHG emissions. Ute's Otis Elevator's Breslau plant in the Czech Republic heating and lighting equipment retrofit and compressed air system improvements save more than 1 million kWh of electricity and 560,000 cubic meters of natural gas annually.
- PG&E Corporation's Pacific Gas and Electric Company (PG&E) through the systematic replacement of equipment has adopted cost-effective technologies. Efforts in this area continue to include focused inspections and maintenance at compressor stations, modifying system operations to reduce venting, and reducing frequency of engine restarts with gas.
- Toyota reduced the energy required to produce a vehicle manufactured in its

North American facilities by 7 percent in fiscal year 2002 through process improvements, such as reducing compressed air usage by improving system operating control, and the development of waste-heat recovery systems in painting shops.

Waste-to-Energy

- DTE Biomass Energy operates 29 landfill-gas recovery projects at sites across the United States. Methane recovered from these projects is converted into electricity, steam, or pipeline-quality gas. DTE Biomass landfill projects have captured the equivalent of more than 25 million metric tons of CO₂.
- Entergy has funded a project in the eastern United States that will collect coal mine methane vented from abandoned mines and convert it to pipeline-quality gas or use it as fuel to generate electricity. The project will reduce GHG emissions by 400,000 metric tons of CO₂e through 2005.
- Interface Flooring Systems has contracted with the City of LaGrange, Georgia to use methane gas from the local landfill to replace up to 20 percent of its natural gas usage at the LaGrange, Georgia manufacturing facility.
- SC Johnson has refined technology to enable cleaner mixtures of landfill gas and natural gas to fuel boilers. This provides over a third of its Waxdale plant's steam energy needs. The company has also installed a 3.2 MW turbine and generator to burn waste methane gas that would otherwise be flared into the atmosphere. SC Johnson burns the methane gas instead of fossil fuels, such as natural gas or coal, to generate electricity and steam for the site's operations, providing about 50 percent of the facility's electricity and 20 percent of its process steam. Through this system, the company expects to reduce Waxdale's emissions of CO₂ and other GHGs by 47 percent and cut fossil fuel energy use nearly in half by 2005.
- United Technologies' subsidiary UTCFC has designed fuel cell power plants fueled by methane from landfills and anaerobic digester gases from wastewater treatment facilities. These systems eliminate methane emissions and avoid burning fossil fuels. UTCFC operates these facilities in California, New York, Massachusetts, and Oregon, as well as in Germany and Japan.

Emissions Trading and Offsets

A number of companies, anticipating the establishment of domestic and international GHG emissions trading systems, are investing in a variety of emissions trading activities. Although the United States has withdrawn from the Kyoto Protocol, U.S.-based companies whose overseas operations will be subject to emissions limits, or whose domestic operations will be subject to state or regional limits, are likely to participate in emissions trading.

Trading activities undertaken by companies include the transfer of Verified Emissions Reductions (VER) to or from other companies, collaboration with developing country partners in projects that generate emissions credits, participation in the development of voluntary trading systems, and participation in state-level programs. In some cases, these activities were undertaken in part to explore or demonstrate the feasibility of Joint Implementation (JI) and the Clean Development Mechanism (CDM), two of the trading

mechanisms established under the Kyoto Protocol.

Emissions Trading

- AEP, Baxter, and D u P o n t are founding and active members of the Chicago Climate Exchange (CCX). These companies made legally binding commitments to reduce their GHG emissions by four percent below the average of their 1998 to 2001 baseline by 2006.
- BP operated an internal emissions trading system between 1999 and 2001 that helped reduce operational GHG emissions by 10 percent. BP's UK Upstream and Petrochemicals assets are now part of the UK Government's emissions trading scheme (ETS). BP carried out the first trades in the UK ETS and has helped customers trade in the market. BP is currently applying the evolving CDM rules and procedures to a BP solar project in Brazil, with the intention of registering the project with the CDM Executive Board. BP is currently piloting CDM projects for a range of technologies.
- Ontario Power Generation (OPG) and US Gen New England (US Gen) successfully completed a GHG emissions trade in April 2000. US Gen sold OPG 1 million metric tons of CO₂e reductions generated by capturing and destroying methane that would otherwise be emitted from the Johnston Landfill in Rhode Island from 1998 to 2000. OPG has committed to have all of its emissions reduction purchases verified by the Ontario, Canada Pilot Emissions Trading Project and report them to Canada's Climate Change Voluntary Challenge and Registry Inc., where they are transferred and retired.

PROJECT-BASED TRANSACTIONS SUMMARY

The following table summarizes the volumes exchanged and number of projects for both allowance trading and project-based transactions. Project-based transactions account for 98 percent of the total volume of assets exchanged since 1996. (2004 these trades were 95 percent). This imbalance (in volume of assets terms) is due to that allowance markets are still emerging.

Emission Credit - Volumes and Transactions

	Volume (vintages up to 2012, In tCO ₂ e)		# of Transactions		Average size of transactions 1998-2004 (tCO ₂ e)
	1998-2004	2004	1998 - 2004	2004	
Project-Based Total, including:	293,611,881	64,870,588	360	44	848,589
Kyoto compliance	151,890,882	61,394,093	126	27	1,234,885
Voluntary	139,148,129	2,299,050	124	9	1,209,984
Retail	1,493,870	98,445	108	6	14,093
Allowance Trading (estimated)	7,218,183	2,088,408	765	97	9,436
TOTAL	300,830,064	66,958,996	1,125	141	267,405

(State and Trends of the Carbon Market 2004", World Bank, June 2004)

Project-Based Transactions

The first figure shows the annual volumes traded through project-based transactions from 1996 to May 2004. This figure includes only the amount of carbon contracted (projects often produce a larger amount of ERs than are actually contracted), and only up to 2012.⁵

Volumes have been between less than one and 30 MtCO₂e from 1996 to 2002, before going up to 78 MtCO₂e in 2003, a 2.5 times increase compared with the previous year. Since the number of projects in 2003 also more than doubled compared with the past years, it is clear that this increase reflects genuine growth in the market. Interviews with market players confirm that this growth is reflected in their transaction pipeline.⁶

Rapid growth in the market appears to be continuing in 2004. A total volume of nearly 65 million tCO₂e was exchanged in the first five months of 2004, which is nearly as much as the whole year 2003, and the market is on track for another doubling.⁷

In all, an estimated 300 million tCO₂e have been contracted in the market through project-based transactions since its inception. These projects could generate up to 525 million tCO₂e of ERs over their lifetime.⁸

The third figure also offers a breakdown of the annual volume by market segment. The bars represent "not-for-Kyoto" emission reductions and those that, while they are pre-compliance, are transactions intended for compliance with the Kyoto Protocol. Although it is not always easy to distinguish—at the margin—between voluntary and pre-compliance transactions, the analysis leaves little doubt that pre-compliance with the

⁵ State and Trends of the Carbon Market 2004", World Bank, June 2004

⁶ State and Trends of the Carbon Market 2004", World Bank, June 2004

⁷ State and Trends of the Carbon Market 2004", World Bank, June 2004

⁸ State and Trends of the Carbon Market 2004", World Bank, June 2004

Kyoto Protocol now motivates the vast majority (96% in 2004) of the transactions in volume terms.⁹

Sellers

From 1996 to 2000 the majority of project-based transactions took place in industrialized countries (both buyers and sellers were located in industrialized countries). The situation, however, has evolved rapidly. The share of ERs contracted in transition economies and developing countries rose steadily from 38 percent in 2001 to 60 percent in 2002, 88 percent in 2003, and 93% in the first months of 2004. Most of the remaining projects are in fact still related to the Kyoto Protocol. For example, one JI project has been undertaken in New Zealand, and the Government of New Zealand has granted Kyoto credits to firms in exchange for their implementing emission reduction projects.¹⁰

Balance among Project Types

Landfill gas used as fuel for energy projects dominated in 2002-2003, but HFC destruction projects now account for the largest share of the emission reductions produced in 2003-2004, with 35% of the total volume supplied. This is a major shift and the result is consistent with the fact that mitigating HFC emissions is comparatively cheap per tonne of CO₂e mitigated because of the extremely high global warming potential of HFC, and with the fact that the first methodology approved by the Executive Board was related to this technology combine to make this type of projects very profitable. Because HFC has an exceptionally high global warming potential, only two HFC projects account for the total volume of reductions.¹¹

Beyond HFCs, the distribution of technology in 2004 is consistent with 2003. Landfill gas capture represents 20 percent of the volume, followed by biomass (12 percent) and hydro (12 percent). Taken together, renewables account for 29 percent of the total volume of project-based ERs transacted; and the majority of projects (53 percent) are related to power generation.¹²

This distribution is consistent with projects involving methane capture, such as landfill gas to energy or biomass waste recovery being those where carbon finance make the most significant contribution to, and might, even at current carbon prices, render projects economically viable.

The size of recent project based transactions (excluding retail segment) range from around 500,000 tonnes of CO₂e and to about 5 million tCO₂e, with one exception at more than 20 million. Over the past two years, and excluding the one 20 million tCO₂e transaction, the average size of non-retail transactions has been roughly 1 million tCO₂e. This marks an increase compared with the previous years, reflecting an increased frequency of projects over 1.5 million tonnes.

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⁹State and Trends of the Carbon Market 2004", World Bank, June 2004

¹⁰State and Trends of the Carbon Market 2004", World Bank, June 2004

¹¹State and Trends of the Carbon Market 2004", World Bank, June 2004

¹²State and Trends of the Carbon Market 2004", World Bank, June 2004

PROJECT-BASED TRANSACTIONS - PRICES

Structure of Transactions

Most project-based transactions to date (about 95% in 2003-2004) follow a commodity model, whereby the buyer of carbon purchases the emission reductions generated by the project, as it would purchase any other good or service. Only a few transactions follow an investment model where the buyer invests in a project for emission reductions.¹³

Kyoto Pre-Compliance

ERs, which are not for Kyoto compliance, command a price between \$0.37 and \$3.00/tCO₂e (weighted average, by volume, \$1.34). The transactions intended for Kyoto compliance, with risk on the buyer, sell at \$3.00 to \$4.25 (average \$3.85) and transactions with risk on the seller commands a higher value of \$3.00 to \$6.37 (average \$5.52).¹⁴

The prices of JI and CDM transactions do not appear to have increased significantly from 2003, except when the buyer requires delivery of compliance-grade credits. The weighted average price of credits in these transactions has increased from \$4.88 (as reported at the end of 2003) to \$5.52.¹⁵

Total Value of Project-Based Transactions

The estimated total value of contracts for the purchase of project-based transactions is approximately \$330 million in 2003, and \$260 million from January to May 2004. Since its inception, the total value of project-based transactions has been between \$500 million and \$800 million.

A sign of increasing liquidity in the market is that companies are beginning to engage in secondary transactions, selling part of their portfolios to other buyers. Not only does this development attest to increased demand, but it is also an indication that some early movers are beginning to see rewards.

MARKET OUTLOOK

Regulatory Drivers

A major recent regulatory development related to the carbon market is the clarification by the European Parliament of the relationships between their rules and the Kyoto Protocol. As the final decisions are made on the implementation of Kyoto and the relationship of EU and other systems to Kyoto the price and market situation will become clearer.

The second major regulatory development of the past six months is the adoption of new methodologies by the CDM Executive Board. As of June 1, 2004, the EB has published five approved methodologies, and the Methodology Panel has proposed consolidated

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¹³State and Trends of the Carbon Market 2004", World Bank, June 2004

¹⁴State and Trends of the Carbon Market 2004", World Bank, June 2004

¹⁵State and Trends of the Carbon Market 2004", World Bank, June 2004

methodologies for renewable energy and waste management, which, if adopted, would considerably expand the range of projects that can be validated under the Protocol. The EB is also expected to register the first project before the end of the year.

Developments now underway, including clarifications of project rules under Kyoto, are likely to have a significant impact on the carbon market, since regulatory uncertainty has been a serious problem. Approved methodologies are already being used for new projects, and it can be expected that, as the number and scope of approved methodologies increases and are clarified, transaction costs for projects should decline.

Despite the progress on methodologies, ratification of the basic agreement is still in question in that Russia is yet to ratify the Kyoto Protocol. Although there have been reports recently that Russia has reached an agreement with the European Union to ratify, Russia has not yet done so. It is not clear when the decision will be officially made and implemented. Second, climate change policies in Canada and in Japan are still not fully established. These two countries face significant shortfalls of emission reductions to meet their Kyoto targets and have, at least in Japan, high domestic abatement costs.

Market Projections

The market for carbon credits from projects is likely to grow steadily in the next 12 to 18 months. This confidence is shown in the buy orders from Japanese and European companies. In addition, the European governments have signaled their willingness to enter the market, and pledged significant resources to buy emission reductions, either directly or through outside intermediaries.

It is much more difficult to project beyond 18 months due to uncertainties with regulators putting rules into effect that create long term predictability and with effective project financing systems becoming available that are capable of handling large volumes. These are both necessary for a major expansion of the market. The Russian decision on Kyoto, which will determine whether the Protocol is ratified, also remains a major variable affecting this market.

The simplified methodologies adopted by the Executive Board for small-scale CDM projects may reduce the transaction costs for those projects enough to make such projects economically viable. Small-scale CDM projects are likely to have annual emission reductions of less than 50 000 tCO₂e. At present, there is no information on the potential number of small-scale projects, nor is their information on the aggregate emission reductions those projects might achieve.

The project types that have an average size sufficiently large to be economically viable account for most of the CDM potential. These project types include energy efficiency measures in the residential, commercial and institutional sectors; energy efficiency in industry; landfill gas capture and utilization; methane reductions in the oil and gas industry; renewable electricity generation; and forestation and reforestation. Other project types, such as recovery of coal-bed methane and reduction/destruction of non-methane GHGs, appear to be economically viable, but do not represent a large share of the total potential.

At present, energy efficiency projects are under-represented relative to their estimated potential. This suggests the existence of factors, such as high administrative costs or other barriers that are not fully reflected in analyses of the achievable potential for these projects. On the other hand, renewable electricity generation and projects that involve non-CO₂ gases, such as methane, are over-represented. The higher global warming potential values of non-CO₂ projects tend to improve the project economics.

Issues that Affect the Potential Demand for CERs

Total demand by industry for Kyoto units in 2010 is estimated at 200 MtCO₂e + 100 MtCO₂e. The European Union (EU) represents 45 to 60% of the estimated industry demand. Purchases by European industry for compliance use will be governed by rules for exchanging Kyoto units for EU allowances. The draft EU Linking Directive allows CERs and ERUs, but not AAUs, to be exchanged for EU allowances. That would create an estimated industrial market for CERs and ERUs in 2010 of 110 MtCO₂e + 65 MtCO₂e.¹⁶

Several governments have already purchased or announced plans to purchase ERUs or CERs. The announced plans represent an annualized demand of roughly 50 MtCO₂e of CERs and ERUs. Government purchases are projected to represent between 45 and 75% of the total demand for Kyoto units.¹⁷

The minimum demand by industry in Europe (about 45 MtCO₂e) and the planned purchases by governments (about 50 MtCO₂e) yield an annual demand for CERs and ERUs of at least 100 MtCO₂e. The median demand by industry in Europe (110 MtCO₂e) combined with estimated government purchases by the European Union, Norway and Switzerland (about 120 MtCO₂e) yields a potential demand for CERs and ERUs of roughly 230 MtCO₂e in 2010. The maximum demand by government and industry could be double the maximum supply of CERs of about 500 MtCO₂e in 2010.¹⁸

A greenhouse gas trading program in Australia or the United States, at the state or national level, could result in allowing the use of Kyoto units for compliance purposes. Such provisions could allow the use of any Kyoto units or be limited to specific types of units, such as CERs.

The Estimated Market Potential of Project Based Reductions

The estimated market potential of the project based reductions is a demand for CERs in 2010 of 250 MtCO₂e (range 50 to 500 MtCO₂e) at a price of \$11.00 /tCO₂e (range + 50%). Virtually all of the estimates fall into this range for the demand based on a series of assumptions. Moreover, the price is consistent with current prices and forecasts of the market price for Kyoto units in 2010.¹⁹

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¹⁶State and Trends of the Carbon Market 2004", World Bank, June 2004

¹⁷State and Trends of the Carbon Market 2004", World Bank, June 2004

¹⁸State and Trends of the Carbon Market 2004", World Bank, June 2004

¹⁹State and Trends of the Carbon Market 2004", World Bank, June 2004

OVERVIEW OF ANALYSIS OF RAMGEN PRODUCTS AND GREENHOUSE GAS REDUCTION

In Part A of the Report an assessment is made of the specific applications to of the Rampressor to opportunities where the greatest potential to reduce emissions of greenhouse gases exists. The GHG emission reductions are achieved by improving the efficiency of air compression, utilizing the heat generated by compression, and using a gas-fired drive where applicable. This analysis includes targeting on applications where the unique ability of the Rampressor to generate usable heat is economically advantageous. The Rampressor will be the first commercial Ramgen product.

In Part B of the Report an assessment is made of the how specific applications of the first Ramgen product to generate electricity, the Rampressor Turbine, can reduce greenhouse gases. This analysis includes consideration of where the greatest opportunity exists to reduce emissions of greenhouse gases by improving the efficiency of electrical generation and by operating in venues where the use of exhaust heat (combined heat and power) is economically viable with this technology.

In Part C of the Report there is a review of the unique capability of the Ramgen Engine to burn methane in waste gases, which currently are a major contributor to greenhouse gases.

Part D has been added to the Report that describes a project now underway to use the Rampressor Turbine in a hybrid configuration with a fuel cell. An analysis on projected greenhouse gas reductions from this hybrid has not been done but the efficiencies gained from a successful development of this hybrid are very high and the benefits for greenhouse gas reduction would be significant. This Section of the Report also includes reviews of the potential eligible of projects to earn carbon credits, which would add further economic incentives for product applications.

PART A

1. Eligibility of the Rampressor Products to Earn Carbon Credits

The Prototype Carbon Fund identifies five Energy Efficiency Improvements Project types that qualify to earn carbon emission reductions. Project types identified are:

1. Supply side energy efficiency improvements – transmission and distribution
2. Supply side energy efficiency improvements – generation
3. Demand-side energy efficiency programs for specific technologies
4. Energy efficiency and fuel switching measures for industrial facilities
5. Energy efficiency and fuel switching measures for buildings

Energy Efficiency Improvement Projects

Supply side energy efficiency improvements – transmission and distribution

“This category comprises technologies or measures to improve the energy efficiency of electricity or district heating transmission and distribution system by up to the equivalent of 15 GWh per year. Examples include upgrading the voltage on a transmission line, replacing a transformer, and increased insulation of the pipes in a district heating system. The technologies or measures may be applied to existing transmission or distribution systems or be part of an expansion of a transmission or distribution system.

Baseline: For retrofit projects, the energy baseline is the technical losses of energy within the project boundary. In the case of new facilities, the energy baseline is the technical losses of energy within the project boundary calculated using a performance standard for the equipment that would otherwise have been installed.”²⁰

Because this category of project is targeted at the energy efficiency of electricity or heat transmissions and distribution, it does not apply to Rampressor products.

Supply side energy efficiency improvements – generation

“This category comprises technologies or measures to improve efficiency of fossil fuel generating units that supply an electricity or thermal system by reducing energy or fuel consumption by up to the equivalent of 15 GWh per year. Examples include efficiency improvements at power stations and district heating plants and co-generation. The technologies or measures may be applied to existing stations or be part of a new facility.

Baseline: The energy baseline is the technical losses of energy within the project boundary. In the case of retrofit measures, the energy baseline is calculated as the monitored performance of the existing generating unit. In the case of new facilities, the energy baseline is calculated using a standard for the equipment that would otherwise have been installed”²¹

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²⁰ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

²¹ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

This category applies to the Rampressor product when operating in combined compression and gas fired cogeneration configurations but not to the Rampressor that is compressor product powered by an electric motor.

Demand-side energy efficiency programs for specific technologies

“This category comprises programs that encourage the adoption of energy-efficient equipment, lamps, ballasts, refrigerators, motors, fans, air conditioners, appliances, etc. at many sites. These technologies may replace existing equipment or be installed at new sites. The aggregate energy savings by a single project may not exceed the equivalent of 15 GWh per year.

Baseline: If energy displaced is fossil fuel: fuel use multiplied by emission coefficient for the fossil fuel, where fuel use is either existing fuel use or related to technology that would have been implemented otherwise.

If energy displaced is electricity: Energy generation = (quantity x power x operating hours) / (1 - technical T&D losses), power being either the weighted average of replaced devices or the weighted average of devices on the market”²²

The efficiency of the Rampressor would qualify to earn carbon emission-reduction credits under this category.

Energy efficiency and fuel switching measures for industrial facilities

“This category comprises any energy efficiency and fuel switching measure implemented at a single industrial facility. Examples include energy efficiency measures (such as efficient motors), fuel switching measures (such as switching from steam or compressed air to electricity) and efficiency measures for specific industrial processes (such as steel furnaces, paper drying, tobacco curing, etc.). The measures may replace existing equipment or be installed in a new facility. The aggregate energy savings of a single project may not exceed the equivalent of 15 GWh per year.

Baseline: Energy use of the existing equipment that is to be replaced or energy use of the facility that would otherwise be built (both adjusted for T&D losses)”²³

The efficiency of the Rampressor would qualify to earn carbon emission-reduction credits under this category.

Energy efficiency and fuel switching measures for buildings

“This category comprises any energy efficiency and fuel switching measure implemented at a single building, such as a commercial, institutional or residential building, or group of similar buildings, such as a school district or university. Examples include technical energy efficiency measures (such as efficient appliances, better insulation and optimal arrangement of equipment) and fuel switching measures (such as switching from gas to

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²² (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

²³ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

electricity). The technologies may replace existing equipment or be installed in new facilities. The aggregate energy savings of a single project may not exceed the equivalent of 15 GWh per year.

Baseline: Energy use of the existing equipment that is to be replaced or energy use of the facility that would otherwise be built (both adjusted for T&D losses)²⁴

The efficiency of the Rampressor would qualify to earn carbon emission-reduction credits under this category when operating at either a commercial or residential site.

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²⁴ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

2. Applications of Rampressor Product to Reduce Emissions of Greenhouse Gases

This Section of the Report contains 1) Description of the Rampressor; 2) Explanation of the Advantages of the Rampressor; 3) Analysis to identify industries that require both electricity and heat; 4) Recommendations of priority applications for the Rampressor; and, 5) Estimate of greenhouse gas reductions from the extensive use of the Rampressor.

1) Description of the Rampressor;

Ramgen is developing two options for this application.

The first option being developed is a single-stage Ramgen Compressor driven directly by a high-speed permanent magnet electric motor as shown in the following concept drawing.

The compressor unit does not require an intercooler and is completely air-cooled. The motor does require some form of liquid cooling and can be set-up with a self-contained closed loop cooling system. There is no lube-oil or lube-oil system is required. The unit features a variable speed drive to compensate for ambient inlet temperature changes, and a throttle valve to control capacity.

The bhp/100 cfm is estimated to be 19.36, with an isothermal efficiency of 63.1%, which we believe to be competitive with conventional designs in this size range. In addition and as a single-stage unit, the heat of compression is concentrated entirely in the discharge and available for capture and re-use.

The Rampressor, shown above, is a compressor product that is more efficient on a direct power consumption basis than current products. In addition, with the Rampressor, unlike other compressors, the heat generated during compression can be used for heating. The combined heat and compression (CHC) system efficiency for the Rampressor is approximately 60% versus less than 20% system efficiency for conventional designs. The Rampressor will significantly increase the efficiency of the compression process because the temperature of its compressed gas exhaust is 650°F, which is suitable for a variety of combined heat & compression applications (CHC). The capital and maintenance costs of the Rampressor will be a fraction of current compressors, which will add to the economic incentives for using this technology. DOE has cited higher efficiency in compression as a national priority.

Description of the Two General Models of the Rampressor Being Developed.

The first model being developed is a single-stage Ramgen Compressor driven directly by a high-speed permanent magnet electric motor as shown in the following concept drawing.

The compressor unit does not require an intercooler and is completely air-cooled. The motor does require liquid cooling and will be set-up with a self-contained closed loop cooling system. There is no lube-oil or lube-oil system is required. The unit features a variable speed drive to compensate for ambient inlet temperature changes, and a throttle valve to control capacity.

The bhp/100 cfm is estimated to be 19.36, with an isothermal efficiency of 63.1%, which is competitive with conventional designs in this size range. In addition as a single-stage unit, since the heat of compression is at 650°F and is concentrated entirely in the discharge it is available for capture and re-use.

The second model Rampressor being developed is a fuel-fired unit, with the electric motor being replaced with a radial inflow gas turbine rotor, combustor and recuperator. The design is a single-shaft, recuperated gas turbine with a pressure ratio of 7:1, the principal difference being that power is extracted from the turbine cycle in the form of compressed air, rather than electric power.

To accomplish this, the Ramgen single-stage compressor component is “oversized” compared to that of a gas turbine designed for power generation, and the excess air is “bled” off the engine and into the compressed air system as shown below. (Compressor/turbine is not to scale in drawing).

In addition to its relative simplicity, the significant advantage is that the \$/100 scf delivered to the system is approximately 50-60% of the electric equivalent, depending on relative fuel and electric costs. The heat of compression remains fully available for capture and re-use at 575°F, and the exhaust gas from the turbine is available to support a cogeneration opportunity at approximately 600°F, and represent additional energy saving opportunities. The attached tabulation makes the necessary comparison based upon an average electric rate of \$0.08/kWh and an average gas price of \$5.00 per mmBtu.

With full utilization of these two “waste” heat streams down to a nominal 300°F, the corresponding \$/100 scf delivered would be approximately 20% of the baseline electric.

Importance of Increasing Efficiency in Industrial Systems

Improvements in industrial motor systems efficiency offer huge opportunities to invest in enhanced efficiency and profitability of American industry. Industrial motor systems represent the largest single electrical end use in the American economy. In 1994, industrial electric motor systems used in production consumed over 679 billion kWh, or roughly, 23 percent of all electricity sold in the United States. Motors used in industrial space heating, cooling, and ventilation systems used an additional 68 billion kWh, bringing the total industrial motor system energy consumption to 747 billion kWh, or 25 percent of all electricity sales. This is roughly equal to the total electrical sales to the commercial sector in 1994 (795 billion kWh).²⁵

Estimates for potential industrial motor system energy savings in the manufacturing sector, using available technology, range from 11 to 18 percent of current annual usage, or between 62 and 104 billion kWh per year. Potential savings in the non-manufacturing sector industries are estimated at an additional 14 billion kWh per year. These savings are equivalent to the potential energy savings in such major commercial end-uses as indoor lighting. In comparison, all utility sponsored demand-side management programs

²⁵“United States Industrial Electric Motor Systems Market Opportunity Assessment”, DOE OIT, December 1999

produced annual energy savings of 62 billion kWh in 1996.²⁶

The estimated energy savings using current technology would translate into dollar savings of up to \$5.8 billion dollars and carbon emission savings of up to 29.5 million metric tons per year. Motor system energy use is highly concentrated by industry and size of plant, so that 50 percent of the estimated savings would occur in 3,500 manufacturing facilities representing only 1.5 percent of the total manufacturing sector. The financial impact for plants using significant amounts of motor system energy is substantial because these plants statistically operate on profit margins averaging 16 percent of operating revenue.²⁷

Improvements to the major fluid systems of pumps, fans, and air compressors, represents up to 62 percent of currently identified potential motor system energy savings. These savings do not include savings associated with improvement to the efficiency of the motors driving the systems.²⁸

The Rampressor products can increase the already identified improvements to energy efficiency in the industrial and manufacturing sectors with higher efficiency air compression and heat recovery from air compression. The increased efficiencies of the Rampressor products, combined with the lower costs of these products, significantly increasing the prospect that the products will be used in large numbers.

Importance of Breakthrough Compression Technology

The importance of a breakthrough compression technology cannot be overstated. The air and gas compression industry is often referred to as the Fourth Utility. Compressed air is one of the most important utility requirements of industrial manufacturing. Air compression is also a critical element in determining the efficiency of all types of combustion engines, including gas turbines, internal combustion engines and hybrid fuel cells. Gas compression is the key to the transportation of natural gas, air conditioning and refrigeration, and efficient compression of CO₂ is increasingly seen as pivotal to effectively dealing with green house gas emissions.

The Rampressor - Superior Performance Characteristics (operating with electric motor)

Single-Stage Rampressor

The single-stage Rampressor offers a unique combination of high operating efficiency at a high-pressure ratio in sizes up to 700hp. The preliminary performance prediction for a Rampressor is that a single-stage 8.0 inch diameter rotor operating at 60,000 rpm would deliver 1500 cfm at an efficiency of 23.5 bhp/ 100.

This single-stage design would be direct drive, oil-free and air-cooled. The rotor would be close-coupled to a direct drive high-speed motor, and both the rotor and the drive, in a

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²⁶“United States Industrial Electric Motor Systems Market Opportunity Assessment”, DOE OIT, December 1999

²⁷“United States Industrial Electric Motor Systems Market Opportunity Assessment”, DOE OIT, December 1999

²⁸“United States Industrial Electric Motor Systems Market Opportunity Assessment”, DOE OIT, December 1999

simple mechanism, would be supported on magnetic bearings. The conventional intercooler and oil system would be eliminated, and the control system functions would be combined with the drive components to reduce cost. The Rampressor's smaller size and weight would further reduce the overall package costs.

Description of Specific Ramgen Products: Ramgen 1500 cfm Compressor

The Rampressor motor is a variable speed device. The speed will vary to accommodate changes in ambient temperature and a simplified density control will be fitted to the inlet to control compressor output. Shock waves are a function of fluid velocity, not weight flow, as is the case with centrifugal designs. Unlike a centrifugal compressor where the flow is reduced as the pressure ratio increases, the curve shape of the Rampressor is vertical and it is difficult to surge the unit. When surge does occur, it is neither a violent nor an unstable event, and early indications from test data suggest that the operating range will be at least as good as conventional centrifugal designs. The variable speed device will allow the unit to deliver full output and design efficiency, even at hot conditions.

The discharge temperature of the Rampressor is estimated to range between 625-650°F, depending on inlet temperature and discharge pressure setting. This will offer the potential for significant waste heat recovery, without compromising the compressor performance, as is the case with conventional intercooled designs. All of the available heat is concentrated downstream of the compressor and is of sufficient quality to be recovered into process water loops or support the production of 150 psig steam, common to many industrial facilities. The combined compressor and heat recovery would create an even more impressive energy efficiency advantage by providing an opportunity to recover 60% of the electrical input energy in the form of useful heat that is normally rejected to the cooling water in current compressor products.

Two-Stage Rampressor

The two-stage Rampressor will be offered at sizes of 600hp and larger. The preliminary design configuration will be a double-ended, or back-to-back high speed motor driven unit equipped with magnetic bearings. The design will feature equal compression ratios per stage.

The range of powers would include 600 through 2000 hp and possibly beyond as conditions dictate. Preliminary design investigations suggest two 9.5 inch rotors mounted back-to-back on a 600hp motor operating at 32,000 rpm would produce 3000 cfm at 19.4 bhp/100. The combination of rotor diameter and speed can be used to produce a near optimum tip speed on each end of the compressor, maintaining the high efficiency levels.

Competitive Advantages of Ramgen's Compression Technology

The four principal advantages of shock compression when compared to existing compressor technologies are as follows:

- 1) Rampressor can achieve exceptionally high compression efficiency,
- 2) Rampressor is capable of producing very high single-stage compression ratios,

- 3) Rampressor simplicity and size making it less expensive to manufacture, and,
- 4) Rampressor offers the opportunity for significant waste heat recovery.

Current large compressor markets are dominated by centrifugal and axial designs. Centrifugal compressors are typically applied at ratios of 1.8 to 2.8 per stage with adiabatic efficiencies that range from 82 to 85%. Centrifugal designs can be applied up to pressure ratios of 4.0 per stage in 17-4PH stainless steel with adiabatic efficiencies of 82 to 83%, and up to a pressure ratio of 4.5 per stage in titanium with adiabatic efficiencies of 75 to 80%. Because conventional axial compressors can develop pressure ratios of 1.2 to 1.4 per stage, they require 4 to 10 stages to achieve usable industrial pressure ratios. The integrated overall compression efficiency of axial efficiencies is in the range of 82-87%. In contrast, Ramgen's shock compression has the potential to develop compression ratios from 2.0 to 15.0 per stage with an associated adiabatic efficiency of 85-90%.

RAMGEN TARGET MARKETS/APPLICATIONS

Overview of Rampressor Compression Market

Experts estimate that up to 18% of the United States electrical energy consumption is used to drive industrial air compressors. Air compression is also a critical element in the efficiency of all types of combustion engines, as well as hybrid fuel cells. Gas compression is key to the transportation of natural gas through pipelines as well as to air-conditioning and refrigeration processes, and cost effective and efficient CO₂ compression is increasingly seen as a key component of dealing with green house gases.

Ramgen's compression technology can be used across a wide range of current compressor applications as both a product and a component. Ramgen intends to first develop a line of single-stage and two-stage, oil-free, direct-drive air and gas compressor products for use in a wide variety of industrial and process applications. The first single-stage Ramgen offering would cover power levels from 50 to 700hp. The two-stage, intercooled Ramgen design will be offered in sizes 700hp and larger.

Ramgen also plans to offer compression rotor components to original equipment manufacturers as a technology insertion into their multi-stage products. A preliminary evaluation on the limits of the technology indicates that maximum flows could reach 230,000 cfm, which is comparable to the largest centrifugal compressors offered today.

Compressor Product Overview

The industrial air compressor market consists of a number of distinct product offerings that fill a combination of size/application requirements, and it may be divided into four segments differentiated based on size. The *homeowner or DIY market* is typically sold through the "big box outlets" that offer products from fractional sizes to 5 hp.

The *small compressor market* consists of the light industrial and commercial users, the largest of which is the auto shop and gas station aftermarket at 175- psig-discharge pressure. The market is generally thought of as 3-25 hp and is sold through industrial distribution channels or direct to national accounts and OEM's. The US market

represents approximately 150,000 units annually, of which 10,000 might be oil-flooded rotary screw designs. Of these 150,000 units, no more than 10% would be oil-free, sold as dry scroll rotary or Teflon-ring piston designs. This 10-100 cfm oil-free segment has always been a difficult market for the air compressor manufacturers to satisfy technically, and undoubtedly, the lack of product has discouraged market development. The available technologies are small; this limitation is due to clearance effects and related drive mechanism complexities.

The *medium compressor market* covers the 10-350 hp range (nominally 100/125- psig discharge), and is normally thought of as either an oil-free or oil-lubricated market. However, it is only in the 50-350 hp segments that customers have real technology choices. The US market represents approximately 22,000 units annually, of which 8500 fall in the 15-50 hp range and 1500 are in the 5-10 hp range. Oil-flooded rotary screw designs serve the bulk of this market, having effected a near total replacement of the reciprocating product over the last 30 years. Oil flooded screws account for the entire number of screw compressor units sold below 25 hp. An estimated 90-95% of the air-cooled oil-flooded rotary units sold are below 350 hp.

Recent introduction of two-stage high efficiency flooded screws in the 100-500 hp range have been very successful. Oil-free rotary designs account for 15% of the units between 25-350 hp, and 25% of the value of units sold. Both rotary screw and rotary hook & claw (tooth) designs are sold in this size range. Hook & claw designs are sold from 25 to 75 hp (265 cfm), and rotary screw designs are sold from 50 hp up to 1000 hp. There are a few centrifugal offerings for oil-free application, but these designs have not proven to be very effective in competing against the dry screw, and there have been operational problems associated with applying these types of units in small systems. The domestic annual market for medium-sized oil-free compressors is valued at approximately \$110 million.

The *large compressor market* covers products from 350 to 10,000 hp. These products are virtually all 3 or four-stage centrifugal designs, as it has been the only practical design to generate these large flows. Centrifugal compressors are inherently oil-free. There are approximately 1000 of these units sold each year in the US to air separation facilities, large manufacturing plants and the chemical/refinery process industries. The domestic market for this class of sizes is valued at approximately \$240 million. The 350-700 hp range is approximately \$65 million of that total.

Rampressor Priority – Applications Using Both Compression and Heat

The maximum greenhouse gas reductions can be achieved with the Rampressor in applications using both compression and heat. In order to identify the most economically viable applications in this Report consideration is given first to industrial need for compression and then matching that with need for heat.

Review of Industrial Compressor Applications

The US Department of Energy conducted nearly 2,000 interviews with representatives of industrial and manufacturing facilities that identified how electricity is used for motors as part of the overall electrical use of each industry. For the US industries represented in these interviews, compressed air systems account for 10 percent of all electricity and roughly 16 percent of all motor system energy use. This study found that electricity used to drive air compressors is heavily concentrated in three manufacturing groups: chemicals, primary metals, and petroleum manufacturing. Particularly in chemicals and petroleum refining, compressed air and other gases are used as feedstocks or are otherwise delivered directly to the production process, without going through a plant-wide air system. In other industries, such as transportation equipment, air compressors are used almost exclusively to power plant-wide systems. The following chart outlines the information from the study.

SIC	Industry Group	Compressed Air GWh/year	% Oil-Free Compressed Air Use	Oil-Free Compressed Air GWh/Year	Total Motor Systems GWh/year	Compressed Air % Motor System Use	Compressed Air % Electric Use
28	Chemicals and Allied Prod	39,960	25.2%	10,070	144,362	27.7%	20.1%
33	Primary Metal Industries	12,609	N/A	N/A	87,935	14.3%	8.3%
29	Petroleum and Coal Prod	7,930	N/A	N/A	51,938	15.3%	15.9%
37	Transportation Equip	5,519	16.4%	905	29,549	18.7%	14.0%
30	Rubber and Misc. Plastic Prod	4,767	6.3%	300	36,610	13.0%	10.9%
26	Paper and Allied Prod	4,533	11.2%	534	99,594	4.6%	3.7%
36	Electronic and Other Elec. Equip	3,008	13.5%	406	13,243	22.7%	9.1%
20	Food and Kindred Prod	2,898	23.3%	675	37,797	7.7%	4.5%
22	Textile Mill Prod	2,392	13.8%	330	16,750	14.3%	7.2%
24	Lumber and Wood Prod	1,901	0.2%	38	22,946	8.3%	8.7%
34	Fabricated Metal Prod	1,777	N/A	N/A	7,296	24.4%	5.2%
35	Industrial Machinery and Equip	1,172	N/A	N/A	7,378	15.9%	3.6%
38	Instruments and Related Prod	721	N/A	N/A	6,487	11.1%	4.9%
32	Stone, Clay, and Glass Products	566	N/A	N/A	2,231	25.4%	1.6%
25	Furniture and Fixtures	460	N/A	N/A	3,694	12.5%	6.9%
27	Printing and Publishing	437	N/A	N/A	5,961	7.3%	2.5%
23	Apparel and Other Textile Prod	398	N/A	N/A	1,168	34.1%	5.1%
31	Leather and	1	N/A	N/A	491	0.3%	0.2%

	Leather Prod						
20 - 39	Overall Manufacturing	91,050	N/A	N/A	575,428	15.8%	10.0%

(DOE/GO-102001-1197)

The total compressed air use and percentage of compressed air to the total electrical demand for the industry are two factors in identifying priority applications for the Rampressor for greatest possible economic viability and impact on greenhouse gases.

Compressed Air Use by Industry

The horsepower of compressors varies greatly across industries. The Rampressor will initial be available up to 700 HP, but larger models will follow as development continues. A factor in determining the initial priority markets for the Rampressor will be identifying those industries with a high percentage of facilities using compressors up to 700 hp, the range initially supplied by Ramgen. The following table shows the percentage of facilities with air compression demands by hp by industry. This sample is taken from a survey conducted by DOE. The survey represents a sampling of 220 facilities. From this survey, 72% of the facilities are using up to 1,000 hp for air compression and no industry has less then 40% of facilities using more then 1,000 hp of air compression. While the categories of this sample do not fit perfectly to the initial Rampressor sized up to 700 hp, the information is very useful in making estimates and determining which industries are likely to offer the greatest potential opportunity for Ramgen technology in the current size categories. (DOE/GO-102001-1197)

Horsepower of compressors

SIC		Total Horsepower of Air Compressors					
		100	100 – 499	500 – 999	1,000	Unknown	Total
20	Food and Kindred Prod.	5 (25%)	10 (50%)	4 (20%)	1 (5%)	0	20 (95%/100%)
22	Textile Mill Prod.	1 (5%)	11 (55%)	2 (10%)	3 (15%)	3 (15%)	20 (70%/100%)
23	Apparel and Other Textile Prod.	12 (60%)	6 (30%)	0	0	2 (10%)	20 (90%/100%)
26	Paper and Allied Prod.	1 (5%)	10 (50%)	1 (5%)	6 (30%)	2 (10%)	20 (60%/100%)
28	Chemicals and Allied Prod.	3 (15%)	4 (20%)	2 (10%)	3 (15%)	8 (40%)	20 (45%/100%)
29	Petroleum and Coal Prod.	2 (9%)	7 (31%)	1 (4%)	4 (18%)	8 (36%)	22 (44%/100%)
30	Rubber and Misc. Plastics	1 (5%)	13 (65%)	3 (15%)	2 (10%)	1 (5%)	20 (85%/100%)
32	Stone, Clay, and Glass Prod.	1 (5%)	8 (40%)	2 (10%)	5 (25%)	4 (20%)	20 (55%/100%)
33	Primary Metal Ind.	2 (10%)	11 (55%)	3 (15%)	3 (15%)	1 (5%)	20 (80%/100%)
34	Fabricated Metal Prod.	3 (15%)	11 (55%)	3 (15%)	0	3 (15%)	20 (85%/100%)
36	Electronic and Other Electric Equip.	4 (20%)	11 (55%)	2 (10%)	1 (5%)	2 (10%)	20 (85%/100%)
Total		35 (16%)	102 (46%)	23 (10%)	28 (13%)	34 (15%)	222 (72%/100%)

(DOE/GO-102001-1197)

Generic Thermal Process by Industry

The use of waste heat is the enormous advantage that the Rampressor has over other options to reduce greenhouse gases by bringing higher efficiency to the compression process. The 625 to 640 degree F waste heat generated by the Rampressor can be

recovered for a number of different processes. The processes where the recovered heat can be used include:

- Direct Contact Water Heater
- Indirect Liquid Heating
- Direct Heating & Drying
- Indirect Air/Gas Heating
- Refrigeration and Freezing
- Dehumidification
- Exhaust Gas Oxidant (non-boiler)
- Steam

The use of waste heat at 650 degree F temperature from the operation of an air compressor is an application only possible with the Rampressor. For reducing greenhouse gases, industries with compressed air demands that also have processes where waste heat from the compressor can be utilized are the priority applications for the Rampressor.

The following table shows the applicability of each generic thermal process to each manufacturing sector.

Thermal Processes by Industry

Industry	Direct Contact Water Heater	Indirect Liquid Heating	Direct Heating Drying	Indirect Air/Gas Heating	Refrig/Freeze	Dehumidification	Exhaust Gas Oxidant (non-boiler)	Exhaust Gas Oxidant (boiler)
20 – Food	X	X	X	X	X	X	X	X
21 – Tobacco			X		X			X
22 – Textiles	X		X	X				X
23 – Apparel	X							X
24 – Wood	X		X					X
25 – Furniture	X							X
26 – Paper	X	X	X				X	X
27 – Printing	X	X	X		X		X	
28 – Chemical	X	X	X	X	X	X	X	X
29 – Petroleum		X	X		X	X	X	X
30 – Rubber Plastics	X	X	X	X	X			X
31 – Leather	X							X
32 – Stone Clay Glass	X	X	X				X	X
33 – Prim Metals	X	X	X		X	X	X	X
34 – Fab Metals	X	X	X	X			X	
35 – Machinery	X	X	X				X	
36 – Electronics	X							
37 – Transport	X							
38 – Measuring Equipment	X							X
39 – Misc.	X							

(“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001)

Rampressor Savings

The combined value of the electricity saved and carbon credits earned by Rampressor technology operating in the U.S. as a stand-alone, combined heat and compression, and gas-fired oil-free air compressor is estimated to equal nearly \$17 million dollars in the first year of operation, and, after twenty years, to result in an annual value of \$2 billion dollars in the United States. These projections are based on the total horse power of new oil-free air compressor shipments in three horse power size ranges: 50-100 HP, 125-600 HP, and 700-5,000 HP. The gas-fired Rampressor in the single-stage categories of 50-100 HP and 125-600 HP can not be configured in the larger multi-stage 700-5,000 HP Rampressor.

These estimates use 2004 as the base year (this is for purpose of example only as no commercial product will be available in 2004) and assume an annual growth rate of 3% for the oil-free air compressors market. The Rampressor is assumed to 10% of the new market shipments in the initial year, and an additional 10% per year until the Rampressor is shipping 50% of the new oil-free HP demand for the three sizes identified.

Rampressor – US Shipments

Ratings				\$ Savings	Market Share - New HP Shipments				
HP	cfm	+/- %		Rampressor per year/ per unit	0%	10%	50%	50%	
					Market Shipments - HP				
Oil-Free					units x 1,000	2003	2004	2014	2024
Average HP Average CFM	50-100	200-400	16.1%	\$5,970	Shipment - HP - thou	79	81	109	147
					Ramgen Shipment - HP - thou	0	8	55	73
	75			Rampressors operating	Rampressors - thou	0	0.11	1	1
	300				Rampressor - thou	0	0.11	6	14
				\$ value of Eff/CHC /Cogen/ Carbon	CO2 + Eff - \$ per year - thou	0	\$1,622	\$103,824	\$256,289
					CO2 + Eff - \$ -thou	0	\$1,622	\$495,848	\$2,334,058
Average HP Average CFM	125-600	500-2500	6.3%	\$9,950	Shipment - HP - thou	450	464	623	838
					Ramgen Shipment - HP -thou	0	46	312	419
	362.5			Rampressors operating	Rampressors - thou	0	0.13	1	1
	1500				Rampressor -thou	0	0.13	7	17
				\$ value of Eff/CHC/ Cogen/ Carbon	CO2 + Eff - \$ per year - thou	0	\$6,663	\$381,898	\$942,717
					CO2 + Eff - \$ -thou		\$6,663	\$1,849,829	\$8,611,371
Average HP Average CFM	700-5000	3000-25000	2.4%	\$30,956	Shipment - HP -thou	630	649	872	1,173
					Ramgen Shipment - HP -thou	0	65	436	586
	2850			Rampressors operating	Rampressors -thou	0	0.02	0.15	0.21
	14000				Rampressor -thou	0	0.02	1	3
				\$ value of Eff/CHC/ Cogen/ Carbon	CO2 + Eff - \$ per year - thou	0	\$7,683	\$422,850	\$1,043,806
					CO2 + Eff - \$ -thou	0	\$7,683	\$2,059,539	\$9,546,137
CHC - Rampressor (40/60)	60%	\$ value Carbon Credit	\$5.52	TOTAL	Shipment - HP thou	1,159	1,194	1,605	2,157
GT - Cogen	80%	\$ per kWh - CA -EIA, 0804	\$0.07	Rampressors operating	Ramgen Shipment - HP -thou	0	119	802	1,078
kW h per bhp =	0.745	Utilization Factor	80%		Rampressors -thou	0	0.26	2	2
kWh = CO2 lbs per year	1.64	Percent of Units using CHC	50%		Total Rampressor -thou	0	0.26	14	34
Industry Growth Rate	3.0%	1 pound to metric tonne =	0.000454		\$ value of Eff/CHC/ Cogen/ Carbon	CO2 + Eff - per year - thou	0	\$15,967	\$908,571
Operational Hours/ year	8,000	thou = 1,000 M =	1,000		CO2 + Eff -thou	0	\$15,967	\$4,405,216	\$20,491,565
TOTAL	100%	1,000,000	1,000,000		units x 1,000				

Review of Potential Industrial Uses of Rampressor Heat

General Considerations for Using Excess Heat

Until the development of the Ramgen compression product the use of excess heat from compression has not been widely considered since the temperature from the current compressors is only in the 325 degree F range. However, there has been significant work done on the use of heat from smaller engine to generate electricity that is applicable to the use of heat from compression. The following information was developed for distributed generation but can be used for assessing priority applications of heat from the Rampressor.

Integration into a specific manufacturing facility always requires site-specific analyses. However, there are several general considerations pertinent to determining how a specific existing operation is altered (from an engineering perspective) when driven or partially supported with exhaust heat.

1. *Heat transfer rates* to the process media are often reduced when using exhaust (as will be the case with the Rampressor) as opposed to a burner-based operation (e.g., a process furnace). Though the quantity of heat energy available from exhaust is often sufficient to maintain a process operation from an energy balance standpoint, the dynamics (temperature and energy transfer profiles) can be significantly different. This is because high temperature (1500-4000°F), luminous flames induce radiant based heating (rays of energy moving at the speed of light), whereas with exhaust energy moves via convection-conduction only. A common remedy is an increase in the heat transfer area (and therefore the equipment size).
2. *Batch systems* and other operations with non-constant temperature/energy requirements, including fermentation, reactor, and mixing vessels, are often dependent upon dynamic heating systems to accommodate “cold” start-ups, and varying endo/exo-therms and temperature profiles. These systems are not always ideal for the combined heat and compression (CHC) system of the Rampressor.
3. *Auxiliary burners*, or maintaining the existing burner system (depending on its flexibility), can increase the flexibility and therefore applications of the Rampressor CHC system. The primary driver for investing in a CHC unit is the compression output. Constant compression delivery typically results in constant heat delivery (although variations in heat recuperation, fuel feed, etc. can allow for some flexibility). However, for a process such as the batch systems addressed above, there may be a row of three burners heating the vessel. If a “batch” cycle ran one hour, needing all three burners for the first 20 minutes, and one or two thereafter, a steady CHC heat stream could replace the “one or two” burners needed constantly, while the third would remain to maintain the initial per-batch heat requirement.
4. *Controls systems* monitoring and maintaining operations incorporating a CHC system may need to be modified or completely re-engineered. Automated control systems on a typical burner based system will monitor one or more parameters of the process and

then adjust fuel/oxidant feeds to the burner accordingly. If a system is heated via exhaust from CHC, the control system might include a by-pass system or on/off auxiliary burner concepts.

As noted earlier, the processes where the recovered heat can be used include: Direct Contact Water Heater; Indirect Liquid Heating; Direct Heating & Drying; Indirect Air/Gas Heating; Refrigeration and Freezing; Dehumidification; Exhaust Gas Oxidant (non-boiler); and, Steam. A review of each of these follows.

Direct Contact Water Heaters (hot water)

General Description

Process hot water often represents the single largest Btu/hr energy requirement for a manufacturer. Development of highly efficient heat exchange concepts for this purpose has resulted in the 'direct contact water heating' scheme. Fundamentally, by raining water down a 'packed' column, which also is the stack for combustion products (natural gas), near ideal heat transfer is achieved. Exhaust leaves the system cooled to less than 10°F above the cool water inlet, and the water is able to reclaim well above 90% of the exhaust energy.²⁹

Industry Uses

Abundance, availability, safety and experience make hot water a first choice for manufacturing processes requiring:

1. Washing/flushing

- Equipment "clean-down" and sanitizing in food industries (meat, dairy, sugar refining, etc.), and pharmaceutical and "bio" processes.
- Continuous washing operations in raw food preparation (cane/beet sugars, meat, etc.), textiles, wood/paper pulp, removing oils and other excess matter (paint, dust etc.) in metals fabrication and molded plastics industries (auto parts, sheet metal, cans, food/beverage containers, etc.), and in synthetic rubber and fiber manufacturing.
- Flushing process piping and batch equipment (paint blenders, fermentation vessels, etc.), particularly for operations using the same process lines/equipment to produce slightly varying products (paints, candy slurries, pharmaceuticals, etc.).

2. Solvents for raw material preparation, leaching, separations/extractions, and emission control operations. Water is typically chosen when these systems handle general solid inorganic, acids, generally polar fluids, and crystalline salts.

3. Crystallization/fermentation/reaction media for industries including wine/malt-beverage, dairy, pharmaceutical, and inorganic chemicals.

4. Heating jackets for vessels/operations below 230°F; including, chocolate tempering, crystallizers, and storage vessels/mixers containing viscous materials.

Hot water generated from direct contact with natural gas derived combustion exhaust has been approved for food manufacturers including dairy, meat plants, and beverages.

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²⁹"Assessment of Replicable Innovative Industrial Cogeneration Applications," Industrial Centers, Inc., June 2001

Use of Rampressor CHC with Direct Contact Water Heaters

To address heat transfer using the Rampressor for hot water, either more packing media, or extending the height of the column (or both) may be necessary to maintain normal operation (with retrofit systems).

The system has to be designed to take into consideration such factors, as some industrial facilities do not have a constant hot water demand. There are currently two off-the-shelf small industrial cogeneration systems available in the marketplace to generate hot water and components of these may be useful with the Rampressor. Use could also be made of direct contact spray aftercooling to decrease the discharge temperature to the mixture to the equilibrium saturation temperature. The compressed air would then exit to a dryer for use in the compressed air system.

Indirect Heating of Thermal Fluids

General Description

This will work particularly well with the Rampressor. Of necessity, it would apply to those "wet" industries where they handle larger quantities of liquids either as a part of the process (pulp & paper) or as part of the product (food & beverage). This could also be used to drive an Absorption Chiller to provide chilled water.

Many operations, requiring energy delivery to a liquid-phase (and/or fluid) stream, require a physical barrier between the fossil burn (energy release) and the process stream. The barrier reduces heat transfer efficiency, but is often necessary. Traditional systems depend upon heat delivery via heat exchangers, fire-tube schemes (heating coils, or multi-pass fluid heaters) and other methods. Many of these systems use flame induced, radiant based heating to deliver well over 50% of the required energy.³⁰

Industry Uses

Situations involving high (and/or variable) pressure systems, separation/purification operations, multi-phase operations, systems are impeded by possibly reactive/degrading components of combustion but this is not an issue with heat from the Rampressor. Specific operations and their manufacturing environments include:

1. Purification, recovery, and separations.
  - Chemicals/refinery distillation (reboilers, etc.), and flash evaporators (polymer processing, slurry separations/purification, brine treatment, etc.).
2. Pressurized process streams (chemical reactors, etc.).
3. Processes/products sensitive to oxidation, other reaction-driven degradation, and/or general fouling (chemical, food, pharmaceutical processing, etc.).
4. Vat or batch systems maintaining a heated fluid (paint/dye blenders, food deep fryers, refinery bottoms storage and subsequent processing, reactor/fermentation vessels, crystallizers, etc.).
5. Thermal fluid, closed-loop-heating systems for processes, often including those already mentioned, requiring especially high and smoothly controlled temperature

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³⁰"Assessment of Replicable Innovative Industrial Cogeneration Applications," Industrial Centers, Inc., June 2001

profiles.

- Systems requiring high temperatures over large areas such as calcium chloride crystallizers.
- Pipe line tracing.
  - Distillation and reactor feed lines whereby preheating feed components simplifies the energy delivery and/or chemistry complexity of that downstream operation.
  - Heat tracing viscous material (crude, confectionery, polymer melts, etc.) pipelines to reduce electric driven pumping.
- Tool heating (including plastics/rubber extruders, molds), paper mill platens and rollers, metal fabrication equipment, laminate setting, and others.
- General polymer processing. Polymer processing plants may require high temperature (> 400°F) energy delivery to several unit operations because of high “pure” polymer melting points (maintained for extrusion, molding, etc.), and endothermic and/or equilibrium limited reactions (whereby light byproducts, often water, must be continuously evaporated and removed for effective/efficient reactor output). Polyester and Nylon 6 are good examples of major international commodities often utilizing thermal fluids systems throughout their production cycle.

Use of Rampressor CHC with Indirect Heating of Thermal Fluids

The wide variety of thermal “fluid” heating applications mentioned above reflects the broad scope in unit operations, engineering techniques, and process chemistries involved in this category. Three general interconnection (with CHC) systems are:

1. Systems not relying on radiant energy delivery.

Systems currently delivering heat to a process fluid via exhaust energy only (or other forced convection media), either through a series of tubes, vessel/pipe jacket, or compact heat exchanger (shell and tube, plate unit, etc.) can be easily adapted to receive CHC based thermal energy. Because the Rampressor’s thermal output is in the form of hot exhaust, the key concerns would be matching the temperature, gas volume, and pressure parameters to those experienced prior to integration. This may require little or no rebuilding of the process heat-exchange equipment.

2. Systems relying on radiant (flame induced) energy transfer.

In systems requiring a higher heat level than is available through the Rampressor, there may be opportunities to have the Rampressor heat supplement the overall system in a cost effective manner.

3. Closed loop, thermal fluid heating systems.

The previous two interconnection categories represent traditional methods of heat transfer to process fluid streams/systems. The second is more common, but also requires a great deal more case by case analysis, because of the variety of techniques and principles incorporated in radiant heat transfer different from those available from exhaust. Thermal fluid heating systems however, represent a stronger possibility for a more heterogeneous heat delivery, retrofit and/or interface system. From a CHC standpoint, the only concern is maintaining total heat transfer characteristics to the heat transfer fluid on return from the process unit(s).

Direct Heating/Drying

General Description

Direct heating and drying refers to exhaust products mixing directly with the process environment (typically process solids and a forced “air” stream). Because radiation transfer is rapid, typically at high temperature, and ceases upon reaching a boundary (the outer layer of process matter), it is often undesirable and unnecessary. Therefore, natural and forced convection heat transfer engineering may dominate dryer design. There are a wide variety of process dryers, kilns, calciners, ovens, etc. that incorporate an even greater range of combinations in forced convection, radiation, and conduction (through the material) heat transfer principles to satisfy the product requirements. In all cases, however, the heat energy supplied to a system must perform the following four tasks:

1. Heat the dryer feed to the “light” component’s vaporization temperature.
2. Vaporize and/or free the liquid/byproducts above the solids’ surface.
3. Heat the solids to the final desired temperature, and for the desired duration of time.
4. Heat the vapor to the final desired temperature.³¹

Industry Uses

Numerous factors, including production throughput, local steam, natural gas and electricity prices, emissions restrictions, and equipment cost considerations, often result in similar solids being dried in very different ways. However, common direct drying/heating operations and their typical product/process applications include:

1. Bringing variable water-weight percent feeds to a desired initial processing concentration.
  - Mined raw materials and/or prepared mixes fed to cement, gypsum, ceramics, and lime processes require crushing, sizing, and drying. Rotary dryers, impact dryers, drum dryers, and others are used to handle large volume, variable composition slurries. Water removal to organize/homogenize process streams for inorganic chemicals manufacture is also common.
2. More complete drying of slurries containing finer solids within certain size/weight specifications is carried out using spray dryers, thin-film dryers, and drum dryers.
  - Within the Stone, Clay, Glass and Cement manufacturing sector (SIC 32), fine dry powders are desirable for handling, packing, and/or to produce a more consistent product. Specific products include kaolin clay, fluid cracking catalysts and ceramics that may use this step to introduce property enriching additives/binders to the material.
  - Emulsion PVC and PVP polymer processes often employ spray drying to rapidly remove water without degrading product.
  - Milk/dairy powders.
  - Organic and inorganic dry soaps, detergents, dyes and pigments.
3. Pre-heating/drying materials.
  - Metals fabrication and/or scrap metal industries use direct heat to remove volatile impurities (oils, plastics, paints, etc.) and/or to reduce energy demand of central furnace operations.
  - Large kilns, calciners, and ovens (primarily in SIC 32) also benefit from

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³¹“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001

- preheated feeds, often containing preheat sections as part of the primary unit (tunnel kilns, etc.).
- Coke processes may preheat coal feeds to reduce moisture content.
  - Glass and mineral wool industries utilize many preheat techniques to reduce energy demands or increase throughput on central furnaces systems.
4. Drying and heating meant to relieve chemically bound light components and/or otherwise modify solid structure. Rotary kilns, shaft kilns, kettle calciners, flash calciners, brick ovens/houses, tunnel kilns, regenerative kilns, and others are included in this grouping.
- Kilns and ovens used for bricks, ceramics, etc. where residence times in hot and dry conditions may last hours to days to obtain desired final qualities in appearance and structure.
  - Kilns and calciners used to produce/process gypsum, plasters, cements, limestone, etc. where energy not only thoroughly removes any remaining water, but also frees intimate impurities, and forces various reactions often resulting in the release of carbon and sulfur oxides. Along with those operations in SIC 32, both the pulp & paper and beet sugar industries use these lime kiln technologies.
5. Drying to remove water (and/or other solvents/chemicals) added, left, or produced during processing.
- Starch, stalk and husk dryers, and fruit peel and feed dryers, used in beet and cane sugar manufacturing, grain mill products, and other SIC 20 manufacturing sectors.
  - Convection dryers in textile manufacturing.
  - Veneer and other lumber/wood-furniture dryers.
  - Pulp dryers, coated and tissue paper dryers in SIC 26.
  - Dryers including conveyor and tray dryers used in non-cellulosic and cellulosic fibers (rayon, acrylics, etc.) processing, polymer rubbers manufacture, for pharmaceuticals, and latex.
6. Granulators, fluidized bed systems, rotary dryers, and tower dryers often used for producing finished grains, sugar, and fertilizer.

Use of Rampressor CHC with Direct Heating/Drying

Many kilns and calciners depend on high temperature (1000-2000°F) exhaust and radiant heating sections that could not be supplied by Rampressor exhaust alone. However, preheating operations can take advantage of the Rampressor heat. Sites with successful existing preheating may see additional turndown (on the primary unit's fuel feed) without high retrofit costs if the system can handle an extra volume of exhaust and assuming temperature conditions are similar to the existing preheated hot gas feed. Drying operations at facilities without processing furnaces, (e.g., calciners) could completely supplement a non-radiant based dryer.

Indirect Air/Gas Heating

General Description

Air heaters or inert gas heaters are commonly considered when products, process operations, or the facility environment is potentially compromised by using direct drying/heating systems. Because issues including plant layout, local regulations, and fuel

type affect these considerations, many of these processes are served by both indirect and direct heating.³²

Industry Uses

Two general processing categories are:

1. Food products cooking, baking, and drying.
  - Roasters used in coffee and cocoa processing.
  - Baking ovens used for breads, cakes, etc.
  - Toasting and drying systems for cereals.
2. Finish drying and curing systems.
  - Dryers following painting and or final cleaning operations in furniture and metals fabrication industries (transportation & industrial equipment, beverage cans, etc.).
  - Dryers used in finishing periodicals and newspaper production processes.

Use of Rampressor CHC with Indirect Air/Gas Heating

Air heaters are often industrial versions of fired furnaces used in HVAC systems. Although many of the operations mentioned above require only modest heat (200-600°F), the heater itself may have radiation induced “hot side” temperatures above 1500°F. A new type of heat exchanger may be needed for some applications.

Refrigeration/Freezing (absorption cooling)

General Description

Refrigeration/freezing refers to a direct process end use in which energy is used to lower the temperature of substances involved in the manufacturing process. Conventional equipment includes industrial chillers and absorption cooling equipment.³³

Industry Uses

Major applications of industrial cooling include:

- Refrigerated storage of unfrozen foods,
- Frozen foods,
- Refrigeration to change the chemical structure of food,
- Freeze drying,
- Industrial process air conditioning, and
- Refrigeration in the petroleum and chemicals industries (reaction heat removal, gas separations, condensation of gases, separations, solidifications, humidity control, etc.).

Use of Rampressor CHC with Refrigeration/Freezing (absorption cooling)

Absorption cooling systems require a source of heat. For an ammonia-water cooling system, the heat is required to separate the water and ammonia. In conventional absorption systems, this heat is supplied by steam heat exchangers, an electrical heater or a gas fired heater. The Rampressor can be the source of heat with a heat exchanger. The

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³²“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001

³³“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001

heating gases may have to be mixed with air or other gases to maintain desired heating gas temperature.

Dehumidification

General Description

Desiccant-based dehumidification systems are used extensively for removing moisture from air or gases in many industrial applications. Some typical industries where such systems are used include chemical, pharmaceutical, food, semi-conductor manufacturing, and vacuum processing. These systems are also used for climate control applications in commercial buildings. Operation of these systems includes a regeneration step where hot air (or other gases) is used to remove moisture from saturated desiccant media.³⁴

Industry Uses

Major applications of dehumidification in the manufacturing sector include:

- Pharmaceutical processing,
- Candy coating,
- Storage and packing,
- Conveying of hygroscopic powders,
- Composite manufacturing,
- Semiconductor manufacturing,
- Printing operations,
- Corrosion prevention,
- Molding operations, and
- Drying operations.

Use of Rampressor CHC with Dehumidification

The clean Rampressor exhaust can be mixed with ambient air to raise the temperature to the desired value. Currently, a variety of heating methods and media are used for supplying hot regenerative air. The heating methods include heating by electricity, steam, or a fuel (usually gas) fired burner. Application of such a scheme may require redesign of the regenerative air system for a retrofit application. For a newer application, such changes can be accounted for during the design phase of the project.

Steam (including boiler systems)

General Description

The temperature of the Rampressor exhaust is high enough to make steam. However, the needs for steam in industrial sites are generally for larger volumes than can be provided by the Rampressor. Additional work has to be done to determine the number of viable steam applications.³⁵

Exhaust Gas as an Oxidant

General Description

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³⁴“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001

³⁵“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001

The Rampressor in the format in which it is used with a Ramgen Turbine will produce exhaust gas, which can be used as an Oxidant. Combustion reactions are highly exothermic. However, their reactants (fuel and oxidant) continuously absorb considerable energy to reach proper combustion temperatures. Exhaust gases from a prime mover, (particularly from a gas turbine due to high oxygen content) provide an excellent preheated oxidant. These gases can be considered as an oxidant source for combustion of fossil fuels used in most heating applications including steam generators or boilers.³⁶

Industry Uses

Applications for using exhaust gases as an oxidant include:

- Central boiler systems,
- Waste VOC incineration systems,
- Kilns,
- Calciners,
- Large ovens,
- Large heat treating operations,
- Large furnaces,
- Forging operations,
- Tempering operations,
- Annealing operations, and
- Cupolas.

Use of Rampressor in Combination with Rampressor Turbine with Exhaust Gas as an Oxidant

Many engineering techniques addressing the principle of preheating the combustion reactant feed (especially the oxidant, because its volume generally dominates the reactant mixture) are in practice. Three categories represent a majority of these techniques:

1. Use of the stack exhaust to indirectly (e.g., with a shell and tube exchanger) heat the air/oxidant feed line.
2. Burner tip techniques that often incorporate ceramics to maintain the final mixing chamber at extremely high temperature, thereby heating the reactants immediately prior to ignition.
3. Using high temperature, high oxygen content, waste-heat streams as a combustion reactant/oxidant (as the DG cogeneration system would offer).

In general, the cogeneration based oxidant system is highly competitive for these options when:

1. The process operation is operated such that its own exhaust is either low in temperature or low in excess oxygen.
2. The process operation uses coal (or other fuels releasing soot and sulfur in the exhaust) as a fuel. In such cases cogeneration offers both a relatively clean preheated feed (so as not to foul the burner equipment) and also reduces the amount of sulfur and particulate released (by reducing the amount of coal needed).

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³⁶“Assessment of Replicable Innovative Industrial Cogeneration Applications,” Industrial Centers, Inc., June 2001

If a system's burner was initially designed for low temperature air feeds, more heat durable components may be needed to handle a hot oxidant. The difference in oxygen content also needs careful consideration when designing the combustion system.

PART B

1. Eligibility of the Ramgen Turbine and Ramgen Fuel Cell Hybrid to Earn Carbon Credits

The Prototype Carbon Fund identifies five Energy Efficiency Improvements Project types that qualify to earn carbon emission reductions. Project types identified are:

1. Supply side energy efficiency improvements – transmission and distribution
2. Supply side energy efficiency improvements – generation
3. Demand-side energy efficiency programs for specific technologies
4. Energy efficiency and fuel switching measures for industrial facilities
5. Energy efficiency and fuel switching measures for buildings

Energy Efficiency Improvement Projects

Supply side energy efficiency improvements – transmission and distribution

Because this category of project is targeted at the energy efficiency of electricity or heat transmissions and distribution, it does not apply to Rampressor Turbine.

Supply side energy efficiency improvements – generation

“This category comprises technologies or measures to improve efficiency of fossil fuel generating units that supply an electricity or thermal system by reducing energy or fuel consumption by up to the equivalent of 15 GWh per year. Examples include efficiency improvements at power stations and district heating plants and co-generation. The technologies or measures may be applied to existing stations or be part of a new facility.

Baseline: The energy baseline is the technical losses of energy within the project boundary. In the case of retrofit measures, the energy baseline is calculated as the monitored performance of the existing generating unit. In the case of new facilities, the energy baseline is calculated using a standard for the equipment that would otherwise have been installed”³⁷

This category applies to the Rampressor Turbine when operating in combined heat and power configurations and/or at power stations and district heating facilities.

Demand-side energy efficiency programs for specific technologies

“This category comprises programs that encourage the adoption of energy-efficient equipment, lamps, ballasts, refrigerators, motors, fans, air conditioners, appliances, etc. at many sites. These technologies may replace existing equipment or be installed at new sites. The aggregate energy savings by a single project may not exceed the equivalent of 15 GWh per year.

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³⁷ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

Baseline: If energy displaced is fossil fuel: fuel use multiplied by emission coefficient for the fossil fuel, where fuel use is either existing fuel use or related to technology that would have been implemented otherwise.

If energy displaced is electricity: Energy generation = (quantity x power x operating hours) / (1 - technical T&D losses), power being either the weighted average of replaced devices or the weighted average of devices on the market”³⁸

The efficiency of the Rampressor Turbine would qualify to earn carbon emission-reduction credits under this category.

Energy efficiency and fuel switching measures for industrial facilities

“This category comprises any energy efficiency and fuel switching measure implemented at a single industrial facility. Examples include energy efficiency measures (such as efficient motors), fuel switching measures (such as switching from steam or compressed air to electricity) and efficiency measures for specific industrial processes (such as steel furnaces, paper drying, tobacco curing, etc.). The measures may replace existing equipment or be installed in a new facility. The aggregate energy savings of a single project may not exceed the equivalent of 15 GWh per year.

Baseline: Energy use of the existing equipment that is to be replaced or energy use of the facility that would otherwise be built (both adjusted for T&D losses)”³⁹

The efficiency of the Rampressor Turbine would qualify to earn carbon emission-reduction credits under this category.

Energy efficiency and fuel switching measures for buildings

“This category comprises any energy efficiency and fuel switching measure implemented at a single building, such as a commercial, institutional or residential building, or group of similar buildings, such as a school district or university. Examples include technical energy efficiency measures (such as efficient appliances, better insulation and optimal arrangement of equipment) and fuel switching measures (such as switching from gas to electricity). The technologies may replace existing equipment or be installed in new facilities. The aggregate energy savings of a single project may not exceed the equivalent of 15 GWh per year.

Baseline: Energy use of the existing equipment that is to be replaced or energy use of the facility that would otherwise be built (both adjusted for T&D losses)”⁴⁰

The efficiency of the Rampressor Turbine would qualify to earn carbon emission-reduction credits under this category when operating at either a commercial or residential site.

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³⁸ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

³⁹ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

⁴⁰ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

2. Priority Applications of Rampressor Turbine to Reducing Greenhouse Gas Emissions

This Section of the Report contains: 1) Description of the Rampressor Turbine; 2) Explanation of the Advantages of the Rampressor Turbine; 3) Analysis to identify industries that require both electricity and heat; 4) Recommendations of priority applications for the Rampressor Turbine; and, 5) Estimate of greenhouse gas reductions from the extensive use of the Rampressor Turbine.

Description of Ramgen Product

The Rampressor Turbine is a small engine (250 kW to 5 MW) for electricity generation: The Rampressor Turbine engine combines the compression efficiency of the Rampressor with advances in small recuperated turbines. Current turbines in this size are capable of reaching net efficiencies in the 28% to 31% range and the best efforts utilizing today's technologies is not expected to exceed efficiency of 34-35%. The Recuperated Ramgen Turbine can approach 40% without the development challenge posed by the need for advanced ceramic components. It is projected in this report that the combination of this high efficiency, low capital cost, and reliability will stimulate large-scale commercial Distributed Generation and Combined Heat and Power. This is of significant national importance at a time when the electrical grid use is at maximum, there is opposition to large scale new power plants near urban centers and security is a priority. A 400 kW engine can be offered for sale in 2008.

Rampressor Turbine

Gas Turbine Market Overview: Advantages of Rampressor Turbine

The Ramgen advances in compression technology can be incorporated into various gas turbine designs as a component to increase efficiency. Ramgen may design and manufacture its own mini-turbine, either as a stand-alone system or as part of a hybrid fuel cell configuration, as a means to insure the technology reaches its full commercial potential. In this report, the 100-250kW sizes are classified as the “microturbine” market and centrifugal compressors in the sizes between 250kW to 2.0MW, where recuperated cycles are being applied, are classified as “mini-turbines”.

The Advantages that the Breakthrough Ramgen Technology Brings to Small Engines and Distributed Generation

There is general agreement in many DOE supported studies that Distributed Generation, with Combined Heat and Power, would: 1) dramatically reduce fossil fuel use and air pollutant emissions; 2) improve power quality, efficiency, reliability and return on investment; and 3) enhance energy security.

Despite these projected benefits, the growth of DG, beyond the considerable onsite generation that already exists in U.S. industries, has not been as rapid as some had expected. The dynamics that have slowed the applications of Distributed Generation include regulatory uncertainty, volatile and rising natural gas prices and the continued resistance to on-site generation. An additional critical factor in the assessment of some experts is that in order for Distributed Generation to come into widespread use a highly

efficient, low cost and reliable small engine with low emissions has to be commercially available.

In order for a “tipping point” to be reached and for there to be widespread increased use of DG in the assessment of many there has to be a small low emissions engine that is competitive with grid power without factoring in the value of the use of the heat. When a reliable small engine can compete with grid power on cost per kW without considering the heat then in the assessment of some experts DG will be used much more extensively. In this report, the assumption is made that large scale additional DG has to be driven by competitively priced electricity. As additional DG begins to be applied widely then the use of the heat will follow naturally in those applications where heat is of value. The only small engine with this potential that is now approaching a commercial model of which this Report is aware is the Ramgen Turbine.

There has been a series of problems with the engine developers meeting their claimed efficiency and selling price targets. Today’s microturbine is 28% LHV net electric efficiency and it appears that, as currently designed, the best these units can achieve is 34%. The efficiency limits their practical application to CHP projects, and as a result limits the number of units sold.

The basic problem is that these recuperated simple cycle designs cannot use the full turbine-rotor inlet temperature (TRIT) potential because of material cost vs. life limitations on the recuperator materials. Turbine rotor inlet and exhaust gas temperatures are physically related through the turbine efficiency and expansion-ratio design values. In addition, expansion ratio is more or less the same as compressor pressure ratio.

A recuperator is a high temperature heat exchanger that serves to re-cycle the waste-heat from the turbine exhaust back to the cycle before fuel addition. In current recuperated turbines, the recuperator inlet temperature is at or very near the metallurgical limit of the 347 stainless steel material, the generally preferred material. Alloys such as Inconel 625 have higher temperature capabilities but cost approximately three times more per pound than the 347 stainless steel. A switch to such alloys will increase the overall cost of the turbine that could negate the value of any potential efficiency gains.

An approach to avoid high recuperator material costs is to decrease the turbine exhaust temperatures (EGT) by increasing the turbine pressure ratio. A Rampressor can be inserted in recuperated micro-turbine engine configurations for greater compression efficiency, higher specific power, and lower overall manufacturing costs.

One example is a current industrial recuperated turbine configuration with limited TRIT to maintain the recuperated inlet temperature for 347 SS. With a 90% effective recuperator and standard assumptions on pressure drop of 1.5% on the air side and 3.5% on the gas side, the cycle efficiency optimizes at a pressure ratio of 3.0:1. To mitigate cost impacts, these machines are commonly designed at 4.0:1 or slightly higher, reducing component size and cost. Built into these optimizations are assumptions on compressor and turbine efficiencies, at 83% and 85% respectively.

Uncooled metallic turbine rotors are commonly applied at TRIT between 950°C (1742°F) and 983°C (1800°F). The recuperator cost goals, however, compel the use of stainless steel alloys, but the 40,000 hour life in service objective limits the EGT between 625°C (1157°F) and 640°C (1185°F). Unfortunately, this necessitates limiting the TRIT and therefore the turbine efficiency potential. With current technology, TRIT is limited to 893°C (1638°F) which limits the turbine efficiency and results in a net electric efficiency of 34% and an uncompetitive manufacturing cost of \$650 per kW.

The Rampressor could improve compressor adiabatic stage efficiency to 85% to 90%, and shift the optimum pressure ratio to 5.0 - 5.7, allowing full use of the 983°C (1800°F) TRIT turbine rotor capability. The cycle efficiency would be 39% LHV net electric efficiency, without introducing the risk and cost of ceramic nozzles and turbine rotors. The increase in pressure ratio and TRIT results in a 48% increase in specific power and a reduction in manufacturing cost of 33% to \$439 per kW. The increase in pressure ratio also decreases the manufacturing cost of the recuperator per kW.

This level of improved efficiency and deduction in manufacturing costs will greatly enhance the potential to meet the DOE advanced micro-turbine program goals, as well as allow broader participation in the “power only” markets where the opportunity is several times larger than for CHP alone.

An increase in pressure ratio to 6.3 with a 90% efficient Rampressor, and a TRIT of 1205°C (2200°F) results in a net electric efficiency of 46% and an additional increase in specific power of 56%. The cost of the Inconel 625 recuperator is more than offset by the increased specific power. The overall result is an additional reduction in overall turbine manufacturing cost of 31% to \$303 per kW, which is more than enough to support a viable commercial operation selling complete machines at \$500 per kW.

Despite the development advances with ceramics, a number of challenges remain before monolithic and composite designs will exhibit the long lifetimes required for production turbine applications. If ceramics were ever to achieve their promised potential combining ceramics with the Rampressor would result in the net electric efficiencies, approaching a full 15% improvement above the DOE advanced micro-turbine program goal. This is significantly better than reciprocating engines in this size class and avoids the emissions problems of those engines.

Another major challenge faced by CHP is the lack of integrated systems. Finding the optimal CHP components that, when integrated, can meet the wide range of facility heating, cooling, and the electric loads are left up to the plant management and their supporting staff. Establishing the product base and a sufficient volume of installations to overcome this problem is an important factor for building the CHP market.

Priority Application of Rampressor Turbine: Industrial Process Heating Systems

The objective of this report is to identify the priority applications for greenhouse gas reduction marketing opportunities with the Ramgen technology. The priority marketing

opportunity for reducing greenhouse gases with the Rampressor Turbine is in industrial processes that can use the heat from small engines.

This section of the Report on Combined Heat and Power in the Industrial Sector draws extensively from several DOE supported studies but each has been modified for purposes of the Report. While none of these studies included the Ramgen technology, as it is not yet commercially available, the information in the studies is useful in identifying priority uses for the Rampressor Turbine for reducing greenhouse gases.

The “Assessment of On-Site Power Opportunities in the Industrial Sector” produced by the DOE Office of Industrial Technology notes that currently a major source of energy for these industrial processes comes from on-site combustion of petroleum products and natural gas. These often outdated and inefficient combustion systems consume over 15% of the total U.S. energy produced. As companies assess upgrading their systems they balance competing priorities on energy efficiency, operating and capital costs, and reliability with compliance with emissions regulations. Reliability is essential as heating furnaces and steam boilers are required to run for long periods with very few interruptions. Any disturbances and/or unscheduled shutdowns are extremely costly to a plant that can take several weeks to bring on-line.

The ability of the combustion system to respond to sudden fluctuations in fuel composition, large scale internal flow patterns, and plant load cycle is critical. This reliability, along with reduced emissions, is why the Advanced Vortex Combustion (AVC) technology being developed by Ramgen in conjunction with NETL is so important. AVC has the promise of bringing greater flame stability and lower emissions to industrial combustion systems, including the Rampressor Turbine, with resulting greater plant reliability and cost savings.

Background on Use of Combined Heat and Power in Industrial Systems

The “Cooling, Heating, and Power for Industry: A Market Assessment” produced by the DOE EERE outlines how Combined Heat and Power (CHP) has the potential to dramatically reduce industrial sector carbon and air pollutant emissions and increase source energy efficiency. Industrial applications of CHP have been used for decades, producing electricity and byproduct thermal energy onsite, and converting 80 percent or more of the input fuel into useable energy. Typically, CHP systems operate by generating hot water or steam from the recovered waste heat and using it for process heating, but it also can be directed to an absorption chiller where it can provide process or space cooling.

The “Cooling, Heating, and Power for Industry: A Market Assessment” produced by the DOE EERE further discusses thermally-driven cooling technologies that are being demonstrated that can potentially utilize the CHP heat output effectively for uses typically reserved for high-value electricity. CHP with cooling has potential applications that could not be economically served by CHP alone, creating an additional use of the byproduct thermal energy when heating loads are minimal. These new technologies,

particularly when combined with the Rampressor Turbine, make it possible for CHP to be a viable means of decreasing carbon and air pollutant emissions while improving energy efficiency in the industrial sector.

In today's marketplace, there are a variety of technology options for cooling, heating, and power. Absorption chillers are available that can be installed with a CHP system to utilize the heat output to produce process cooling. Engine-driven chillers are considered cooling, heating and power applications since they provide thermal output, in the form of process or space cooling, and displace electricity that typically would be purchased to provide this cooling. Desiccant technology can also be used in conjunction with CHP waste heat to provide cooling.

The overwhelming majority of CHP systems currently installed in the U.S. are used for industrial applications, with increasing usage by commercial and institutional building owners and district energy systems worldwide. There is a general understanding that a 2-4 year payback is required for industrial facilities to purchase equipment that will reduce their energy costs. If a carbon credit market is established in the U.S., revenue from that source may be a significant economic factor in the viability of many projects. The higher efficiencies and lower costs of the Rampressor Turbine will also be a positive factor in the economic calculations.

Market Potential by Region and Industry

A DOE supported study outlines how industrial CHP potential tends to be strongest in areas with relatively high electricity prices and concentrations of industrial sites. Current CHP potential is concentrated in the Pacific (led by sites in the Southern California Edison and PG&E service areas), East North Central (led by sites in the ComEd and Detroit Edison service areas), Mid Atlantic (led by sites in the Niagara Mohawk, PECO Energy, and PSE&G service areas), and South Atlantic (led by sites in the Georgia Power, Carolina Power & Light, and Duke Energy service areas) regions.

If CHP technologies improve as projected by DOE studies and as demonstrated by the development of the Ramgen technology, several regions of the U.S. stand to benefit as CHP potential grows regionally. This growth would occur mostly in the South Atlantic (led again by sites in the Georgia Power, Carolina Power & Light, and Duke Energy service areas) and East North Central (led again by sites in ComEd and Detroit Edison's service area), with some notable expansion in the West South Central region (led by sites in the Houston Lighting and Power and TU Electric service areas). Economically viable CHP projects are expected to expand as the technologies improve their cost and performance, thus not requiring high retail electricity prices for the economics to be favorable.

The "Steam System Opportunity Assessment for the Pulp and Paper, Chemical, Manufacturing, and Petroleum Refining Industries" produced by the DOE EERE focuses on the CHP potential led by Chemicals, Primary Metals, Paper, and Food. Together, these four industries represent over half of the market potential as these industries account for over half of manufacturing electricity, boiler energy, and total energy

consumption. They are therefore important in determining the priority applications for the Rampressor Turbine.

High thermal-to-electric ratios are often cited as an indicator of strong CHP potential. In particular, the ratio of boiler energy to electrical energy is important, since the thermal output from CHP systems supplant boiler thermal output (steam or hot water). Some applications use the output of a turbine directly in a process to supplant direct process heating (such as using turbine exhaust to heat a dryer for clay products), but these are generally thought of as niche applications. Three industries, Food, Paper, and Chemicals offer some of the strongest boiler to electric energy ratios, calculated at 2.8, 3.8, and 2.4 respectively. These are considerably higher than the manufacturing average of 1.4.

While the Primary Metals industry offers a very low boiler to electric energy ratio (0.3), it accounts for a large potential (4.4 GW) due to baseload power generation with large combined cycle units as well as some CHP with large turbines. In these applications, the cost to generate is very low, and competitive with the grid even with low thermal utilization.

Summary of Industrial Sites Analyzed

SIC/Industry	All Industry		Sites Analyzed			
	Annual MWhs Consumed	Number of Sites	Annual MWhs Consumed	Share of All Industry MWhs	Number of Sites	Share of All Industry Sites
20 Food	72,922,480	21,132	71,781,296	98.4%	12,509	59%
21 Tobacco	2,116,506	140	2,113,080	99.8%	99	71%
22 Textiles	41,006,540	6,176	40,696,588	99.2%	4,110	67%
23 Apparel	10,061,381	23,779	8,704,864	86.5%	6,934	29%
24 Lumber	28,851,582	37,133	26,913,270	93.3%	8,925	24%
25 Furniture	8,624,855	12,271	8,046,328	93.3%	4,736	39%
26 Paper	66,931,396	6,535	66,594,600	99.5%	4,715	72%
27 Printing	24,182,360	62,453	19,401,482	80.2%	11,160	18%
28 Chemicals	153,730,720	12,364	153,059,552	99.6%	7,308	59%
29 Petroleum	44,179,784	2,143	44,005,792	99.6%	1,355	63%
30 Rubber/Plastics	57,901,232	16,794	56,972,436	98.4%	11,691	70%
31 Leather	1,162,735	1,854	1,073,904	92.4%	588	32%
32 Stone/Clay/Glass	40,012,440	16,557	38,631,760	96.5%	6,544	40%
33 Primary Metals	147,990,672	6,614	147,710,928	99.8%	5,131	78%
34 Fabricated Metals	59,505,096	38,582	57,278,620	96.3%	21,632	56%
35 Machinery	62,882,532	56,620	58,717,032	93.4%	20,898	37%
36 Electrical Equip.	61,826,456	17,329	60,920,272	98.5%	9,701	56%
37 Transportation Equip.	65,833,432	12,659	65,239,504	99.1%	6,192	49%
38 Instruments	21,952,144	11,929	21,316,654	97.1%	5,743	48%
39 Misc. Mfg.	6,787,301	18,373	5,796,095	85.4%	3,556	19%
Admin/Auxiliary	21,325,939	11,623	N/A	N/A	N/A	N/A
Totals	999,787,583	393,060	954,974,056	95.5%	153,527	39%

(U.S. Department of Commerce, 1996 County Business Patterns, and U. S. Department of Energy, 1998 Manufacturing Energy Consumption Survey)

The CHP Cooling Market

The CHP cooling market potential is led by the Electronics industry, followed by Chemicals, Petroleum, and Food. As shown below, these four industries account for over two-thirds of the current cooling capacity as well as the potential for CHP-driven absorbers. The Electronics and Petroleum industries have relatively low estimated installed bases of cooling. However, they have good potential for CHP cooling due to favorable CHP economics. Cooling is a leading viable use of the CHP waste heat in selected industries. Cooling, for example, constitutes over 95 % of the CHP market potential in the Petroleum sector, and is almost 50 % in the Electronics industry (compared with only 15 % for all manufacturing). Comparing the potential CHP cooling tons with the estimated installed cooling capacity, these two industries each show that the potential CHP cooling is a third or more of the installed cooling base, compared with Food and Chemicals, where the potential is a much smaller share of the installed cooling capacity.

Comparison of Estimated Installed Cooling Capacity vs. CHP Cooling Potential

	Cooling Energy Consumed (Trillion BTU)							
	Electric	Gas	Electric	Gas	Total			
Food (20)	48	1	3,300	13	3,313	106	3%	12%
Chemicals (28)	32	11	2,200	143	2,343	182	8%	79%
Petroleum (29)	6	3	400	39	439	143	33%	27%
Electronics (36)	7	0.3	500	4	504	191	38%	2%
All Mfg	138	21	9,600	273	9,873	941	10%	29%

(EIA Manufacturing Energy Consumption Survey (MECS) 1994)

Analysis of Nine Industrial Groups

Another DOE study has analyzed the potential for on-site power generation in the U.S. industrial sector. This study focuses its emphasis on DOE *Industries of the Future* (IOFs). Total purchases for electricity and steam for these IOFs are in excess of \$27 billion annually. Energy related costs are very significant for these industries. The nine industrial groups include:

1. Agriculture
2. Forest Products
  - Lumber and Wood Products
  - Paper and Allied Products
3. Mining
4. Glass
5. Petroleum

6. Chemicals
7. Metals
  - Steel
  - Aluminum
  - Metal Casting

Although not currently part of the Industries of the Future program, the food industry was included in this DOE study because of its close relationship to the agricultural industry and its success with on-site power generation.

Existing On-Site Generation in the Industrial Sector

On-site power generation is not a new concept for the U.S. industrial sector. Historically, the on-site power market was driven by the availability of waste fuels, locally high retail electricity prices, and attractive wholesale power purchase agreements. Existing on-site generation capacity in the industrial sector, exclusive of emergency generator sets, is in excess of 45,000 MW with a vast majority (42,000 MW) being combined heat and power

On-site generation provides the capability to reduce energy costs, comply with environmental regulations and insure a reliable power supply. On-site generation can ease congestion in the local utility's electric grid. Electric market restructuring is exacerbating the price premium for peak electricity use and for reliability, which is creating considerable market interest in on-site generation.

Remaining Potential for On-Site Generation in the Industrial Sector

The total remaining potential for in-fence on-site generation in the industrial sector is estimated by one DOE study at 140,000 MW. This potential represents the total site (in-fence) electric demand for power-only and CHP. The Industries of the Future represent approximately 113,000 MW or 79% of the total opportunity.

A high percentage of applications employing large on-site power plants have already been implemented as CHP. These applications dominate the installed capacity of CHP. Much of the remaining potential can be characterized as smaller discrete loads, mechanical drive applications providing chilled water, compressed air, refrigeration, and liquids pumping and facilities with smaller electric and/or thermal loads. The Rampressor Turbine will bring new economically viable CHP opportunities in these specific situations.

Remaining CHP Potential in the Industrial Sector

The remaining CHP potential in the industrial sector is estimated at 88,000 MW. Remaining potential for the Industries of the Future industry groups is estimated at 61,000 MW or 69% of the total CHP opportunity. The CHP potential includes systems that export power to the grid meaning that the estimated potentials are in excess of site (in-fence) demand requirements.

CHP Economics and Greenhouse Gases

The economics of CHP with current technology indicated that a 5 MW CHP system is

competitive with delivered electricity prices for 37 percent of industrial customers. A 1 MW reciprocating engine system is competitive in 20 percent of the industrial sector. The Rampressor Turbine will be more price competitive than current small engine options used in these calculations. This Report assumes that the Rampressor Turbine will be competitive in many regions in supplying electricity alone on a cost per kW basis.

CHP's competitiveness is sensitive to fuel price and declines as natural gas prices increase for each CHP system. The environmental value of CHP is most important for its potential to reduce global climate change emissions. One study estimates 130 GW being generated from CHP, which would produce a 70 million metric ton reduction in carbon equivalent emissions. This is equivalent to approximately 285 million tons of CO₂ emissions.

Barriers to On-Site Generation

Barriers to on-site generation in the industrial sector have included institutional, market and technical issues:

- high electric utility backup power costs, onerous grid interconnection requirements, and deferral rates;
- local environmental regulations that do not recognize the offsetting plant boiler and central station generation emissions;
- unfavorable depreciation tax treatment for on-site generation (non-core investment, requiring quick pay-back); and,
- high costs for equipment, installation, environmental compliance, and maintenance for smaller on-site generation systems;

Database Used in Studies

The DOE IOF used the Major Industrial Plant Database (MIPD), which contains a detailed description of over 18,000 of the largest industrial facilities in the U.S. The MIPD covers approximately 18,000 of the estimated 250,000 manufacturing facilities in the U.S. This database was employed in studies to aggregate the electrical capacity and steam use for each site and to sort them by size and power-to-steam (P/S) ratios for each 2-digit SIC (20-39) and for selected 4-digit SICs. Using the extensive data studies were able to identify the following criteria, which can be used in assessing specific sites for CHP applications as well as making general conclusions:

- $P/S < 0.4$ – Sites with a power to steam ratio of less than 0.4 have a relatively high steam load. The CHP system can be sized either to a) the steam load with export of excess power or b) sized to the on-site electric demand, meeting only part of the on-site steam requirements. A simple cycle gas turbine system with a P/S of 0.6 was selected as a representative CHP technology for these sites.
- $0.4 < P/S < 1.5$ – Sites with power to steam ratios between 0.4 and 1.5 are sized to the steam load and provide only part of the on-site electric requirements. A higher efficiency recuperated gas turbine or combined cycle with a P/S of 1.0 was selected as a representative CHP technology for these sites.
- $P/S > 1.5$ – Sites with a power to steam ratio of greater than 1.5 are not CHP candidates because their on-site steam load is too small compared to their

electrical requirements, however, they do represent a potential population of power-only systems.

Database of Small Industrial Sites

An important information source used by studies to estimate remaining on-site generation potential for the small industrial sector was the MarketPlace Database developed by iMarket, Inc. It was used to identify sites by SIC code that have average electric loads between 100-1,000 kW. Unlike MIPD, this database has limited site operating data. The database presorts facilities into ranges of power use. Because there is no direct steam consumption data, direct computation of P/S ratios is not possible. SIC categories having adequate steam loads were selected based on power and steam data profiles contained in the DOE 1994 Manufacturing Energy Consumption Survey (MECS). To determine CHP potential, a reciprocating engine system with a P/S of 0.8 was selected as a representative CHP technology for these sites.

Database of Operating CHP and Small Power Plants

The Hagler-Bailly database of CHP and small power plants was used by studies to identify the number of operating CHP plants in the industrial sector. For each of the industrial SIC categories, existing CHP capacity was subtracted from the total CHP market to arrive at the remaining CHP potential.

Agriculture Analysis

For the agricultural sector, information was obtained from the U. S. Census Bureau and electric demand in this sector was based on the amount of total farm expenditures.

Power-Only and CHP Technical Potential

To determine the remaining power-only and CHP technical potential for the industrial sector, a total of 8,900 IOF facilities from MIPD, 18,000 mining establishments and 1.91 million farms from the census data and over 25,000 Industries of the future sites less than 1 MW from the MarketPlace database were reviewed.

Remaining Power-Only and CHP Potential Sites with Plant Size less than 1 MW

		Power Only Market (MW)	CHP Market			Existing CHP (MW)	Remaining CHP Potential* (MW)
SIC	SIC Description		Plant Size		Total (MW)		
			100-1,000kW	>1,000 kW			
IOF							
1-7	Agriculture	11,462	n.a.	n.a.	n.a.	751	n.a.
11,12,14	Mining	12,844	n.a.	n.a.	n.a.	492	n.a.
20	Food and Kindred Products	5,896	2,683	9,997	12,680	4,594	8,086
24	Lumber and Wood Products	1,171	595	1,946	2,542	806	1,736
26	Paper and Allied Products	10,459	1,168	33,584	34,751	8,553	26,198
28	Chemicals and Allied Products	29,364	1,780	25,352	27,132	17,692	9,440
29	Petroleum and Coal Products	3,949	154	12,253	12,407	5,618	6,789
32	Stone, Clay, Glass, Concrete*	0	n.a.	2,699	2,698	774	1,924
33	Primary Metals Industries*	38,389	294	9,520	9,814	2,873	6,941
	IOF Total	113, 534	6, 674	95, 350	102, 025	42,153	61,115
Non-IOF							

21	Tobacco and Allied Products	77	16	87	103	131	-
22	Textile Mill Products	2,576	766	3,011	3,777	651	3,126
23	Apparel Manufacturing	335	n.a.	163	163	0	163
25	Furniture	583	n.a.	402	401	68	333
27	Printing and Publishing	685	n.a.	404	404	19	385
30	Rubber and Misc. Plastics	2,546	2,772	1,641	4,413	787	3,626
31	Leather and Tanning	92	n.a.	98	98	0	98
34	Fabricated Metal Products	2,558	4,050	1,676	5,726	78	5,648
35	Industrial Machinery and Equip.	4,095	4,787	1,598	6,385	149	6,236
36	Electrical and Electron. Equip.	4,958	n.a.	987	987	180	807
37	Transportation Equipment	4,884	1,169	4,243	5,412	808	4,604
38	Instruments and Related Prod.	2,483	972	590	1,562	59	1,503
39	Miscellaneous Manufacturing	374	784	343	1,128	402	726
	Non IOF Total	26,246	15,316	15,243	30,558	3,332	27,226
20-39	Total Manufacturing	139,779	21,990	110,593	132,583	45,485	88,369

("Assessment of On-Site Power Opportunities in the Industrial Sector", DOE, OIT, March 2001)

One major challenge faced by CHP is the lack of integrated systems. Finding the optimal CHP components that, when integrated, can meet the wide range of facility heating, cooling, and the electric loads are left up to the plant management and their supporting staff. Establishing the product base and a sufficient volume of installations to overcome this problem is an important factor for building the CHP market.

The estimated remaining CHP technical potential for the industrial sector with plant sizes 1 MW or less is approximately 88,000 MW. The paper, chemical, food, primary metals and refining sectors account for approximately 61,000 MW or 61 percent of the remaining CHP potential. Paper has the largest remaining CHP potential of any one sector, accounting for 26,000 MW. However, significant remaining potential exists in other industries such as textiles, rubber and plastics, metals fabrication and equipment, industries that have not aggressively implemented CHP. The CHP potential includes larger systems that export excess power to the grid. Often, the thermal load of an industrial site can support a CHP system with greater electrical capacity than just the site requirements.

The power-only potential is estimated at approximately 140,000 MW for the industrial sector. The power-only market potential represents the in-fence opportunity. CHP technical potential is sorted by site capacity requirements. Nearly half of the remaining potential is for installations greater than 50 MW. There is also a significant technical potential for sites less than 1 MW where there has been little activity due to the high cost of packaging and installing small CHP systems.

Remaining CHP Potential Sorted by Capacity

In the study on IOFs the forest products industry had the largest remaining CHP potential followed by chemicals and food. The chemical industry had the highest capacity saturation having implemented approximately 65 percent of its total CHP market, while refining has achieved 45 percent saturation. Non-IOFs had installed only 10 percent of their potential CHP, which makes them important to consider.

This study (which did not consider the Rampressor Turbine) came to a series of conclusions. Applications between 0-4 MW will be dominated by reciprocating engines and small industrial gas turbines. Most of the potential for small CHP exists in the food and non-IOF industries. The food, chemical and non-IOFs contain the largest number of applicable sites in the 4-50 MW size range that is suitable for combustion turbines. For sites having a potential greater than 50 MW, suitable for combustion turbines or combined cycle plants, the paper industry has the largest share of applicable sites followed by the refining and steel industries. Less potential exists for large CHP in the food and non-IOF sectors than there is for small CHP.

In the food industry, the highest CHP potentials are in meat and poultry processing, canned fruits and vegetables, and malt beverages. The study did not include the entire food industry, and there are opportunities in other sectors of food. In the paper industry, the highest potentials are in the paper and paperboard sectors. In the chemical industry, the highest CHP potentials are in the basic organic and inorganic chemicals sectors. Chlor-alkali production and nitrogenous fertilizers also represent important sectors within the chemical industry.

Chillers

Manufacturing industries used 217 trillion Btu of electricity for HVAC in 1994, accounting for about 8 percent of total manufacturing electricity use. A reciprocating engine-driven chiller is a means to reduce peak electrical demand in a facility by displacing electric chillers or other air conditioning equipment. Overall energy costs are reduced by lowering electrical demand costs. Engine-driven systems also provide good part-load performance as compared to an electric motor drive. Recovery of engine jacket and exhaust heat can supplement process heating or increase the system cooling capacity with an absorption chiller.

Refrigeration

Refrigeration equipment is a major consumer of energy for industrial process cooling and dehumidification. The engine-driven refrigeration market is growing, promising lower operating costs and enhanced reliability. Engine-driven systems are often used for capacity expansions, particularly when expensive transformer upgrades may be required. The longer operating hours of refrigeration applications improves the economics for engine-driven equipment, especially when heat recovery is employed. Refrigeration applications in subsectors of the food industry include small and large capacity refrigeration compressors. Primary applications include dairies, food processing, meatpacking and refrigerated warehouses. Total installed refrigeration capacity is estimated between 3-4 millions tons.

Energy Savings from CHP

One DOE study estimated the results of the total annual energy savings attributable to existing and remaining industrial CHP using the stated assumptions for annual load factor, average utility heat rate, T&D losses, and typical boiler efficiency. The study concluded that the existing 44 GW of industrial CHP saves 1.5 quads of energy per year, and the remaining potential of 88 GW could save an additional 2.7 quads of energy per

year.

The net overall cost savings that are calculated by subtracting total cost of CHP fuel, O&M, and capital from the annual savings. CHP cost savings are estimated for avoided electric power purchases and avoided boiler fuel using the cost and performance estimates for the representative technologies shown in the following table. These calculations show that existing CHP saves U.S. industry nearly \$2.5 billion annually and the remaining CHP potential could save an additional \$5.4 billion annually. If all of the remaining industrial CHP potential were implemented, the current \$69 billion annual energy bill could be reduced by about 8 percent.

Energy Savings for Industrial CHP

Size	CHP	CHP	CHP	Avoided	Net
Range	Capacity	Output	Energy	Avoided	Energy
(MW)	(MW)	(GWh)	Used	Utility	Saved
				Electric	
				(Quads)	
<i>Existing CHP</i>					
0.1-1	60	360	0.004	0.004	0.001
1-4	369	2,214	0.027	0.024	0.009
4-20	2,442	14,652	0.172	0.158	0.087
>20	41,372	248,232	2.290	2.685	1.055
Total	44,243	265,458	2.493	2.871	1.534
<i>Remaining CHP Potential</i>					
0.1-1	21,930	131,580	1.437	1.423	0.539
1-4	6,070	36,420	0.450	0.394	0.204
4-20	11,337	68,022	0.799	0.736	0.404
>20	48,983	293,898	2.711	3.179	1.249
Total	88,320	529,920	5.397	5.731	2.397

("Cooling, Heating, and Power for Industry: A Market Assessment", DOE EERE, August 2003)

Environmental Benefits

CHP environmental benefits including a reduction in both NO_x and CO₂ emissions.

Industry Specific Initiatives

Power and Steam Consumption

		Electricity		Steam*	
SIC	Industry	Trillion Btu	\$Millions	Trillion Btu	\$Millions
20	Food	200	\$3,200	570	\$1,300
24	Lumber	75	\$1,200	20	\$80
26	Paper	250	\$3,000	880	\$1,700
28	Chemicals	540	\$5,400	1,200	\$2,500
29	Oil Refining	120	\$1,500	300	\$600
32	Glass	120	\$1,700	210	\$70
33	Primary Metals	480	\$4,800	170	\$370
	Total IOF	1,800	\$20,700	3,200	\$6,700
20-39	Total Manufacturing	2,700	\$36,000	3,600	\$8,400
	IOF Share		58%		80%

(DOE EIA 1994 Manufacturing Energy Consumption Survey)

Existing CHP in the “Industries of the Future” (IOFs)

The DOE supported study focused on the IOFs concludes there is currently over 45,000 MW of installed CHP capacity, generating 142 billion kWh that represents over 15 percent of total industrial requirements. The IOFs account for 94 percent of the total electricity that is generated from industrial sources. The paper and chemical industries account for nearly 75 percent of the total production.

SIC	Industry Segment	Existing CHP SIC Capacity (MW)		Industry Segment	Existing CHP Capacity (MW)
1-7	Agriculture	750		Glass	
	Mining		3211	Flat Glass	50
10	Metal Mining	120	3229	Pressed and Blown	80
				Glass	
12	Coal Mining	250		Total	130
14	Nonmetallic Minerals	120	331	Steel	1,000
	Total	490		Aluminum	
20	Food	4,700	3341	Secondary Smelting	800
	Forest Products		3353	Sheet, Plate, and Foil	1,000
24	Lumber and Wood	800		Total	1,800
26	Paper	8,600		Metal Casting	
	Total	9,300	3322	Malleable Iron	0.1
				Foundries	
28	Chemicals	17,700		Total	0.1
2911	Petroleum Refining	5,600			
	IOF Total				41,500
	Total Industrial				48,000
	IOF Share				86%

(“Cooling, Heating, and Power for Industry: A Market Assessment”, DOE EERE, August 2003)

Components of Net Electrical Requirements

Total CHP capacity in the IOFs is over 42,000 MW representing 87 percent of the total CHP capacity in all agriculture, mining, construction, and manufacturing sectors.

SIC	Industry	Purchases/ Transfers	On-Site Generation	Transfers Out	Net Requirements
			(million kWhrs)		
20	Food	59,700	7,000	1,800	64,900
24	Lumber	20,800	2,000	1,000	21,800
26	Paper	72,200	58,800	9,300	122,000
28	Chemicals	160,000	46,800	7,700	199,000

2911	Petroleum Refining	34,800	13,800	1,400	47,200
33	Primary Metals	55,200	4,800	Net	60,000
	Total IOF	403,000	133,000	21,200	515,000
20-39	Total	804,000	143,000	28,200	918,000
	Manufacturing IOF Share	50%	94%	75%	56%

("Cooling, Heating, and Power for Industry: A Market Assessment", DOE EERE, August 2003)

Existing CHP Capacity

Installations are dominated by natural gas, representing the primary fuel source for 61 percent of installations. As shown in the table, other fuels are important in specific applications.

Approximately 85 percent of CHP systems in the IOFs use combined cycle and steam turbines. Approximately 22 percent of CHP installations have capacities of 50 MW and greater that account for almost 75 percent of the total existing CHP capacity. Approximately 28 percent of existing CHP installations in the IOFs is smaller than 4 MW, however, these sites account for only 2 percent of the capacity.

CHP Fuel Type in the IOFs

Fuel Type	Existing CHP Capacity (MW)	Share	Description
Natural gas	25,400	61.3%	Most widely used fuel with broad application in all
Coal	6,600	15.8%	industries Widely distributed but with higher concentrations in
Oil	1,100	2.6%	agriculture, mining, paper, and primary metals 60% used in petroleum refining
Wood	2,300	5.6%	94% used in forest products
Black liquor	2,200	5.2%	Used solely in the paper industry
Waste Energy	700	1.8%	Byproduct of sulfuric acid manufacture and refining and steel industries
Waste fuels	2,900	6.9%	Coal wastes, refinery gas, petroleum coke, mine gas,
Agricultural	200	0.5%	coke oven gas 88% used in Food Industry the rest in Agriculture
waste Renewable	130	0.3%	Bio-methane, municipal solid waste, more important
Waste			in municipal applications
Total IOF	41,500		

("Cooling, Heating, and Power for Industry: A Market Assessment", DOE EERE, August 2003)

Combined Heat and Power in the Building Industry

A DOE supported study that addresses the building industry uses the term Integrated Energy Systems (IES) to capture what the study considers the advances beyond what is thought of as traditional CHP. Integrated Energy Systems is defined as combining on-site power or distributed generation technologies with thermally activated technologies

to provide cooling, heating, humidity control, energy storage and/or other process functions using thermal energy normally wasted in the production of electricity/power.

The analysis concludes that the potential building sector market for IES is almost 17 GW in 2010, growing to over 35 GW by 2020, and includes IES systems with absorption chillers, engine-driven chillers (EDCs), and CHP-only systems.

IES includes significant opportunities for CHP with absorption units and engine-driven chillers (45 percent of the office potential). Of 4.5 GW, CHP with absorbers represent over 3.6 GW and EDCs 1.1, giving offices almost half of the total EDC potential.

Hospitals and colleges, while already established in CHP use, each offer over 7 GW of potential for IES, respectively. Schools, retail, and hotels are smaller segments, but with their significant heating and cooling loads offer additional IES potential. Military bases also offer potential for IES, but generally for CHP-only systems as military bases do not generally have base-wide cooling distribution systems.

The study notes that overall market potential results of the sensitivity analysis indicate that improvement in the installed cost and efficiency increases the market size dramatically. The studies future scenarios increased the potential market from 35 to almost 70 GW, nearly doubling the market size. The development of the Rampressor Turbine is an example of the projected improvements in installed cost and efficiency.

The study notes that while the overall size of CHP in the buildings sector is somewhat uncertain, most agree that this market is led by educational facilities, with the health care sector also important. Some of the factors that drive the favorable economics of CHP in these building types are:

- Occupancy levels are generally high, with students or patients occupying the facilities constantly, creating high load factors that help amortize the investment in CHP systems;
- The balance between thermal and electric loads in these building types is relatively high, compared with other building types;
- Multiple buildings under common ownership, so that electricity, heating, and cooling loads can be aggregated and served by a central system that is larger and more cost effective than several smaller systems;
- Close proximity of buildings, so that connecting buildings with hot water/steam/chilled water distribution piping is not cost prohibitive;
- Buildings are not leased to tenants, so there is a higher degree of control.

Absorption Chillers

The study considers absorption chillers an important option for IES building applications. They employ CHP thermal output during cooling periods when heating uses are limited to domestic hot water loads or zonal heating, which may be small in many building types. These units involve a complex cycle of absorbing heat from the CHP system to create chilled water.

While the absorption chiller technology has been around since the late 1800s, historically the manufacturing base for these units was largely in Japan. Japan had developed these units to help reduce dependency on high cost imported fuels, and recognized the benefits of higher efficiency levels that could be attained. During this period, however, availability and lead time for U. S. orders lagged that of conventional electric chillers, and thus only a small niche market emerged. In the 1990s, however, several of the largest U. S. manufacturers of electric chillers developed offerings, and were able to reduce costs and lead times, and improve availability. As a result, the market for absorption chillers has been growing.

Engine-Driven Chillers

Engine-driven chillers (EDCs) are conventional chillers driven by an engine, in lieu of an electric motor. They employ the same thermodynamic cycle and compressor technology that electric chillers use, but use a gas-fired reciprocating engine to drive the compressor. As a result, EDCs can be economically used to provide cooling where gas rates are relatively low and electric rates are high. Another benefit offered by EDCs is the better variable speed performance, which yields improved part load efficiencies. EDCs operate in a CHP system when the waste heat produced by the engine is recovered, and used for space heating and/or domestic hot water loads.

Desiccant Dehumidification Systems

Desiccant systems incorporate desiccant materials, which absorb moisture from air. After a period of exposure to humid air, these materials become saturated, and require regeneration if they are to be reused. This regeneration is typically accomplished by exposing the desiccant material to heated air. Desiccant systems typically incorporate a desiccant wheel, which rotates between a stream of ventilating air (from which it removes humidity) and a stream of heated air (which regenerates the desiccant material). By rotating through these two streams of air, the desiccant wheel dehumidifies the ventilating air and rejects the moisture to the heated air stream. Desiccant systems become a part of a CHP system when they use the waste heat from the CHP system to provide regeneration.

MARKET POTENTIAL

To achieve widespread use in buildings, IES treats the heating, ventilation, air conditioning, water heating, lighting and power loads as an integrated system, serving the majority of these loads either directly or indirectly from the CHP output. The market analysis used in this study examined the potential for IES when the system helps cooling needs, and not just as conventional CHP.

The study used a spreadsheet-based model to estimate the achievable economic potential for IES by comparing various options with traditional equipment. The model calculated fuel use, on-site electricity generation, electric and natural gas bills, installed cost, and economic return on investment for individual facilities. The model was configured to analyze commercial buildings throughout the U.S. using load profiles that estimate cooling, heating, hot water, electricity loads based on a number of different cities in the U.S., and simple payback was used as the economic decision measure.

As shown in the Table below, the potential includes both system turnover in existing buildings, as well as IES in new buildings. Included in this market is a significant potential for new absorption chillers (8.9 million tons), thermal storage (3.2 million tons), and engine driven chillers (2.4 million tons). Together, if implemented this market potential would represent almost 18 million metric tons of reduced carbon dioxide emissions (based on carbon equivalent) annually by the year 2020, and would contribute significantly to meeting the goals originally established by the Kyoto Protocol. This reduction in carbon emissions is based on displacing grid emissions from average U. S. utility plants.

Building Market Potential for IES

	2010		2020	
	Turnover	New	Turnover	New
Capacity (GW)	10.4	3.5	19.0	16.5
Displaced Electricity (GWh)	66,900	45,800	133,900	117,100
Capital (\$million)	6,380	4,150	12,750	10,430
Incremental NG Use from CHP(Tbtu)	550	380	1,090	960
Boiler Fuel Displaced (Tbtu)	90	60	180	170
Absorber (1000 tons)	3,310	2,340	6,630	5,860
Storage (1000 tons)	740	670	1,480	1,730
Engine Driven Chiller (1000 tons)	630	470	1,260	1,180
CO2 Displaced From Utility (MtC)	12.1	8.3	24.3	21.2
MtC Produced By Incremental NG	7.9	5.4	15.8	13.9
CO2 Displaced from Boiler (MtC)	1.8	1.2	3.5	3.3
CO2 Displaced (MtC)	6.0	4.1	12.0	10.6

("Integrated Energy Systems (IES) for Buildings: A Market Assessment", DOE EERE, August 2002)

This market potential, when compared to the estimated IES capacity of 3.5 GW currently installed, yields a current penetration level of less than 10 percent. Increasing market penetration to 50 percent levels would add over 14 GW of capacity.

The results indicated that each cooling option had a unique operating scheme that offered the best economics. For CHP with absorption chillers, operating the chiller to serve both baseload and peak loads, as well as operates on-peak and off-peak made the most sense. This was compared to using the absorption unit for peaking only, baseload only, and examining on-peak only operation as well as using storage for peaking operation.

CHP-only systems are by far the most representative, with 22 GW in potential. This large share is primarily due to the base of buildings (both single buildings and multi-building facilities) that possess hot water/steam or forced air distribution systems but do not have chilled water distribution. This base of buildings is almost two-thirds of the base of buildings that have gas and electric utility service. Further assessment could determine the potential for distribution of small heat activated air conditioning throughout buildings with hot water or steam loops and no chilled water distribution systems.

Analysis of Building Types

The study found that most IES is concentrated in education and health care buildings.

While some barriers still exist in this sector, such as the price of backup power and the regulated market for surplus power, other barriers such as first cost have not been factors. Similarly, hospitals have a smaller but still significantly large installed base of CHP, and lodging has a small base of applications.

The potential for IES is highest in office buildings, with over 10 GW of total IES, including significant opportunities for CHP with absorption units and engine-driven chillers (45 percent of the office potential). Of that 4.5 GW, CHP with absorbers represent over 3.6 GW and EDCs 1.1, giving offices almost half of the total EDC potential. Hospitals and colleges, while already established in the CHP use, each offer over 7 GW of potential for IES, respectively. Schools, retail, and hotels are smaller segments, but with their significant heating and cooling loads offer additional IES potential plus opportunities for absorption units and EDCs. Schools and apartment buildings offer the highest share of CHP with absorbers or EDCs, constituting over half of the total IES potential for that building type.

Military bases and supermarkets, offer potential for IES, but not much for absorbers or EDCs. Supermarkets tend to lack chilled water distribution systems, and military bases do not generally have base-wide cooling distribution systems.

Regional Analysis

The economics of IES is driven largely by the relative prices of gas and electricity. Portions of the U.S. where low gas prices and/or high electric prices prevail often offer good conditions for IES, areas where both high electric and low gas prices offer ideal conditions.

The top states for IES potential include Alaska, California, New York, New Jersey, Vermont, and New Hampshire. Other states in the New England, Middle Atlantic, Mountain, West South Central, and East North Central are strong prospects.

- The Pacific region, dominated by a large building population, high electric prices and intensive cooling loads in California, offers the most IES potential. This region also offers the most potential for CHP with absorption or EDCs. There is some CHP-only potential due to still significant heating loads.
- The Middle Atlantic and New England regions each have a significant share of the IES potential. These regions also offer a major portion of the CHP-only potential but offer a lower percentage of the CHP with absorption or EDCs. The West North Central region also follows this pattern, but on a smaller scale.
- The South Atlantic, West South Central, and East North Central regions, offer a balance between CHP with absorption or EDC cooling (about 40-45 percent of the total) and straight CHP (about 55-60 percent). Some states (TX, IL, and MI) offer somewhat attractive energy prices for IES.
- East South Central and Mountain regions, with some intense cooling loads and generally low gas prices (although not the past two years), have the highest percentage of CHP with absorption or EDCs of any region. New Mexico and Arizona offer particularly strong energy prices for IES. Relatively small heating loads (on a regional basis) in these

areas contribute to a small CHP-only potential.

CHP Potential at Federal Sites

One DOE study considered Combined Heat and Power (CHP) systems at Federal building. The information in this section was taken from that study. CHP was highlighted in the Bush Administration's National Energy Policy Report as a commercially available and feasible option that offers extraordinary efficiency and environmental benefits.

The U.S. Department of Energy's (DOE's) Federal Energy Management Program (FEMP) completed a market assessment of the potential for CHP applications at Federal facilities and the associated costs and benefits including energy and emissions savings. Results of the assessment indicate that CHP can make significant contributions toward our energy-conservation and emissions-reduction goals and save the government money. The assessment was meant to help Federal agencies understand and capitalize on the opportunities to implement CHP technologies and offers a broad overview of when and where CHP systems are most likely to serve the Federal sector's best interest.

The study noted that as of February 2002, over 50 Federal sites were already benefiting from CHP systems, and another 50 were working on projects to install 100 MW of additional CHP capacity.

The market assessment considered 7 building types for 28 Federal agencies. The Federal building types with greatest CHP potential are hospitals, industrial, and R&D facilities. Agencies with most potential are the military, Department of Veterans Affairs (VA), and Department of Energy (DOE).

Total potential CHP capacity was estimated to be 1500–1600 MW under base-case assumptions using gas reciprocating engines or gas-combustion turbine technologies. This would represent approximately 13% of all electricity use in the Federal sector.

The assessment reveals significant Federal potential for CHP in the Southwest (California to Texas), Northeastern metropolitan areas (New York to Washington, D.C.), and the Southeast (Florida, Georgia, and Alabama). The 1.5 GW estimated in the base-case scenario has an average simple payback of 7 years and could save the Federal government \$170 million per year in energy costs.

The National Energy Policy Report highlights the potential for CHP to cost effectively assist agencies meet goals related to energy security, cost reduction, and environmental quality. The United States has over 50 gigawatts (GW) of installed CHP capacity, which produces about 7% of the nation's electricity. CHP offers an opportunity to meet increased energy needs, reduce transmission congestion, cut emissions, increase power quality and reliability, and increase overall energy security. Networks of interconnected CHP systems can offer increased power security for the grid as well.

Federal facilities are directed by Executive Order 13123 to lead by example in meeting national energy and environmental goals. This Order specifically directs agencies to "strive to reduce total energy use and other air emissions at the source.

Recognizing the federal mandates and benefits of CHP, DOE, the U.S. Environmental Protection Agency (EPA) and the private sector have embarked on a joint effort to double the amount of CHP capacity in the U.S. by 2010.

Potential CHP Capacity

The total amount of CHP potential capacity for Federal facilities nationwide is estimated to be between 1500 and 1600 MW. Under base case assumptions, CHP systems would produce 7.7 TWh of electricity. Producing 7.7 TWh with CHP would represent over 13% of the 57 TWh of electricity purchased by the Federal government in FY 2000. This CHP capacity would provide electricity and thermal energy for about 580 million ft² of building space in 9% of all Federal sites. The potential will be greatest in large sites with central plants or mechanical rooms and high electricity rates.

With more than 400 MW, Federal hospitals represent the building category with the highest CHP potential. They also show the most promising target of opportunity, since over two-thirds of large hospitals are expected to have CHP potential. Industrial buildings are next in potential capacity, at 340 MW, and are second in percentage of sites at 42%. R&D facilities, office buildings, and service buildings offered 270, 250, and 210 MW, respectively.

Federal CHP Potential by Geographic Area

The regions with the largest Federal CHP potential are the Western, Central, and Southeast Regions.

Nearly all CHP potential is found among nine agencies: the three military services, VA hospitals, DOE, NASA, General Services Administration (GSA), the U.S. Postal Service, and the Department of Justice. The first three agencies (military, VA and DOE) represent 83% of the total CHP potential identified in the base case. The military services (over 50% of total) have significant potential CHP capacities in most types of buildings, while the VA has capacity mainly in hospitals. Energy and NASA capacity is concentrated in R&D and industrial buildings, while GSA and the Postal Service have capacity in the "office" category.

Potential CHP capacity by Federal agency and building category, MW

Agency	Hospital	Industry	Office	Prison	R&D	School	Service	Total
Air Force	43	57	31	0	85	7	116	339
Veterans Affairs (VA)	311	0	1	0	1	0	0	314
Army	55	101	52	2	24	3	33	270
Navy	27	36	39	0	43	2	58	205
Department of Energy	0	113	15	0	64	0	2	195
National Aeronautics and Space Administration (NASA)	0	17	10	0	43	0	3	73
General Services Administration (GSA)	0	1	68	0	0	0	0	69
United States Postal Service	0	0	48	0	0	0	0	48
Justice	0	3	0	34	0	0	0	37
Health and Human Services	6	0	0	0	2	0	0	9

Treasury	0	8	0	0	0	0	0	8
Transportation	0	0	2	0	4	0	0	7
Interior	0	2	2	0	0	3	0	7
Agriculture	0	0	0	0	3	0	0	3
Other	0	0	0	0	2	0	0	7
Grand total (MW)	443	338	269	36	274	16	212	1588

("CHP Potential at Federal Sites", DOE EERE, May 2002)

Federal Building Conclusions: CHP

There is significant potential—1,500 to 1,600 MW of capacity—for CHP to serve Federal facilities. Regions with the greatest CHP potential are the southwest, northeastern metropolitan areas, and the southeast. Agencies with the most potential are the military, VA, and DOE, especially in hospital, industrial, and R&D facilities.

The 1.5 GW identified under the base-case scenario would be sufficient to power more than a million homes and save the Federal government \$170M per year in energy costs. To install the 1.5 GW of electrical CHP generating capacity (all cases where the simple payback period is under 10 years) would require an estimated \$1.5–\$2 billion in capital investments. The source (primary) energy savings from this level of CHP investment are estimated to be 50 trillion Btus per year, and projected carbon dioxide emissions would be reduced by 2.7 million metric tons per year compared to gas-fired central-electric-power and thermal alternatives.

Details on CHP Potential by State

The following table shows potential CHP capacity by state for leading agencies in megawatts. Details of CHP capacity by state and agency represent probabilities as calculated in the model.

Table C-1: Potential CHP capacity by state for leading agencies (MW)

State	Air Force	VA	Army	Navy	Energy	NASA	GSA	Postal Service	Justice	Total
AK	13	0	6	1	0	0	3	1	0	24
AL	1	12	4	0	0	1	0	0	0	18
AR	3	8	2	0	0	0	0	0	1	14
AZ	10	4	9	1	0	0	1	1	1	27
CA	75	27	13	99	29	25	32	26	5	331
CO	19	4	14	0	9	0	12	4	2	64
CT	0	2	3	4	0	0	1	1	0	11
DC	0	5	11	0	0	0	0	0	0	16
FL	35	14	0	17	0	27	0	0	4	97
GA	32	10	6	0	0	0	0	0	4	52
HI	4	0	10	13	0	0	1	1	0	29
IA	0	4	0	0	0	0	0	0	0	4
IL	2	15	8	2	1	0	0	0	2	30
IN	0	7	7	4	0	0	0	0	1	19
KS	6	4	16	0	0	0	0	0	1	27
LA	5	6	10	3	0	10	5	2	1	42
MA	8	8	4	2	0	0	4	3	1	30
ME	0	1	0	0	0	0	0	0	0	1
MI	2	9	7	0	0	0	3	4	0	25
MN	2	6	6	5	0	0	0	0	0	19
MO	10	7	9	0	7	0	0	0	0	33

MS	3	5	2	0	0	0	0	0	0	10
NC	0	9	4	3	0	0	0	0	1	17
ND	11	1	0	0	0	0	0	0	0	12
NH	1	1	1	1	0	0	0	0	0	4
NJ	10	4	17	5	1	0	3	5	0	45
NM	17	2	10	0	37	1	3	1	0	71
NV	4	1	1	0	4	0	0	0	0	10
NY	6	27	4	5	10	0	0	0	1	53
OH	3	9	7	0	34	0	0	0	0	53
OK	8	4	4	0	0	0	0	0	1	17
OR	0	5	0	0	0	0	0	0	0	5
PA	0	16	2	1	0	0	0	0	2	21
RI	0	1	0	7	0	0	0	0	0	8
SC	0	0	0	0	13	0	0	0	0	13
SD	8	2	0	0	0	0	0	0	0	10
TN	0	14	7	3	38	0	0	0	0	62
TX	38	22	40	19	4	10	0	0	8	141
UT	0	3	0	0	0	0	0	0	0	3
VA	0	10	13	10	0	0	0	0	0	33
VT	0	1	0	0	0	0	0	0	0	1
WA	0	6	5	0	7	0	0	0	0	18
WI	0	9	8	0	0	0	0	0	0	17
WV	0	8	0	0	0	0	0	0	0	8
Total	339	314	270	205	195	73	69	48	37	1550

("CHP Potential at Federal Sites", DOE EERE, May 2002)

Under base case assumptions, the six states with the largest Federal CHP potential are California, Texas, Florida, New Mexico, Colorado, Tennessee, and the New England States.

Market Analysis Conclusions for Rampressor Turbine

Using information from the DOE studies and information provided by Ramgen, the following conclusions were made as to the priority applications of the Rampressor Turbine to reduce greenhouse gases:

- 1) A priority application for the Rampressor Turbine is the food processing industries (particularly meat and poultry processing, canned fruits and vegetables, and malt beverages). This industry has high electrical use, high need for heat for processing, has experience with CHP, and can effectively use small engine.
- 2) There are strong potential applications for use of the Rampressor Turbine in chemicals; forest products (includes paper), fabricated metal products; industrial machinery and equipment; transportation equipment; textiles; and rubber. All of these have a high electrical use, a high use of heat, currently use CHP and have uses for small engines.
- 3) A priority application for the Rampressor Turbine is CHP with cooling. New technologies are opening this up with more economically viable systems and the efficiencies and the cost advantages of the Rampressor Turbine is an effective match with these new systems.
- 4) A priority application for the Rampressor Turbine is in the building industry with Hospitals and Colleges each of which have the potential for 7 GW of CHP. The

24 hour usage of these facilities and the experience they have with CHP creates special opportunities with the higher efficiency of the Ramgen Turbine coming on line.

- 5) A priority application for the Rampressor Turbine is with selected DoD and DOE facilities and VA hospitals. The federal government is a leader in CHP. In addition to benefiting from the saving of economically viable systems, a model will be created that leads to more private sector investment.

Rampressor Turbine – Distributed Generation

Distributed Generation

		2004	2009	2014	2019	2024
	kWhrs (3% growth per year) – kWh (billion)	4,279	4,960	5,750	6,666	7,728
new kWh/year consumption	New kWh (billion) Demand/ year	124.627591	144.477535	167.489060	194.165725	225.091291
	New kWh (billion) Demand	124.627591	806.142338	1,596.204717	2,512.103549	3,573.881321
kWh/year generated by RPS	kWhrs/year - RPS	1,246,275,905	40,307,116,922	79,810,235,848	125,605,177,473	178,694,066,032
	kWh - RPS	1,246,275,905	112,108,669,888	429,819,165,244	963,548,906,593	1,747,704,159,373
kWh/year saved (50% efficiency improvement - 40% to 80%) - RPS	kWh/year – RPS	623,137,953	20,153,558,461	39,905,117,924	62,802,588,736	89,347,033,016
	kWh - RPS	623,137,953	56,054,334,944	214,909,582,622	481,774,453,297	873,852,079,687
\$ kWh/year saved w/ RPS in CHP	\$ - kWh/year - RPS	\$46,236,836	\$1,495,394,038	\$2,960,959,750	\$4,659,952,084	\$6,629,549,850
	\$ - kWh - RPS	\$46,236,836	\$4,159,231,653	\$15,946,291,031	\$35,747,664,435	\$64,839,824,313
lbs. CO2 generated by new kWh/year	lbs CO2/year	1,646,330,471	53,245,701,455	105,429,321,555	165,924,439,441	236,054,861,229
	lbs CO2	1,646,330,471	148,095,552,923	567,791,117,287	1,272,848,105,609	2,308,717,194,532
mtCO2e generated by new kWh/year	mtCO2e/year	747,434	24,173,548	47,864,912	75,329,696	107,168,907
	mtCO2e	747,434	67,235,381	257,777,167	577,873,040	1,048,157,606
lbs. CO2 generated by new kWh/ year – RPS	lbs. CO2/year - RPS	823,165,235	26,622,850,727	52,714,660,778	82,962,219,721	118,027,430,614
	lbs. CO2 - RPS	823,165,235	74,047,776,461	283,895,558,643	636,424,052,805	1,154,358,597,266
mtCO2e generated by new kWh/year – RPS	mtCO2e/year - RPS	373,717	12,086,774	23,932,456	37,664,848	53,584,453
	mtCO2e - RPS	373,717	33,617,691	128,888,584	288,936,520	524,078,803
\$ value of credits for CO2e reductions/ year - RPS	\$ mtCO2e reduced/ year - RPS	\$186,859	\$6,043,387	\$11,966,228	\$18,832,424	\$26,792,227
	\$ mtCO2e – RPS	\$186,859	\$16,808,845	\$64,444,292	\$144,468,260	\$262,039,402
\$ kWh and CO2e credits/year - RPS	\$ kWh + mtCO2e/ year - RPS	\$46,423,695	\$1,501,437,425	\$2,972,925,978	\$4,678,784,508	\$6,656,342,077
	\$ kWh + mtCO2e - RPS	\$46,423,695	\$4,176,040,498	\$16,010,735,322	\$35,892,132,695	\$65,101,863,714
Percent of US kWh consumption		100%				
new kWh market/ year		1%	2%	3%	4%	5%
CO2 - lbs. (kWhrs X 1.321)		1.321				
CO2 lbs X .000454 = MTCO2e		0.000454				
Efficiency - 40% - 80% (MTCO2e/2)		50%				
kWh (billion)		1,000,000,000				
\$ value Carbon Credit		\$5.52				
\$ per kWh - EIA, 0804		\$0.07				

PART C

1. Eligibility of the Ramgen Engine to Earn Carbon Credits

The Prototype Carbon Fund identifies five Energy Efficiency Improvements Project types and four Other Project Activities that qualify to earn carbon emission reductions. Energy Efficiency Improvement Project types identified are:

1. Supply side energy efficiency improvements – transmission and distribution
2. Supply side energy efficiency improvements – generation
3. Demand-side energy efficiency programs for specific technologies
4. Energy efficiency and fuel switching measures for industrial facilities
5. Energy efficiency and fuel switching measures for buildings

Projects: Other Project Activities that are identified:

1. Agriculture
2. Switching fossil fuels
3. Emission reductions by low-greenhouse gas emitting vehicles
4. Methane recovery and avoidance

Energy Efficiency Improvement Projects

Supply side energy efficiency improvements – transmission and distribution

Because this category of project is targeted at the energy efficiency of electricity or heat transmissions and distribution, it does not apply to Rampressor.

Supply side energy efficiency improvements – generation

“This category comprises technologies or measures to improve efficiency of fossil fuel generating units that supply an electricity or thermal system by reducing energy or fuel consumption by up to the equivalent of 15 GWh per year. Examples include efficiency improvements at power stations and district heating plants and co-generation. The technologies or measures may be applied to existing stations or be part of a new facility.

Baseline: The energy baseline is the technical losses of energy within the project boundary. In the case of retrofit measures, the energy baseline is calculated as the monitored performance of the existing generating unit. In the case of new facilities, the energy baseline is calculated using a standard for the equipment that would otherwise have been installed”⁴¹

This category applies to the Rampressor when operating in combined heat and compression and gas fired cogeneration configurations.

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⁴¹(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

Demand-side energy efficiency programs for specific technologies

“This category comprises programs that encourage the adoption of energy-efficient equipment, lamps, ballasts, refrigerators, motors, fans, air conditioners, appliances, etc. at many sites. These technologies may replace existing equipment or be installed at new sites. The aggregate energy savings by a single project may not exceed the equivalent of 15 GWh per year.

Baseline: If energy displaced is fossil fuel: fuel use multiplied by emission coefficient for the fossil fuel, where fuel use is either existing fuel use or related to technology that would have been implemented otherwise.

If energy displaced is electricity: $\text{Energy generation} = (\text{quantity} \times \text{power} \times \text{operating hours}) / (1 - \text{technical T\&D losses})$, power being either the weighted average of replaced devices or the weighted average of devices on the market⁴²

The efficiency of the Rampressor would qualify to earn carbon emission-reduction credits under this category.

Energy efficiency and fuel switching measures for industrial facilities

“This category comprises any energy efficiency and fuel switching measure implemented at a single industrial facility. Examples include energy efficiency measures (such as efficient motors), fuel switching measures (such as switching from steam or compressed air to electricity) and efficiency measures for specific industrial processes (such as steel furnaces, paper drying, tobacco curing, etc.). The measures may replace existing equipment or be installed in a new facility. The aggregate energy savings of a single project may not exceed the equivalent of 15 GWh per year.

Baseline: Energy use of the existing equipment that is to be replaced or energy use of the facility that would otherwise be built (both adjusted for T&D losses)⁴³

The efficiency of the Rampressor would qualify to earn carbon emission-reduction credits under this category.

Energy efficiency and fuel switching measures for buildings

“This category comprises any energy efficiency and fuel switching measure implemented at a single building, such as a commercial, institutional or residential building, or group of similar buildings, such as a school district or university. Examples include technical energy efficiency measures (such as efficient appliances, better insulation and optimal arrangement of equipment) and fuel switching measures (such as switching from gas to electricity). The technologies may replace existing equipment or be installed in new facilities. The aggregate energy savings of a single project may not exceed the equivalent of 15 GWh per year.

Baseline: Energy use of the existing equipment that is to be replaced or energy use of the

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⁴²(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

⁴³(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

facility that would otherwise be built (both adjusted for T&D losses)’’⁴⁴

The efficiency of the Rampressor would qualify to earn carbon emission-reduction credits under this category when operating at either a commercial or residential site.

Projects: Other Project Activities

Agriculture

“The CDM - EB⁴⁵ considers that more work is needed on this category before proposing simplified baseline and monitoring methodologies.”⁴⁶

The definition of this category is currently under development by the CDM EB. This analysis suggests that it is highly unlikely that this category, when definition is finalized, will have relevance to applications of Ramgen technology.

Switching fossil fuels

“This category comprises fossil fuel switching in existing⁴⁷ industrial, residential, commercial, and institutional or electricity generation applications. Fuel switching may change efficiency as well. If fuel switching is part of a project activity focused primarily on energy efficiency, the project activity falls in category II.D or II.E. Measures must both reduce anthropogenic emissions by sources and directly emit less than 15 kilo tonnes of carbon dioxide equivalent annually.

Baseline: current emissions per unit of output using IPCC default emission factors.”⁴⁸

Ramgen Engine projects using recovered GHG as a fuel stream will qualify under this category to earn carbon emission-reduction credits, following technical review of emission restrictions

Emission reductions by low-greenhouse gas emitting vehicles

“This category comprises low-greenhouse gas emission vehicles. Measures must both reduce anthropogenic emissions by sources and directly emit less than 15 kilo tonnes of carbon dioxide equivalent annually.

Baseline: energy use per unit of service for the vehicle that would otherwise have been used times the average annual units of service per vehicle times the number of vehicles replaced times the emission coefficient for the fuel used by vehicle otherwise used.”⁴⁹

This category is not applicable to the stationary Ramgen Engine.

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⁴⁴(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

⁴⁵UNFCCC - Clean Development Mechanism – Executive Board (CDM - EB)

⁴⁶(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

⁴⁷This does not preclude project participants from proposing simplified baselines for switching of fossil fuels for new applications.

⁴⁸(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

⁴⁹(World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

Methane recovery and avoidance

“This project category comprises methane recovery from coal mines, agro-industries, landfills, wastewater treatment facilities and other sources. Measures must both reduce anthropogenic emissions by sources and directly emit less than 15 kilo tonnes of carbon dioxide equivalent annually.

Baseline: The emission baseline is the amount of methane that would be emitted during the crediting period in the absence of the project activity. The baseline shall cover only the capture and flaring that would not have happened in the absence of the project activity.”⁵⁰

The Ramgen Engine using waste GHG as the fuel stream will qualify for this category.

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⁵⁰ (World Bank, Carbon Finance Unit, *Small Scale CDM Projects: An Overview*, May 14, 2003)

2. Applications of Ramgen Engine to Reduce Emissions of Greenhouse Gases

Ramgen Engine, with the patented AVC system (advanced vortex combustion), is projected to further increase efficiency and provide the capability for use of waste gases as fuel. The Ramgen engine can be sized from 800 kW to 10 MW. The work underway on the Rampressor and Ramgen Turbine engine set the technology base for the Ramgen engine.

The Ramgen engine is projected to have the capability to use coal mine methane, landfill methane, and methane from biomass, economically as a fuel. There is strong interest on the part of some scientists in a strategy to reduce greenhouse gases by primarily addressing gases such as waste methane. Methane is 21 times more potent as a Greenhouse Gas than carbon dioxide. Using methane, released in the process of mining coal and from land fills, as fuel for the Ramgen Engine is projected to be economically viable.

Overview

Methane, the waste gas addressed in this Section of the Report, is rated as the second largest anthropogenic⁵¹ greenhouse gases contributor, after carbon dioxide, to global warming. Methane, per volume, traps 21 times more heat in the atmosphere than carbon dioxide; therefore eliminating 1 ton of methane from the atmosphere is equivalent to eliminating 21 tons of carbon dioxide. Coalmine and landfill methane is selected as a

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⁵¹Source of greenhouse gas resulting from human activity

priority application of the Ramgen engine because they both emit significant amounts of methane into the atmosphere adding to greenhouse gases, offers the potential to capture methane economically in a short time frame using available and proven capture technology; and, provide a methane fuel stream the Ramgen engine has unique capabilities to use efficiently and economically to generate electricity.

Reducing methane is considered by some experts to be critical to a realistic strategy over the next 40 years on greenhouse gases. These experts argue that a comprehensive multi-gas mitigation strategy would be less expensive and more effective in mitigating climate change than focusing primarily on CO₂. A study directed by the Chief of the Goddard Institute for Space Studies at NASA, Dr. James Hansen, concluded that the climate effects of reducing greenhouse gases other than CO₂ can exceed CO₂ reductions in the shorter timeframe and is much more feasible.

The effect of methane on anticipated future temperature increase is intensified because of the consequences of the atmospheric lifetimes of these gases. Methane remains in the atmosphere for approximately 10 years compared to 100 years for carbon dioxide and if methane emissions were significantly reduced in the short term, the effect on atmospheric concentrations would be seen within a decade or two, much more quickly than similar reductions in CO₂ emissions.

In developing an effective greenhouse reduction strategy, consideration also needs to be given to whether it is more practical and economic to capture 1 ton of methane compared to 21 tons of carbon dioxide. If the volume of methane suitable for capture economically is sufficient to affect the greenhouse gas effect there are very important reasons to make methane capture a priority.

The USEPA has estimated that a roughly 10% reduction of the total world wide emission of anthropogenic methane per year would stop the annual rise in methane concentrations in the atmosphere. Of the total 3,193 to 7,983 MMTCO₂e of methane emitted from all anthropogenic sources in 1990, 420 to 1,260 MMTCO₂e came from landfills, 462 to 756 MMTCO₂e came from coalmines, and 1,470 to 4,202 MMTCO₂e came from livestock manure management. These sources, together, emit a volume of waste methane equal to 2,352 between 6,218 MMTCO₂e per year and constituted 73% to 78% of the total methane emitted from anthropogenic sources in 1990.

Extensive information was developed in an earlier Report to DOE by Ramgen on the greenhouse gas reductions that can be projected from the extensive use of the Ramgen engine with coalmine methane. As part of the earlier Report a highly innovative approach was developed, with leadership from experts at the Jim Walter Company, to maximize the use of coalmine methane. This Report uses that information and approach as a base to identify applications at specific mines of this innovative, realistic and highly promising program for reducing greenhouse gases. The information on carbon credits has also been updated for both coalmine methane and landfill methane.

Methane from coal mines, landfills and biomass are reviewed in this report and scenarios

are developed as to how the introduction of the Ramgen technology could reduce these greenhouse gases. Landfills and coal mines require systems to collect methane but biomass (that is not in landfills) is more complicated as the methane has to be extracted rather than collected. One issue with biomass is what amount of methane can be economically obtained by gasification or other systems to extract methane from biomass prior to the methane being emitted into the atmosphere as the biomass decays.

The objective of Part C of this report is to describe specific opportunities for commercially viable application of Ramgen technology with waste gases and to outline scenarios by which reductions in greenhouse gas emissions would result. The scenarios consider the available methane resources, the costs of securing this methane and estimated revenues from sales of electricity and from carbon credits in the developing international markets. Scenarios for one year and for 20 years have been developed for greenhouse gas emission-reductions using, for purposes of example, data from the year 2000 to 2020. These scenarios were developed to provide examples of the reduction in greenhouse gas that could result from the widespread use of Ramgen Engines and are dependent upon the projections of the technology capabilities and the accuracy of available government and industry data.

COAL MINE METHANE

Methane Emitted During Coal Mining

Methane is trapped in coal seams and is released during mining. The volume of methane emitted during mining and post-mining activities is variable, depending on the particular coal seam and the type of mining technique employed. In 2000, coalmine operations worldwide released an estimated 474 MMTCO₂e of methane. Methane released by coalmining is emitted into the atmosphere during active mining and escapes from sealed areas after active mining where methane collects and eventually escapes.

Worldwide coalbed methane emissions are expected to increase as coal production grows during the next decade. By 2020, the total methane released by coalmine operations is expected to reach at least 588 MMTCO₂E. Deep underground mining activity is increasing and deep mines tend to have coal with higher concentrations of methane than the coal seams closer to the surface.

Efforts to Find Commercial Use for Coal Mine Methane

Extensive interest has been focused in recent years on lowering the volume of methane released from coal mines and vented to the atmosphere. The factors driving the search for a means to do this include:

1. Coalmine methane has been identified as a major greenhouse gas and has been targeted because there has been hope for finding an economically viable use for it,
2. Gas monitoring and gathering systems that have been developed at US mines over the past 30 years create the opportunity to control gas flow with greater precision and safety,
3. Underground coal mines in the U.S. and other countries are experiencing economic challenges and are searching for ways to increase revenue, and,

4. The deregulation of the U.S. electrical industry offers new opportunities for marketing electricity generated at mines using methane as the primary fuel stream. Coalmines have enormous electricity demands and are a major electricity purchaser; giving coal mines a significant incentive to generate electricity onsite using now wasted methane as fuel, as long as the electricity can be generated at a cost lower than the purchase price of electricity supplied by the grid.

A limited number of coal mines are producing high quality methane that can be sold as natural gas. At these mines, significant progress has been made with the development of gas collection systems. The effort to find a commercial use for low quality methane, which is the majority of methane emitted from coal mines, has concentrated on developing a technology that can use the at low concentration methane as fuel to generate electricity. Currently there is no commercially accepted way to generate revenue using this gas. For example, there are only a small number of mines in the U.S. that use recovered methane from their degasification system as fuel for internal combustion engines to generate electrical power and these mines are using methane at 70% concentrations rather than the 1% from ventilation systems. There are demonstration projects operating at coal mines in China, Australia, England, and Germany but they all face the problem of marginal financial return due to the high capital requirements and operating cost.

Jim Walter Resources (JWR)

Blue Creek No. 4, No. 5, and No. 7 Mines

Located in Jefferson and Tuscaloosa Counties, Alabama, the JWR mines are among the deepest and gassiest mines in the country. Opened in the early to mid-1970s, the mines cover an 80,000 acre area and have vertical shafts ranging from 1,300 to 2,100 feet in depth. The in-situ gas content of coal is about 500 to 600 cubic feet per ton and the total amount of methane liberated from these mines is estimated to be between 2,200 – 5,800 cubic feet per ton of coal produced.

JWR has been a leader in the development of coalmine, methane recovery projects in the United States. The company's Blue Creek mines -- the Nos. 4, 5, and 7 mines -- are currently recovering and selling approximately 34 million cubic feet of gas per day (Alabama Oil & Gas, 2002). Methane is produced using three recovery methods: 1) vertical degasification (holes drilled from the surface into the virgin coalbed); 2) horizontal degasification (holes drilled in the coalbed from active workings inside the mine); and 3) gob degasification program (holes drilled from the surface into the caved area behind the longwall faces).

Since the late 1980s, JWR has been producing between 25 – 35 mmcf/d of methane. As of December 2001, there were 256 wells producing approximately 27 mmcf/ d. The quantity of methane recovered in 2001 represents 45 percent of total methane liberated from the mines. Depending on the mine, recovery from vertical pre-mine wells in 2001 made up approximately 15 - 35 percent of production, while gob wells and in-mine boreholes made up the remaining 65 - 85 percent of production.

Jim Walter Resources

Mine	Production Coal (tons)	Ventilation Emissions (mmcf/d)	Estimated Methane Drained (mmcf/d)	Estimated Total Methane Emissions Liberated (mmcf/d)	Methane cf/ton (mmcf/day)	Estimated Methane Used
Blue Creek #4	2.5	8.0	8.0	15.9	2,290	8.0
Blue Creek #5	1.5	13.2	10.4	23.6	5,865	10.4
Blue Creek #7	1.8	14.7	9.8	24.5	4,887	9.8
Total	5.8	35.9	28.2	64		28.2

EPA 430-K-04-003

Using Coal Mine Methane as Fuel to Generate Electricity

Tom McNider at the Jim Walter Resources mining complex in Alabama gave leadership to the developing a strategy in this report for managing and using coal mine methane. The Jim Walter mines are among the gassiest coalmines in the nation and Jim Walter Resources is an industry leader in coalmine methane management, recovery, and utilization. Mr. McNider identified a methane recovery strategy focused on using the volume of methane drained from longwall bleeder and gob wells, where the methane is at a higher concentration than in the main ventilation shaft. With this strategy, methane is removed at higher concentrations in the bleeder vent and gob wells, increasing the potential for total methane recovery from the mine as compared to total methane vented from the main ventilation shafts. This also results in a greater volume of the overall mine methane liberation from the coalseam being at concentrations where the Ramgen engine can use it as a fuel stream. This strategy is also a benefit because the bleeder vent and gob wells are in close proximity to each other making the infrastructure to capture this gas much easier and cost effective compared to other studies. This strategy is economically viable because of the capability of the Ramgen engine to use methane at low concentrations as fuel to produce sufficient returns from the sale of electricity to justify the required investment in this recovery system.

This strategy is different than the approach currently being pursued by the methane recovery and use projects operating today, where the effort is focusing on developing technologies for using methane at concentrations of 1% and less that is currently vented from the main mine shaft. The approach developed for this report involves extensive use of modern degasification technology to bring greater volumes of methane at higher concentrations out of coalmines. In reviewing materials, no other reference to this approach has been found and industry experts consulted were unaware of this approach in operation.

The majority of the methane emitted from underground mining exits the mine through:

- 1) The ventilation system which dilutes the methane in the working areas of a mine to less than 1% methane and emits it into the atmosphere; and,
- 2) The degasification system, which removes methane from non working, parts of the mine to reduce the amounts of methane that have to be diluted and moved

through the ventilation system. Degasification systems can also be used prior to mining by drilling wells to remove a portion of the methane trapped in and around the coal seams.

All coal mines have ventilation systems, which in the U.S. dilute methane to 1% concentration or less, but many mines do not have modern degasification systems. The installation of degasification systems depends on the methane concentration of the mine and, particularly in developing nations, on the investment the owners are willing to make. For the necessary safety of miners, the methane in working areas of the mine is diluted and vented as quickly as possible to avoid explosions. The difficulty of using this methane while continuing to ensure the safety of the working mine, greatly limits the use of this methane.

Because methane recovered from the degasification system is not directly from the working area of the mine it can be collected at higher concentration levels. Methane that collects at closed areas of mines also is at higher concentrations levels.

Technical Capability to Collect Methane

In 1970, some U.S. coal mines with methane at high levels of concentration began to develop their degasification systems to collect high quality methane and deliver it for sale to natural gas pipelines. This expansion of the degasification systems resulted in removing more methane from mines, which both improved safety and produced significant financial returns to the mines. The United States and Jim Walter Resources are world leaders in developing and installing these technologies to collect and monitor methane in mines. The advancements required to collect high concentration gas lead to the development of the capability to collect methane in low concentrations.

Despite the technical capability to do so, there has been very limited economic incentive to capture this low concentration and low quality methane, as there has not been a commercial use for it. By utilizing techniques that are both widely understood and implemented methane-dilution strategies a system was developed where 75% of the methane now being vented into the atmosphere from mines could be utilized to generate electricity. The strategy includes using existing technology to collect methane from non-working areas of the mine at concentration of 3% and higher, and using the projected capabilities of the Ramgen Engine to burn this methane as fuel.

Project Sites Identified for Ramgen Engine

Blue Creek #4

Fan 4-4 on Blue Creek #4 is situated in an area that includes the exhaust shaft for the southern part of the #4 mine. Shaft 4-4 has two fans installed but only one is currently operating. The fan is currently exhausting 890,000 cubic feet per minute of air. The methane content in the air stream is 0.06%. This is 750 thousand cubic feet of pure methane exhausted in a twenty-four hour period. Shaft 4-4 was put into service in 1980. The mining reserves are to the north of the mine. All of the longwall mining near shaft 4-4 is complete and the mined out area is separated from the active works with seals. The

methane content of this shaft is extremely low because of the age of the shaft and the abandoned old work areas are sealed. The mine is currently considering shutting down fan 4-4 if fan 4-5 to the north can adequately ventilate the mine.

Forty-eight wells were evaluated to determine the status of each well and if the well could be produced and at what flow and quality. The majority of these wells were either both plugged and abandoned, or they have a negative well head pressure (pull into the mine). There were only two positive pressure wells out of this group and their flow was estimated at 117 thousand cubic feet per twenty-four hour period at an average concentration of 97.05%. This is 114 thousand cubic feet per day of pure methane, of which, only about 40 thousand cubic feet per day is predicted that could be produced. There were seven wells in this area, which are producing methane for Black Warrior Methane. These wells will become low-quality wells at some future point in time.

Blue Creek #4

Fan 4-5 on Blue Creek #4 is situated in an area that includes the exhaust shaft for the northern reserves of the #4 mine. The 4-5 shaft has two fans operating on it in parallel exhausting approximately 950,000 cubic feet per minute of air per fan. The methane content in the air stream of each fan is 0.45%. This is 6 million cubic feet of pure methane exhausted from each fan in a twenty-four hour period. The 4-5 shaft was put into service in 1982. The mine reserves are to the north of the 4-5 shaft with three sections and one longwall gob ventilated back to this exhaust shaft. Mining is advancing to the north and west of this shaft. The current mine plan shows this shaft being in service as a fan shaft for the life of the mine.

Fifty-seven wells were evaluated to determine the status of each well and if the well could be produced and at what flow and quality. The majority of the wells in this area are positive (natural pressure out of the mine) and could be produced as low quality gob wells. The flow from these wells was estimated at approximately 5 million cubic feet of gas per twenty-four hour period at an average quality of 96.53%. This is 4.8 million cubic feet per day of pure methane, of which, approximately 1.6 million cubic feet per day could be produced on a continual basis. Much of the infrastructure to collect the gas and transport it to the selected site would have to be replaced. In addition, compression units would have to be installed. In this area, the remainder of the non producing gob wells are either both plugged and abandoned, or the well could not be found. Seven Black Warrior production wells will become available for low quality production shortly.

Blue Creek #5

Fan 5-7 on Blue Creek #5 is situated in an area that includes the exhaust shaft for the eastern reserve of the #5 mine. This shaft has two fans operating on it in parallel exhausting approximately 725,000 cubic feet of air per minute per fan. The methane content in the air stream of each fan is 0.66%. This is 6.8 million cubic feet of pure methane exhausted from each fan in a twenty-four hour period. The 5-7 shaft was put into service in 1995. The underground reserves are in the same general area as the shaft with three sections and two longwall gobs ventilated back to it. Mining will advance to the east with 5-9 fan shaft as an exhaust. Depending on the mine plan, 5-7 fan shaft

could be phased out totally as an exhaust shaft within three years.

Eighty-four gob wells were evaluated to determine the status of each well and if the well could be produced and at what flow and quality. The majority of the wells are being produced directly for pipeline quality gas by Black Warrior Methane or the gas is processed by a Nitrogen Rejection Plant and then delivered to the pipeline. The Nitrogen Rejection Plant is processing 5.9 million cubic feet of gas per twenty-four hour period at an average quality of 66.42% methane. The result of this is 3.9 million cubic feet per day from this area that is available for sales. Only nine wells in this area remain that could be produced. The flow from these wells was estimated at 624 thousand cubic feet per twenty-four hour period at an average quality of 90% methane. This is 565 thousand cubic feet per day of pure methane, of which, approximately 188 thousand cubic feet per day is predicted that could be produced on a continual basis. The Nitrogen Rejection Plant will probably utilize most of this gas since the wells are in the general area of the plant. There are two longwall gobs in the 5-7 fan site area that are produced for pipeline gas that will be available for low quality production in the future.

Blue Creek #7

Fan 7-1 on Blue Creek #7 is situated in an area that includes the exhaust shaft for the southern reserves of the #7 mine. This shaft has two fans operating on it in parallel exhausting approximately 760,000 cubic feet of air per minute per fan. The methane content in the air stream of each fan is 0.30% methane. This is 3.3 million cubic feet of pure methane exhausted from each fan in a twenty-four hour period. Shaft 7-1 was put into service in 1980. The reserves are to the west and north of the shaft. There is one continuous mine section operating to the west and two in the north. Two longwall gobs are ventilated to this fan shaft. Mining will be complete in the west in approximately one year and all mining will be in the north. Shaft 7-1 will remain as a fan shaft primarily to ventilate the service and production shaft and main air courses in this area. The methane concentration liberated through the fans would be higher but the leakage is high and therefore dilutes the gas concentration.

Fifty-eight gob wells were evaluated to determine the status of each well and if the well could be produced and at what flow and quality. The majority of the wells in this site have a negative well heads pressure (pull into the mine). There were only two wells with a measurable flow out of the well. 580 thousand cubic feet of gas was measured out of these two wells with an average concentration of 96.56% methane. This is 560 thousand cubic feet of pure methane a day. Approximately 190 thousand cubic feet of methane could be produced from these wells on a continual basis. There are seven wells at this site, produced by Black Warrior Methane. These wells will become available later.

Blue Creek #7

Fan 7-4 on Blue Creek #7 is situated in an area that includes the exhaust shaft for the northern reserves of the #7 mine. This shaft has two fans operating on it in parallel exhausting approximately 800,000 cubic feet of air per minute per fan. The methane content in the air stream of each fan is 0.5% methane. This is approximately 5.8 million cubic feet of pure methane exhausted from each fan in a twenty-four hour period. Shaft

7-4 was put into service in 1980. The 7-6 shaft will become an exhaust shaft and replace the fans at the 7-4 shaft in approximately one year.

Thirty-one wells were evaluated to determine the status of each well and if the well could be produced and at what quality. The majority of the wells surveyed has either negative well-head pressure (pressure into the mine) or is producing gas for the Nitrogen Rejection Plant. 900,000 cubic feet per day of flow was measured from the wells producing for the Nitrogen Rejection Plant at an average concentration of 69.7% methane. This is 628,000 cubic feet per day of pure methane. No flow could be measured from the remaining wells in the area. There are four wells in the area being produced by Black Warrior Methane that will become available for low quality production at a future time.

Description of the Capability of the Ramgen Engine to Reduce Greenhouse Gas

Description of Ramgen Engine

The Ramgen Engine is projected to have compelling advantages using coalmine methane emissions as fuel to generate electricity compared with other technological options that include the following:

- 1) The Ramgen Engine can efficiently use the coalmine methane at low concentrations as fuel. This is achieved by using the coalmine methane as the engine's combustion air. In the Ramgen engine the combustion air and the fuel are pre-mixed, compressed and combusted instantaneously, methane in the combustion air is simply added to the fuel.
- 2) The Ramgen Engine is projected to have low emissions of NO_x; this includes situations when operating on fuel with less than 80% methane content. Reciprocating internal combustion engines have high NO_x levels in general. Gas turbines have very high NO_x levels when the methane content is less than 80% as would generally be the case with methane from coalmines. Solving the problem of high NO_x, when it is technically possible, is expensive and this is a critical advantage for the Ramgen engine in many geographic areas with strict emission standards. Air quality regulations are becoming stricter in many markets around the world, making this advantage increasingly important.
- 3) The Ramgen Engine does not require pre-compression of fuel. Gas turbines require the pre-compression of fuel and, consequently, have additional capital costs and operating expenses. Pre-compression of fuel becomes dangerous when methane content is below 45%±, which adds to the expense of solving the problem (by treating the gas to remove the contaminants).
- 4) The Ramgen Engine can operate at partial loads without losing significant efficiency, while gas turbines cannot. This is a critical advantage on a coal mine as the volume of methane and its concentration changes regularly for reasons ranging from mine operations to air pressure.
- 5) The Ramgen Engine is less susceptible to damage from impurities in the fuel stream than other engines due to reduced combustion surface area in contact with the impurities. In addition, replacement and repairs to the Ramgen Engine caused by impurities in fuel are, relative to other engines, simpler and less expensive to repair.

- 6) The Ramgen Engine has fuel flexibility, high efficiency, low emissions, low maintenance costs and low capital costs. Other options for using coal mine methane as fuel do not combine all five of these characteristics.

Approach to Using Methane from Coal Mines

Because so much of the methane currently released from coalmines, up to 90%, is in the very dilute ventilation air, research efforts to use this methane have focused on technologies, which can make use of methane at concentrations of 1% or less. In the very different approach outlined in this Report the focus is on a greatly increased use of degasification systems to remove the majority of methane from mines at higher concentration levels.

Two factors are important to this approach for using large volumes of this greenhouse gas. The first factor is the capability of the Ramgen engine to generate electricity fueled with methane at concentrations as low as 3%. The second is the emergence of very strong economic incentives for mines to install the expensive modern degasification systems.

The outlined approach involves installing modern degasification systems that would remove 75% of the methane in mines at concentrations of 3% or higher. These systems would improve mine safety because the percentage of methane removed from the active area of the mine via the main ventilation system is reduced dramatically to only 25%. Assumptions made in developing the approach in this Report include the following:

- 1) Very strong economic incentives will be needed to drive the installation of the degasification systems required for widespread use of low and medium quality coalmine methane. The required degasification technology is expensive and in general there is legitimate reluctance in the coal mining industry to even consider making changes in current practices because of the priority which must be given to safety;
- 2) The economic incentives to drive both the installation of degasification technology to capture the low quality coalmine methane and the Ramgen technology to use it would be from the sale of: a) electricity from Ramgen engines, and b) carbon credits. These revenues would create strong incentives for mines to fully apply existing degasification technology, further refine systems to monitor and safely collect gas from bleeder vents and install Ramgen Engines;
- 3) The projected capabilities of the Ramgen engine would result in electricity from the waste gas costing \$.015 kW or less to produce, this assumes no charge for the methane with the mine making it's profit from the sale of electricity and of carbon credits, and;
- 4) The base value for the sale of carbon credits from coal mines is valued at \$5.52 a MTCO₂E.

Approach to Collecting Methane from Coal Mines Suitable for Use by Ramgen Engine as Fuel

Using the above assumptions, 75% of the total methane released from coalmining

worldwide can be captured and used as a fuel steam for the Ramgen engine. The data used to estimate the methane emitted annually by coal mines has been collected primarily from government sources in the largest coal producing countries. These data sources are the most current and reliable available. This hypothetical approach was developed with assistance of industry experts in coalmine methane recovery.

The approach involves the use of a combination of methane from degasification systems including Gob wells, Bleeder vents, and wells at sealed areas of mines. Industry analysts estimate that sealed areas of active coalmines hold an additional supply of methane equal to 20% of the annual methane liberated from the coal seam. The methane in the sealed areas is not included in official estimates of the methane emitted from coal mining, but it does eventually escape the sealed area and contribute to methane accumulation in the atmosphere and has therefore, been included in this Report.

Coalmine ventilation experts believe if there is the economic incentive to apply modern ventilation techniques the following can be achieved:

1) Installing Bleeder vents. By making full use of this technology, 25% of the total methane from the mine can be extracted through bleeder vents at a 3% methane concentration. Operating primarily on this sole source of fuel the Ramgen engines would run at 60% of full power. If it proves impractical to have the bleeder vent methane at 3% an alternative that would produce significant results would be to use a combination of bleeder vent methane at 2% and gob well methane at higher concentrations, but for this Report it is assumed the methane from the bleeder vents, with full application of modern technologies, can be run safely at 3%;

2) Installing Gob wells and wells at Sealed Areas of active mines. By using a combination of the methane released from the Gob wells, and methane from wells at sealed areas, 50% of the mine methane can be gathered into a fuel source to operate the Ramgen engine. The methane from the sealed areas is estimated to be collectible at 90% concentrations during early stages of collection and decrease to 50% concentration in late stages of collection.

In this approach, the methane drainage system of the mine has been optimized to decrease the methane vented through the main ventilation shaft from as high as 90% to 25% of the total methane released from the coal mine. While the highly dilute ventilation methane has not been utilized in this scenario, in practice a portion of the methane released in the main ventilation system could also be utilized by the Ramgen engine as combustion air.

2) Description of Greenhouse Gas Reductions

Greenhouse Gas Reductions from Installing Bleeder Vents

The scenario assumes that the capability of the Ramgen engine to use the 3% methane in the bleeder vent exhaust as the primary fuel for the engine to generate electricity and the economic incentive of carbon credits will provides the economic incentive for more mines to install extensive bleeder ventilation systems and to safely operate those systems

at 3% methane, while Bleeder vent systems improve the overall coalmine safety, as mines install bleeder systems additional attention must be focused on engineering designs to run the bleeder vents with methane content at 3%.

An example of the significance of this approach is that if the technology for bleeder vents and the Ramgen Engine had been in place at coal mines in the 2000 the following would have been achieved:

- 1) The bleeder vents would have emitted 25% of the total methane liberated from the coal seam; and,
- 2) An estimated 118 MMTCO₂E of methane would have been used as fuel for the Ramgen to generate electricity.

Under this scenario by 2020 an estimated 147 MMTCO₂E of methane vented from coal mines would be captured annually from bleeder vents and used for the generation of energy.

Greenhouse Gas Reductions from Installing Gob Wells

An example of the significance of this approach is that if the technology for collecting methane from gob wells and from wells at sealed areas of mines and using the methane as fuel for the Ramgen Engine had been in place at coal mines in the 2000 the following would have been achieved:

- 1) The gob wells would have emitted 50% of the total methane liberated from the coal seam; and,
- 2) An estimated 237 MMTCO₂E of methane would have been used as fuel for the Ramgen to generate electricity.

By 2020 an estimated annual 294 MMTCO₂E of methane emitted from coalmines would be captured and destroyed by generating energy through the Ramgen engine from these sources. An additional 95 MMTCO₂E of methane in 2000 and 118 MMTCO₂E of methane in 2020 trapped in sealed areas of the mines can be used as fuel. The eligibility of this methane to qualify for carbon credits is not confirmed, but methane trapped in sealed areas of the mine does eventually seep out of the mine and into the atmosphere. Many experts believe that it will be judged eligible when the carbon credit criteria are fully developed. For the purpose of this report the kWh generated by utilizing this methane are counted, while the value of the carbon credits are not included.

Methane in the sealed areas of active mines has been freed from the coalbed during mining. Coalmine experts estimate that the closed areas of active mines can contain as much as an additional 20% of the total methane generally estimated to be liberated from the coal seam. In most cases, this methane is at a concentration level of 40-60%.

Greenhouse Gas Reductions from Installing Bleeder Vents and Installing Gob Wells

The total reduction of coalmine methane emissions from bleeder vents, gob wells, and the sealed areas of active coalmines equals 450 MMTCO₂E in 2000 and 559 MMTCO₂E in 2020. The application of the Ramgen engine in this scenario would exceed the low-EPA estimated methane emission reduction to halt the atmospheric build-up of methane set at

319 MMTCO₂E per year and would equal 70% of the high-EPA estimate of 798 MMTCO₂E per year.

For coal mine operators the economic incentive to install this technology would have equaled \$4,831,280,418 per year in 2000 and \$5,997,991,569 per year in 2020 from the combined sales of electricity⁵² and carbon credits earned⁵³ by utilizing and preventing the methane in bleeder vents, gob well, and sealed areas of active coalmines from entering the atmosphere.

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⁵² Assumes \$0.02 profit per kWh

⁵³ \$5.52 per MTCO₂e

LANDFILL METHANE

Background on Methane Emitted from Landfills

The majority of information in the Report on methane from landfills was in an earlier report to DOE from Ramgen. That information has been summarized here and updated. It is included since the use of landfill methane is a major source of credits in the developing carbon markets. A very detailed description of landfill sites in the United States with methane emissions has also been added and included in the appendix to this Report.

In the United States 33% of methane emitted into the atmosphere from human activities comes from landfills. Worldwide, while the United States emits the largest volume of methane, emissions from landfills in developing nations are increasing rapidly due to the affects of population growth and industrialization. China now ranks second to the US in landfill emissions. Landfill emissions are considered a major potential source of reductions in greenhouse gases, but economically viable systems have not yet been developed to drive collecting and using this methane.

Methane is produced and emitted from the anaerobic decomposition of organic material in landfills. The total emissions of greenhouse gases from landfills is determined by the volume of organic material contained in a landfill, the extent of the anaerobic decomposition, and volume of methane collected for either power generation or flaring. The EPA has current totals for methane emissions from landfills in the U.S. and worldwide totals from 1993. The projected future emissions of landfill methane are estimated by using the current EPA numbers for the U.S. and then applying the 1993 ratio of methane emissions from the U.S. compared to the world total.

The estimate for world totals is conservative due to the declining total landfill methane emission in the U.S. and the rising total landfill methane emissions from the developing world. Based on these estimates and the capabilities of the Ramgen Engine a scenario is described in which the introduction of the Ramgen engine results in greenhouse gas reductions. As was the case with using coalbed methane, in order for a large number of Ramgen engines to be installed at landfills there needs to be strong economic incentives in addition to the capital and operating cost advantages of the Ramgen products. In this analysis, these incentives would come from the sale of electricity and of carbon credits.

Worldwide, landfills are estimated to produce up to 1,470 MMTCO₂E of methane per year. Two-thirds of the total emission of methane from landfills worldwide comes from developed countries, and eleven countries are estimated to produce 70% of the total emission of landfill methane. In existing U.S. landfills methane recovery systems can capture between 50% and 85% of the total methane produced at a landfill. In landfills now being constructed, nearly 100% methane capture can be achieved. Over the next 20 years, an increasing percent of methane can be captured, as it will be coming from the newer landfills with better gathering systems. Landfill-methane emission recovery options create interest both because of both the potential for energy generation and the reduction of methane and other harmful gases into the atmosphere.

In the United States, and much of the developed world, large modern landfills are required by regulation to collect and destroy the gases being emitted, which include methane. At landfills where collection systems are required, there is stronger economic justification to use the gas to generate electrical power since the investment already is in place to gather the emissions. However, if engines to generate electricity using these emissions that can be sold profitably are not available, then the gas that is captured will be flared. At most landfills this is currently the case. Since flaring would also justify receiving carbon credits, generating electricity itself has to produce profit for the operator.

An advantage for generating electricity at landfills is that many landfills are located in close proximity to urban areas with large power demands and access to the power grid. For older and/or smaller landfills in the U.S. that are not required by regulation to install collection systems for the gas emissions and for landfills in less regulation nations the revenues from power and carbon credit sales has to be sufficient to justify the cost of the collection system as well as the electrical generating system.

Efforts to Establish Commercial Use for Landfill Methane

There are many examples of landfill-methane recover projects in operation in both the developed and the developing world. However, there is not a solid estimate of the total volume of landfill methane actually being utilized on an on-going basis. The USEPA estimates that substantial opportunities exist for methane recovery and utilization projects. In the United States, for example there are 200 current landfill-methane recovery plants and EPA lists an additional 700 strong candidates for primary electricity generation. The extent to which the current 200 projects produce power on a regular basis is not clear. Currently there is only limited recovery of landfill methane internationally and the USEPA identifies Brazil, Russia, Eastern Europe, India, and China as all having substantial opportunity to expand their recovery and use of landfill methane.

The collection system for landfill gases, which includes methane, is not technically complicated and the very serious safety issues that are a priority with coalmining are not present, as people are not working underground. Vertical wells are drilled into the landfill spaced, depending on the landfill, between 150 to 300 feet apart. Horizontal trenches are also used in some landfills but are less durable. All of the wells and/or trenches are connected by horizontal piping to a central point where a motor/blower provides a vacuum to remove the gas from the landfill.

TABLE A. Total Volume of Methane Released from Landfills Worldwide
International Landfill Methane Emissions - 1993

Country	MMTCO₂E – Low	MMTCO₂E - High
United States	616	921
United Kingdom	77	231
Brazil	54	168
India	15	61

Poland	8	31
Others	848	3,005
Total	1,618	4,392

(EPA-430-R-93-012)

The following chart reflects the total estimated United States emissions in MMTCO₂E per year of methane for 2000-2020.

**TABLE B. United States Landfill Projected Baseline Methane Emissions
Projected Methane Generated at Landfills in the United States**

Year	2000	2005	2010	2015	2020
	MMTC O₂E	MMTC O₂E	MMTC O₂E	MMTC O₂E	MMTC O₂E
Municipal Solid Waste Landfill	304	319	317	300	277

(EPA-430-R-93-012)

Description of Greenhouse Gas Reductions Resulting from Use of the Ramgen Engine

Using the Ramgen engine with methane emissions from landfills has similarities as well as differences to methane emissions from coalmines. In both cases the technology to gather the methane has been developed and what is preventing the widespread use of the methane as fuel is not having an engine or other power source with the realistic prospect of producing sufficient profit to justify the economic risk of investment. The engines that currently do operate using the methane have difficulties with emissions and with reliability of long term power output.

The economics of using landfill methane to generate electricity can be projected to change dramatically over the next 3 to 5 years. The introduction of the Ramgen engine will provide a more economically viable power plant for this fuel and the development of a carbon credit trading system may provide additional economic incentives.

The inclusion of landfill methane emission in an international carbon credit trading system is not as clear cut as with coal mine methane. The uncertainty of obtaining carbon trading credits for captured landfill methane emissions is due to the fact the some nations, including the U.S., which already regulate emissions from some landfills. It is possible that the final criteria for carbon credits will specify that if there is a regulation that already requires that methane be destroyed it will not qualify as a carbon credit for trading.

While this uncertainty exists with carbon credits, there are strong indications that most landfill methane will qualify. In fact some sales have already occurred. As cited in Part A of this Report, Ontario Power Generation, a Canadian utility, has purchased credits from US Gen, a subsidiary of the U.S.-based PG&E National Energy Group. US Gen created reductions by capturing and destroying methane produced at a landfill.

The Pew Center on Global Climate Change lists three examples where landfill projects are participating in transactions of emissions reductions and carbon credits, two projects in Canada and one in the United States. The Prototype Carbon Fund of the World Bank has identified numerous landfill-gas energy projects internationally and has accredited the reduction of methane gas utilized by the projects.

In total, there are over 3,000 landfills in the United States with over a million tons of solid waste and a much larger number internationally. The estimates done by many organizations on methane recovery from landfills have not included the completely different economic dynamic that would emerge if this methane does qualify for carbon credits and the Ramgen Engine is commercially available.

Assumptions Used in Developing Description of Greenhouse Gas Reductions

In developing a scenario that provides an overview of the greenhouse gas reductions resulting from the introduction of the Ramgen engine with landfill methane the most significant assumptions included the following:

- 1) Significant economic incentives, from the sale of electricity from Ramgen engines and of carbon credits, will be the driving force for landfill operators to collect and use methane to generate and sell electricity;
- 2) The projected capabilities of the Ramgen engine would result in electricity from the waste gas costing \$0.02 kWhr to produce, this assumes no charge for the methane with the landfill operator making its profit from the sale of electricity and carbon credits; and,
- 3) The value of carbon credits is set at \$5.22 per ton of CO₂E.

The objective of this description is to outline the general reasons why the prospects of reducing greenhouse gases by using landfill methane as fuel for the Ramgen engine warrants serious consideration and a more detailed study of implementation.

Projected Greenhouse Gas Reductions

The most complete data located for landfill methane is for the United States. The emission of the United States in 1993 was equal to between 616 and 921 MMTCO₂E. The average of these is equal to approximately 768 MMTCO₂E per year. This data has been used as a means to make basic projections using the Ramgen engine on landfill methane. Additionally, the projected percentage of global landfill methane emission for the United States has been applied to USEPA landfill-methane emission estimates for a rough determination of the total global landfill methane emitted per year. Using the approach identified here it is estimated the Ramgen engine, assuming the development of a carbon credit trading system that includes landfill methane, would result in a 35% reduction in emissions world wide and a 50% reduction in the United States.

For purpose of example, under the above assumptions the total amount of methane reduced would have equaled 423 MMTCO₂E in 2000. In 2020, the reduction would be 375 MMTCO₂E per year.

Financial Incentives to Install Technology

For the purpose of example, using data for the United States from 2000 and the percentage of worldwide landfill methane emissions the United States contributed in 1993, the total amount of electricity produced from landfill gas globally by the Ramgen engine could have been 110 Billion kWhrs in 2000. In 2020, this would total 97 Billion kWhrs and 375 MMTCO₂E per year. The estimated revenue from sale of electricity, after operating cost, to the landfill operators would be \$2,204,204,081 in 2000 and \$1,954,142,782 in 2020.

The total amount of methane reduced would equal 423 MMTCO₂E in 2000 and 375 MMTCO₂E in 2020. At a value of \$5.22 per ton of carbon dioxide equivalence, the value to landfill operators of credits would be \$2,208,060,000 in 2000 and \$1,957,500,000 in 2020.

The total value of electricity generated and methane destroyed would have equaled \$4,412,264,081 in 2000 and could equal \$3,911,642,782 by 2020

PART D

Rampressor Turbine in Hybrid Configuration with Fuel Cell

Ramgen is currently working on a development project in conjunction with the University of Alaska Fairbanks with support from DoD with the Project Goal: To demonstrate a fuel cell turbine hybrid designed to achieve the highest electrical efficiency possible by using the innovative high efficiency compression of the Ramgen technology matched to the best available hardware, including heat exchanger and fuel cell.

Based on computer modeling, efficiencies of 70% are possible in a fuel cell/Rampressor Turbine hybrid with a correctly designed interface. The limited number of hybrid-systems built to date, have not achieved the goal of very high and reliable efficiencies because of the lack of a suitably sized turbine and problems associated with the pressurized fuel cell design. The problems with the fuel cell/turbine interface included the mismatch between the available turbine and fuel cell components.

Work is now in progress to design: 1) a Rampressor Turbine Engine suited for use in a hybrid configuration with a fuel cell; and, 2) an effective interface for the Rampressor Turbine Engine with a selected fuel cell system

The Objectives that will be accomplished by the design include:

- 1) Complete fuel cell hybrid cycle analysis as basis to complete interface design;
- 2) Identification of optimized system performance capabilities utilizing high performance fuel cells, and make final decision on fuel cell for this project (this work will include on site reviews of the limited number of hybrid systems built to date);
- 3) Design of Rampressor Turbine Engine suited for use in hybrid configuration with fuel cell and the interface designed for this project; and,
- 4) Design of the Rampressor-Turbine super-sonic technology engine/fuel cell hybrid system.

Following the completion of the design the project plan is to:

- 1) Build a turbine system using the Ramgen shock wave compression matched with conventional combustion and turbine components to demonstrate the high efficiency achievable with this technology;
- 2) Integrate the fuel cell and turbine in breadboard configuration to address system integration and control system issues; and,
- 3) Demonstrate full scale system at military base in Alaska.

The Rampressor Turbine Engine is particularly well suited for use in hybrid configuration with fuel cell and the interface designed for this project. The Rampressor Turbine engine utilizes super-sonic inlet technology to provide a compressor that can reach desired pressure ratio in one stage with superior simple cycle efficiency.

The Rampressor Turbine concept is a 4:1 pressure ratio Rampressor compressor with an

industrially-designed combustor and turbine. Historically the fuel cell hybrid demonstrations have suffered from not optimizing the turbo machinery to match the flow requirements of the fuel cell. Ramgen will optimize the sub-components of the gas turbine engine, namely, the compressor, combustor, and turbine to contribute directly to the improvement of the overall system efficiency.

The projected improvement in the Rampressor efficiency over existing industrial compressors will result in an overall improvement in the integrated Fuel Cell/Rampressor Turbine hybrid system efficiency. When considering the “direct” configuration, the high Rampressor II efficiency will result in an overall increase in the gas turbine efficiency and a resulting increase in the integrated hybrid system. The “indirect” configuration will result in greater flexibility and optimization of the turbo machinery subsystems.

Extensive computer modeling has indicated that very high electrical generation efficiencies (up to 70%) can be achieved by combining fuel cells and turbines in hybrid systems. High temperature fuel cells provide efficiencies of nearly 50% when operating on natural gas, and provide high grade heat for operating a turbine in a hybrid configuration, resulting in very high electrical efficiencies. In addition, the projected low capital and operating costs of the Rampressor Turbine are projected to reduce the current high cost per kW of fuel cells.

The goal of this project is to demonstrate a highly efficient technology on a military base in Alaska that can be developed in a form that is economically viable to implement at sites on a large scale in the United States and throughout the world.

APPENDIX

1. Rampressor

Rampressor Methodology

Electrical Power Savings – Oil-free Air Compression

1. Consumption of kWh for Rampressor/ year

Consumption of kWh/ year for Rampressor = (Efficiency of Compression – BHP) (Size of Unite - Average cfm/ 100) (hp/ kWh Conversion) (Operational Hours/ year) (Number of Rampressor Units Operating)

2. Consumption of kWh for Industry without Rampressor/ year

Consumption of kWh/ year for Industrial Standard Oil-free Air Compressor = (BHP – Industry Average/ by the BHP Rampressor) X Savings of kWh/ year for Rampressor

3. kWh Savings and Percentage of kWh Savings of Rampressor vs. Industry Standard

Total kWh Savings = (Industrial Standard Oil-free Air Compression – Consumption of kWh/ year) – (Rampressor – Consumption of kWh/ year)

Percentage of kWh Savings = kWh savings for Rampressor/ kWh consumption of Industry Standard Unit

Combined Heat and Compression/ Gas-Fired Rampressor – Oil-free Air Compression

4. Heat Recovery for Rampressor

Total kWh of Recoverable Heat = Total kWh consumption for Rampressor units * 50% of Rampressor units operating where use for recoverable heat exists and is economically viable * *70% of Rampressor units operating where CHC is viable but cogeneration with gas turbine is not* * 60% heat recovery * 80% utilization of recovered heat

5. Gas Fired Rampressor

Total kWh of Recoverable Heat = Total kWh consumption for Rampressor units * 50% of Rampressor units operating where use for recoverable heat exists and is economically viable * *30% of Rampressor units operating where CHC and cogeneration with gas turbines is viable* * **80% heat recovery** * 80% utilization of recovered heat

6. Net kWh Savings with CHC

Net kWh Savings for Rampressor = (Industry kWh Consumption) - (Rampressor kWh Savings + Rampressor CHC kWh Recovery + Gas Fired Rampressor kWh Recovery)

Percentage of Rampressor Net Savings = (kWh Savings + Rampressor CHC kWh Recovery + Gas Fired Rampressor kWh Recovery)/ Industry kWh Consumption

7. \$ Savings per Unit per Year

(Efficiency – Industry) (Average cfm/100) (0.746 kW/hp) (% savings w/Rampressor) (1 year usage) (cost of electricity \$/kWh) = \$ savings of Rampressor unit/year

Example: 500-2500 cfm market

(24 bhp/100) (15) (0.745 kW/hp) (0.125) (8000 hrs) (0.07 kWh) = \$18,799
kWh/year = 268,557

8. CO2 Savings per Unit per Year

(kWh/year) (1.64 CO2 – US national average CO2 generated per kWh) = total CO2

Example: 500-2500 cfm market

(268,557 kWh) (1.64 CO2) = 440,433.7 CO2

9. Number of Total Units Replaced per Year

Total HP Sold/Average HP in Size Category/2 (50% market penetration)

Example: 500-2500 cfm market 2004

463710 cfm/ 362 hp/ 2 = 640 units replaced in 2004

10. Total \$ Savings per Year

(Total Rampressors Operating) (\$ Savings per Unit) = Total \$ Savings per Year

Example: 500-2500 cfm market 2004

(640) (\$18,799) = \$12,023,959

11. Total CO2 Reductions per Year

(Total kWh Savings) (1.64 CO2) = Total CO2 Reductions per Year

Example: (171770838.8 kWh) (1.64 CO2) = 281704175.7 CO2

[illegible]

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Industrial Electricity Consumption

Fuel Consumption,
1998;
Level: National and
Regional Data;
Row: NAICS Codes;
Column: Energy
Sources;
Unit: Physical
Units or Btu.

NAICS		Total	Net Electricity(b)	Electricity
Code(a)	Sub-sector and Industry	(kWh)	(kWh)	(% Total)
Total – United States				
311	Food	305,996,400,000	62,457,000,000	20.41%
311221	Wet Corn Milling	50,706,300,000	7,060,000,000	13.92%
312	Beverage and Tobacco Products	31,654,800,000	7,122,000,000	22.50%
313	Textile Mills	74,447,400,000	29,759,000,000	39.97%
313210	Broadwoven Fabric Mills	20,810,100,000	10,917,000,000	52.46%
314	Textile Product Mills	14,361,900,000	5,152,000,000	35.87%
315	Apparel	14,068,800,000	5,251,000,000	37.32%
316	Leather and Allied Products	2,344,800,000	761,000,000	32.45%
321	Wood Products	147,722,400,000	21,170,000,000	14.33%
321113	Sawmills	48,947,700,000	5,882,000,000	12.02%
321114	Wood Preservation	3,517,200,000	212,000,000	6.03%
3212	Veneer, Plywood, and Engineered Woods	64,482,000,000	10,556,000,000	16.37%
3219	Other Wood Products	30,775,500,000	4,519,000,000	14.68%
322	Paper	804,266,400,000	70,364,000,000	8.75%
322110	Pulp Mills	58,033,800,000	1,371,000,000	2.36%
322121	Paper Mills, except Newsprint	337,651,200,000	25,582,000,000	7.58%
322122	Newsprint Mills	51,292,500,000	14,741,000,000	28.74%
322130	Paperboard Mills	288,410,400,000	15,073,000,000	5.23%
323	Printing and Related Support	28,723,800,000	14,946,000,000	52.03%
323110	Commercial Lithographic Printing	15,827,400,000	8,136,000,000	51.40%
324	Petroleum and Coal Products	1,061,608,200,000	37,059,000,000	3.49%
324110	Petroleum Refineries	1,019,108,700,000	34,725,000,000	3.41%
325	Chemicals	1,085,642,400,000	169,233,000,000	15.59%
325110	Petrochemicals	161,205,000,000	2,367,000,000	1.47%
325120	Industrial Gases	53,637,300,000	32,578,000,000	60.74%
325181	Alkalies and Chlorine	43,085,700,000	14,602,000,000	33.89%
325188	Other Basic Inorganic Chemicals	79,723,200,000	31,490,000,000	39.50%
325192	Cyclic Crudes and Intermediates	23,448,000,000	2,063,000,000	8.80%
325199	Other Basic Organic Chemicals	300,427,500,000	21,335,000,000	7.10%
325211	Plastics Materials and Resins	115,481,400,000	19,223,000,000	16.65%

325212	Synthetic Rubber	31,361,700,000	2,444,000,000	7.79%
325222	Noncellulosic Organic Fibers	30,775,500,000	7,121,000,000	23.14%
325311	Nitrogenous Fertilizers	73,568,100,000	3,897,000,000	5.30%
325312	Phosphatic Fertilizers	17,879,100,000	1,394,000,000	7.80%
326	Plastics and Rubber Products	95,843,700,000	53,517,000,000	55.84%
326199	Other Plastics Products	29,603,100,000	20,774,000,000	70.18%
327	Nonmetallic Mineral Products	284,013,900,000	39,277,000,000	13.83%
3272	Glass and Glass Products	59,499,300,000	12,297,000,000	20.67%
327310	Cements	104,050,500,000	11,362,000,000	10.92%
331	Primary Metals	755,025,600,000	159,626,000,000	21.14%
331111	Iron and Steel Mills	490,063,200,000	46,350,000,000	9.46%
331112	Electrometallurgical Ferroalloy Products	8,793,000,000	0	0.00%
3312	Steel Products from Purchased Steel	15,827,400,000	5,020,000,000	31.72%
3313	Alumina and Aluminum	129,257,100,000	72,028,000,000	55.72%
331312	Primary Aluminum	62,137,200,000	57,355,000,000	92.30%
3315	Foundries	68,292,300,000	18,323,000,000	26.83%
331511	Iron Foundries	29,310,000,000	10,335,000,000	35.26%
331521	Aluminum Die-Casting Foundries	5,862,000,000	1,605,000,000	27.38%
331524	Aluminum Foundries, except Die-Casting	5,568,900,000	1,225,000,000	22.00%
332	Fabricated Metal Products	129,257,100,000	51,499,000,000	39.84%
333	Machinery	62,430,300,000	28,178,000,000	45.14%
334	Computer and Electronic Products	60,085,500,000	40,207,000,000	66.92%
334413	Semiconductors and Related Devices	19,344,600,000	13,372,000,000	69.13%
335	Electrical Equip., Appliances, and Components	33,999,600,000	16,216,000,000	47.69%
336	Transportation Equipment	143,032,800,000	57,223,000,000	40.01%
337	Furniture and Related Products	25,792,800,000	8,646,000,000	33.52%
339	Miscellaneous	25,792,800,000	11,811,000,000	45.79%
	Total	5,186,404,500,000	889,474,000,000	17.15%
	Northeast Census Region			
	RSE Column Factors:	263,790,000	900,000	0.34%
311	Food	22,861,800,000	5,740,000,000	25.11%
311221	Wet Corn Milling	0	10,000,000	
312	Beverage and Tobacco Products	3,517,200,000	690,000,000	19.62%
313	Textile Mills	9,379,200,000	1,942,000,000	20.71%
313210	Broadwoven Fabric Mills	1,465,500,000	427,000,000	29.14%
314	Textile Product Mills	1,172,400,000	0	0.00%

315	Apparel	3,810,300,000	920,000,000	24.15%
316	Leather and Allied Products	586,200,000	0	0.00%
321	Wood Products	7,034,400,000	1,329,000,000	18.89%
321113	Sawmills	1,758,600,000	319,000,000	18.14%
321114	Wood Preservation	293,100,000	-50,000,000	-17.06%
3212	Veneer, Plywood, and Engineered Woods	2,344,800,000	550,000,000	23.46%
3219	Other Wood Products	2,344,800,000	509,000,000	21.71%
322	Paper	101,412,600,000	9,866,000,000	9.73%
322110	Pulp Mills	0	0	
322121	Paper Mills, except Newsprint	78,843,900,000	4,837,000,000	6.13%
322122	Newsprint Mills	7,034,400,000	0	0.00%
322130	Paperboard Mills	7,620,600,000	1,280,000,000	16.80%
323	Printing and Related Support	5,568,900,000	2,711,000,000	48.68%
323110	Commercial Lithographic Printing	2,931,000,000	1,418,000,000	48.38%
324	Petroleum and Coal Products	94,378,200,000	3,253,000,000	3.45%
324110	Petroleum Refineries	88,223,100,000	2,820,000,000	3.20%
325	Chemicals	50,120,100,000	13,397,000,000	26.73%
325110	Petrochemicals	0	0	
325120	Industrial Gases	0	0	
325181	Alkalies and Chlorine	2,931,000,000	0	0.00%
325188	Other Basic Inorganic Chemicals	3,810,300,000	0	0.00%
325192	Cyclic Crudes and Intermediates	0	182,000,000	
325199	Other Basic Organic Chemicals	4,689,600,000	0	0.00%
325211	Plastics Materials and Resins	4,396,500,000	1,255,000,000	28.55%
325212	Synthetic Rubber	1,465,500,000	0	0.00%
325222	Noncellulosic Organic Fibers	586,200,000	0	0.00%
325311	Nitrogenous Fertilizers	586,200,000	0	0.00%
325312	Phosphatic Fertilizers	0	0	
326	Plastics and Rubber Products	16,413,600,000	8,912,000,000	54.30%
326199	Other Plastics Products	4,982,700,000	3,432,000,000	68.88%
327	Nonmetallic Mineral Products	41,620,200,000	6,153,000,000	14.78%
3272	Glass and Glass Products	14,655,000,000	2,449,000,000	16.71%
327310	Cements	13,775,700,000	1,567,000,000	11.38%
331	Primary Metals	74,154,300,000	16,747,000,000	22.58%
331111	Iron and Steel Mills	47,775,300,000	0	0.00%
331112	Electrometallurgical Ferroalloy Products	0	0	
3312	Steel Products from Purchased Steel	2,931,000,000	0	0.00%
3313	Alumina and Aluminum	10,258,500,000	0	0.00%
331312	Primary Aluminum	0	0	
3315	Foundries	5,275,800,000	0	0.00%
331511	Iron Foundries	0	0	

331521	Aluminum Die-Casting Foundries	879,300,000	0	0.00%
331524	Aluminum Foundries, except Die-Casting	0	91,000,000	
332	Fabricated Metal Products	20,223,900,000	6,939,000,000	34.31%
333	Machinery	9,086,100,000	4,448,000,000	48.95%
334	Computer and Electronic Products	13,775,700,000	8,205,000,000	59.56%
334413	Semiconductors and Related Devices	4,689,600,000	2,807,000,000	59.86%
335	Electrical Equip., Appliances, and Components	5,568,900,000	2,899,000,000	52.06%
336	Transportation Equipment	12,896,400,000	3,882,000,000	30.10%
337	Furniture and Related Products	2,637,900,000	0	0.00%
339	Miscellaneous	6,155,100,000	0	0.00%
	Total	502,080,300,000	102,210,000,000	20.36%
		0	0	
	Midwest Census Region	0	0	
		0	0	
	RSE Column Factors:	234,480,000	800,000	0.34%
		0	0	
311	Food	136,877,700,000	25,098,000,000	18.34%
311221	Wet Corn Milling	44,551,200,000	6,018,000,000	13.51%
312	Beverage and Tobacco Products	5,862,000,000	1,386,000,000	23.64%
313	Textile Mills	1,465,500,000	463,000,000	31.59%
313210	Broadwoven Fabric Mills	0	37,000,000	
314	Textile Product Mills	0	0	
315	Apparel	879,300,000	281,000,000	31.96%
316	Leather and Allied Products	0	0	
321	Wood Products	14,068,800,000	2,922,000,000	20.77%
321113	Sawmills	2,051,700,000	358,000,000	17.45%
321114	Wood Preservation	293,100,000	34,000,000	11.60%
3212	Veneer, Plywood, and Engineered Woods	7,620,600,000	1,128,000,000	14.80%
3219	Other Wood Products	4,103,400,000	1,402,000,000	34.17%
322	Paper	134,532,900,000	16,666,000,000	12.39%
322110	Pulp Mills	7,327,500,000	291,000,000	3.97%
322121	Paper Mills, except Newsprint	73,275,000,000	8,424,000,000	11.50%
322122	Newsprint Mills	1,758,600,000	0	0.00%
322130	Paperboard Mills	20,223,900,000	2,847,000,000	14.08%
323	Printing and Related Support	10,551,600,000	5,305,000,000	50.28%
323110	Commercial Lithographic Printing	6,155,100,000	3,066,000,000	49.81%
324	Petroleum and Coal Products	166,187,700,000	7,283,000,000	4.38%
324110	Petroleum Refineries	157,101,600,000	6,420,000,000	4.09%
325	Chemicals	134,532,900,000	39,041,000,000	29.02%
325110	Petrochemicals	0	0	
325120	Industrial Gases	8,793,000,000	0	0.00%
325181	Alkalies and Chlorine	586,200,000	0	0.00%

325188	Other Basic Inorganic Chemicals	15,534,300,000	0	0.00%
325192	Cyclic Crudes and Intermediates	6,448,200,000	0	0.00%
325199	Other Basic Organic Chemicals	21,689,400,000	0	0.00%
325211	Plastics Materials and Resins	16,413,600,000	4,386,000,000	26.72%
325212	Synthetic Rubber	1,758,600,000	404,000,000	22.97%
325222	Noncellulosic Organic Fibers	0	0	
325311	Nitrogenous Fertilizers	13,775,700,000	0	0.00%
325312	Phosphatic Fertilizers	0	0	
326	Plastics and Rubber Products	34,292,700,000	19,642,000,000	57.28%
326199	Other Plastics Products	13,775,700,000	9,434,000,000	68.48%
327	Nonmetallic Mineral Products	79,430,100,000	10,191,000,000	12.83%
3272	Glass and Glass Products	15,534,300,000	3,294,000,000	21.20%
327310	Cements	26,085,900,000	2,763,000,000	10.59%
331	Primary Metals	402,133,200,000	54,092,000,000	13.45%
331111	Iron and Steel Mills	318,013,500,000	19,270,000,000	6.06%
331112	Electrometallurgical Ferroalloy Products	6,448,200,000	0	0.00%
3312	Steel Products from Purchased Steel	7,620,600,000	0	0.00%
3313	Alumina and Aluminum	28,723,800,000	0	0.00%
331312	Primary Aluminum	0	0	
3315	Foundries	32,534,100,000	11,465,000,000	35.24%
331511	Iron Foundries	19,051,500,000	7,097,000,000	37.25%
331521	Aluminum Die-Casting Foundries	3,810,300,000	1,078,000,000	28.29%
331524	Aluminum Foundries, except Die-Casting	3,517,200,000	787,000,000	22.38%
332	Fabricated Metal Products	66,240,600,000	26,371,000,000	39.81%
333	Machinery	32,534,100,000	12,445,000,000	38.25%
334	Computer and Electronic Products	10,551,600,000	6,711,000,000	63.60%
334413	Semiconductors and Related Devices	879,300,000	659,000,000	74.95%
335	Electrical Equip., Appliances, and Components	12,017,100,000	4,884,000,000	40.64%
336	Transportation Equipment	85,878,300,000	32,745,000,000	38.13%
337	Furniture and Related Products	11,724,000,000	3,007,000,000	25.65%
339	Miscellaneous	5,862,000,000	0	0.00%
	Total	1,347,673,800,000	271,992,000,000	20.18%
		0	0	
	South Census Region	0	0	
		0	0	
	RSE Column Factors:	234,480,000	800,000	0.34%
		0	0	
311	Food	89,395,500,000	21,857,000,000	24.45%

311221	Wet Corn Milling	5,275,800,000	943,000,000	17.87%
312	Beverage and Tobacco Products	14,655,000,000	3,490,000,000	23.81%
313	Textile Mills	60,964,800,000	26,601,000,000	43.63%
313210	Broadwoven Fabric Mills	19,051,500,000	10,377,000,000	54.47%
314	Textile Product Mills	11,724,000,000	4,129,000,000	35.22%
315	Apparel	8,793,000,000	3,529,000,000	40.13%
316	Leather and Allied Products	293,100,000	0	0.00%
321	Wood Products	87,343,800,000	11,313,000,000	12.95%
321113	Sawmills	21,689,400,000	3,193,000,000	14.72%
321114	Wood Preservation	2,637,900,000	194,000,000	7.35%
3212	Veneer, Plywood, and Engineered Woods	41,620,200,000	6,351,000,000	15.26%
3219	Other Wood Products	0	1,576,000,000	
322	Paper	470,132,400,000	31,231,000,000	6.64%
322110	Pulp Mills	46,602,900,000	828,000,000	1.78%
322121	Paper Mills, except Newsprint	156,222,300,000	8,016,000,000	5.13%
322122	Newsprint Mills	29,603,100,000	8,157,000,000	27.55%
322130	Paperboard Mills	215,721,600,000	8,801,000,000	4.08%
323	Printing and Related Support	8,793,000,000	4,954,000,000	56.34%
323110	Commercial Lithographic Printing	4,982,700,000	2,559,000,000	51.36%
324	Petroleum and Coal Products	552,200,400,000	20,443,000,000	3.70%
324110	Petroleum Refineries	540,476,400,000	19,511,000,000	3.61%
325	Chemicals	839,731,500,000	104,181,000,000	12.41%
325110	Petrochemicals	160,032,600,000	0	0.00%
325120	Industrial Gases	35,172,000,000	0	0.00%
325181	Alkalies and Chlorine	35,758,200,000	10,007,000,000	27.99%
325188	Other Basic Inorganic Chemicals	31,947,900,000	16,531,000,000	51.74%
325192	Cyclic Crudes and Intermediates	15,534,300,000	0	0.00%
325199	Other Basic Organic Chemicals	271,996,800,000	0	0.00%
325211	Plastics Materials and Resins	94,085,100,000	13,340,000,000	14.18%
325212	Synthetic Rubber	28,137,600,000	1,803,000,000	6.41%
325222	Noncellulosic Organic Fibers	30,189,300,000	0	0.00%
325311	Nitrogenous Fertilizers	48,654,600,000	2,259,000,000	4.64%
325312	Phosphatic Fertilizers	15,534,300,000	0	0.00%
326	Plastics and Rubber Products	36,637,500,000	20,080,000,000	54.81%
326199	Other Plastics Products	7,620,600,000	5,695,000,000	74.73%
327	Nonmetallic Mineral Products	112,843,500,000	16,615,000,000	14.72%
3272	Glass and Glass Products	22,568,700,000	4,989,000,000	22.11%
327310	Cements	39,275,400,000	4,464,000,000	11.37%
331	Primary Metals	195,204,600,000	53,915,000,000	27.62%
331111	Iron and Steel Mills	103,464,300,000	15,778,000,000	15.25%
331112	Electrometallurgical Ferroalloy Products	0	0	

3312	Steel Products from Purchased Steel	3,810,300,000	0	0.00%
3313	Alumina and Aluminum	60,671,700,000	25,629,000,000	42.24%
331312	Primary Aluminum	19,930,800,000	18,570,000,000	93.17%
3315	Foundries	12,310,200,000	0	0.00%
331511	Iron Foundries	6,741,300,000	0	0.00%
331521	Aluminum Die-Casting Foundries	1,172,400,000	257,000,000	21.92%
331524	Aluminum Foundries, except Die-Casting	1,465,500,000	269,000,000	18.36%
332	Fabricated Metal Products	31,068,600,000	12,697,000,000	40.87%
333	Machinery	15,827,400,000	8,587,000,000	54.25%
334	Computer and Electronic Products	15,241,200,000	10,806,000,000	70.90%
334413	Semiconductors and Related Devices	4,396,500,000	3,296,000,000	74.97%
335	Electrical Equip., Appliances, and Components	14,655,000,000	7,419,000,000	50.62%
336	Transportation Equipment	31,654,800,000	14,410,000,000	45.52%
337	Furniture and Related Products	9,379,200,000	0	0.00%
339	Miscellaneous	10,551,600,000	4,695,000,000	44.50%
	Total	2,617,383,000,000	385,142,000,000	14.71%
		0	0	
	West Census Region	0	0	
		0	0	
	RSE Column Factors:	293,100,000	900,000	0.31%
		0	0	
311	Food	56,861,400,000	9,763,000,000	17.17%
311221	Wet Corn Milling	879,300,000	89,000,000	10.12%
312	Beverage and Tobacco Products	7,620,600,000	1,556,000,000	20.42%
313	Textile Mills	2,637,900,000	754,000,000	28.58%
313210	Broadwoven Fabric Mills	293,100,000	77,000,000	26.27%
314	Textile Product Mills	0	0	
315	Apparel	1,172,400,000	521,000,000	44.44%
316	Leather and Allied Products	0	0	
321	Wood Products	39,568,500,000	5,606,000,000	14.17%
321113	Sawmills	23,448,000,000	2,012,000,000	8.58%
321114	Wood Preservation	293,100,000	34,000,000	11.60%
3212	Veneer, Plywood, and Engineered Woods	12,603,300,000	2,527,000,000	20.05%
3219	Other Wood Products	3,517,200,000	1,032,000,000	29.34%
322	Paper	98,188,500,000	12,601,000,000	12.83%
322110	Pulp Mills	0	0	
322121	Paper Mills, except Newsprint	29,603,100,000	4,305,000,000	14.54%
322122	Newsprint Mills	12,896,400,000	4,580,000,000	35.51%
322130	Paperboard Mills	44,844,300,000	2,145,000,000	4.78%
323	Printing and Related Support	3,810,300,000	1,975,000,000	51.83%
323110	Commercial Lithographic Printing	2,051,700,000	1,093,000,000	53.27%
324	Petroleum and Coal	248,548,800,000	6,080,000,000	2.45%

	Products			
324110	Petroleum Refineries	233,307,600,000	5,973,000,000	2.56%
325	Chemicals	61,257,900,000	12,614,000,000	20.59%
325110	Petrochemicals	0	0	
325120	Industrial Gases	0	0	
325181	Alkalies and Chlorine	3,517,200,000	0	0.00%
325188	Other Basic Inorganic Chemicals	28,430,700,000	2,688,000,000	9.45%
325192	Cyclic Crudes and Intermediates	0	0	
325199	Other Basic Organic Chemicals	1,758,600,000	0	0.00%
325211	Plastics Materials and Resins	586,200,000	242,000,000	41.28%
325212	Synthetic Rubber	0	0	
325222	Noncellulosic Organic Fibers	0	5,000,000	
325311	Nitrogenous Fertilizers	10,551,600,000	0	0.00%
325312	Phosphatic Fertilizers	0	0	
326	Plastics and Rubber Products	8,499,900,000	4,884,000,000	57.46%
326199	Other Plastics Products	3,224,100,000	2,213,000,000	68.64%
327	Nonmetallic Mineral Products	50,120,100,000	6,318,000,000	12.61%
3272	Glass and Glass Products	6,741,300,000	1,566,000,000	23.23%
327310	Cements	24,620,400,000	2,569,000,000	10.43%
331	Primary Metals	83,533,500,000	34,872,000,000	41.75%
331111	Iron and Steel Mills	20,810,100,000	0	0.00%
331112	Electrometallurgical Ferroalloy Products	0	0	
3312	Steel Products from Purchased Steel	1,465,500,000	0	0.00%
3313	Alumina and Aluminum	29,603,100,000	25,280,000,000	85.40%
331312	Primary Aluminum	24,620,400,000	23,037,000,000	93.57%
3315	Foundries	0	0	
331511	Iron Foundries	0	175,000,000	
331521	Aluminum Die-Casting Foundries	293,100,000	0	0.00%
331524	Aluminum Foundries, except Die-Casting	0	78,000,000	
332	Fabricated Metal Products	12,017,100,000	5,492,000,000	45.70%
333	Machinery	4,982,700,000	2,698,000,000	54.15%
334	Computer and Electronic Products	20,517,000,000	14,486,000,000	70.60%
334413	Semiconductors and Related Devices	9,379,200,000	6,610,000,000	70.48%
335	Electrical Equip., Appliances, and Components	1,758,600,000	1,014,000,000	57.66%
336	Transportation Equipment	12,603,300,000	6,185,000,000	49.07%
337	Furniture and Related Products	2,051,700,000	0	0.00%
339	Miscellaneous	3,224,100,000	0	0.00%
	Total	719,267,400,000	130,130,000,000	18.09%

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trillion =	1,000,000,000,000
million =	1,000,000

International - Coalmine Methane

Country	Year	Degas w/ Bleed and 20% closed areas - MTCO2E	kWh	\$ @ 0.02 profit/kWh	\$ Value Carbon Credits	kWh\$ w/ MTCO2E\$
World Total	2000	450,485,583	117,230,000,000	\$2,344,600,000	\$2,486,680,418	\$4,831,280,418
	2005	499,480,645	129,980,000,000	\$2,599,600,000	\$2,757,133,160	\$5,356,733,160
	2010	522,191,298	135,890,000,000	\$2,717,800,000	\$2,882,495,965	\$5,600,295,965
	2015	546,669,615	142,260,000,000	\$2,845,200,000	\$3,017,616,275	\$5,862,816,275
	2020	559,273,835	145,540,000,000	\$2,910,800,000	\$3,087,191,569	\$5,997,991,569
China	2000	162,855,745	42,380,000,000	\$847,600,000	\$898,963,712	\$1,746,563,712
	2005	179,802,273	46,790,000,000	\$935,800,000	\$992,508,547	\$1,928,308,547
	2010	196,249,243	51,070,000,000	\$1,021,400,000	\$1,083,295,821	\$2,104,695,821
	2015	212,542,503	55,310,000,000	\$1,106,200,000	\$1,173,234,617	\$2,279,434,617
	2020	228,835,763	59,550,000,000	\$1,191,000,000	\$1,263,173,412	\$2,454,173,412
United States	2000	74,626,205	19,420,000,000	\$388,400,000	\$411,936,652	\$800,336,652
	2005	83,695,095	21,780,000,000	\$435,600,000	\$461,996,924	\$897,596,924
	2010	85,347,478	22,210,000,000	\$444,200,000	\$471,118,079	\$915,318,079
	2015	86,423,448	22,490,000,000	\$449,800,000	\$477,057,433	\$926,857,433
	2020	83,887,233	21,830,000,000	\$436,600,000	\$463,057,526	\$899,657,526
Russia	2000	36,621,408	9,530,000,000	\$190,600,000	\$202,150,172	\$392,750,172
	2005	46,458,848	12,090,000,000	\$241,800,000	\$256,452,841	\$498,252,841
	2010	48,149,658	12,530,000,000	\$250,600,000	\$265,786,112	\$516,386,112
	2015	49,878,895	12,980,000,000	\$259,600,000	\$275,331,500	\$534,931,500
	2020	43,000,373	11,190,000,000	\$223,800,000	\$237,362,059	\$461,162,059
Ukraine	2000	27,129,815	7,060,000,000	\$141,200,000	\$149,756,579	\$290,956,579
	2005	34,123,620	8,880,000,000	\$177,600,000	\$188,362,382	\$365,962,382
	2010	37,582,095	9,780,000,000	\$195,600,000	\$207,453,164	\$403,053,164
	2015	38,504,355	10,020,000,000	\$200,400,000	\$212,544,040	\$412,944,040
	2020	39,311,333	10,230,000,000	\$204,600,000	\$216,998,558	\$421,598,558
Australia	2000	18,983,185	4,940,000,000	\$98,800,000	\$104,787,181	\$203,587,181
	2005	20,059,155	5,220,000,000	\$104,400,000	\$110,726,536	\$215,126,536
	2010	22,172,668	5,770,000,000	\$115,400,000	\$122,393,127	\$237,793,127
	2015	23,517,630	6,120,000,000	\$122,400,000	\$129,817,318	\$252,217,318
	2020	25,976,990	6,760,000,000	\$135,200,000	\$143,392,985	\$278,592,985
Germany	2000	16,293,260	4,240,000,000	\$84,800,000	\$89,938,795	\$174,738,795
	2005	12,757,930	3,320,000,000	\$66,400,000	\$70,423,774	\$136,823,774
	2010	7,647,073	1,990,000,000	\$39,800,000	\$42,211,843	\$82,011,843
	2015	7,647,073	1,990,000,000	\$39,800,000	\$42,211,843	\$82,011,843
	2020	7,647,073	1,990,000,000	\$39,800,000	\$42,211,843	\$82,011,843
Poland	2000	14,909,870	3,880,000,000	\$77,600,000	\$82,302,482	\$159,902,482
	2005	14,717,733	3,830,000,000	\$76,600,000	\$81,241,886	\$157,841,886
	2010	13,142,205	3,420,000,000	\$68,400,000	\$72,544,972	\$140,944,972
	2015	12,604,220	3,280,000,000	\$65,600,000	\$69,575,294	\$135,175,294
	2020	11,835,670	3,080,000,000	\$61,600,000	\$65,332,898	\$126,932,898
India	2000	9,491,593	2,470,000,000	\$49,400,000	\$52,393,593	\$101,793,593
	2005	10,029,578	2,610,000,000	\$52,200,000	\$55,363,271	\$107,563,271
	2010	10,721,273	2,790,000,000	\$55,800,000	\$59,181,427	\$114,981,427

	2015	11,374,540	2,960,000,000	\$59,200,000	\$62,787,461	\$121,987,461
	2020	12,027,808	3,130,000,000	\$62,600,000	\$66,393,500	\$128,993,500
Kazakhstan	2000	6,801,668	1,770,000,000	\$35,400,000	\$37,545,207	\$72,945,207
	2005	7,493,363	1,950,000,000	\$39,000,000	\$41,363,364	\$80,363,364
	2010	7,493,363	1,950,000,000	\$39,000,000	\$41,363,364	\$80,363,364
	2015	7,493,363	1,950,000,000	\$39,000,000	\$41,363,364	\$80,363,364
	2020	7,493,363	1,950,000,000	\$39,000,000	\$41,363,364	\$80,363,364
South Africa	2000	6,801,668	1,770,000,000	\$35,400,000	\$37,545,207	\$72,945,207
	2005	8,377,195	2,180,000,000	\$43,600,000	\$46,242,116	\$89,842,116
	2010	8,377,195	2,180,000,000	\$43,600,000	\$46,242,116	\$89,842,116
	2015	8,377,195	2,180,000,000	\$43,600,000	\$46,242,116	\$89,842,116
	2020	8,377,195	2,180,000,000	\$43,600,000	\$46,242,116	\$89,842,116
United Kingdom	2000	5,418,278	1,410,000,000	\$28,200,000	\$29,908,895	\$58,108,895
	2005	3,996,460	1,040,000,000	\$20,800,000	\$22,060,459	\$42,860,459
	2010	3,996,460	1,040,000,000	\$20,800,000	\$22,060,459	\$42,860,459
	2015	3,804,323	990,000,000	\$19,800,000	\$20,999,863	\$40,799,863
	2020	3,804,323	990,000,000	\$19,800,000	\$20,999,863	\$40,799,863
Czech Republic	2000	5,418,278	1,410,000,000	\$28,200,000	\$29,908,895	\$58,108,895
	2005	5,110,858	1,330,000,000	\$26,600,000	\$28,211,936	\$54,811,936
	2010	4,457,590	1,160,000,000	\$23,200,000	\$24,605,897	\$47,805,897
	2015	3,804,323	990,000,000	\$19,800,000	\$20,999,863	\$40,799,863
	2020	3,189,483	830,000,000	\$16,600,000	\$17,605,946	\$34,205,946
Mexico	2000	1,344,963	350,000,000	\$7,000,000	\$7,424,196	\$14,424,196
	2005	2,113,513	550,000,000	\$11,000,000	\$11,666,592	\$22,666,592
	2010	1,806,093	470,000,000	\$9,400,000	\$9,969,633	\$19,369,633
	2015	1,921,375	500,000,000	\$10,000,000	\$10,605,990	\$20,605,990
	2020	1,921,375	500,000,000	\$10,000,000	\$10,605,990	\$20,605,990
Additional Countries	2000	63,789,650	16,600,000,000	\$332,000,000	\$352,118,868	\$684,118,868
	2005	70,745,028	18,410,000,000	\$368,200,000	\$390,512,555	\$758,712,555
	2010	75,048,908	19,530,000,000	\$390,600,000	\$414,269,972	\$804,869,972
	2015	78,776,375	20,500,000,000	\$410,000,000	\$434,845,590	\$844,845,590
	2020	81,965,858	21,330,000,000	\$426,600,000	\$452,451,536	\$879,051,536
MTCO2e	\$5.52					
\$ kWh	\$0.02					

US Landfills

Landfill City	State	Waste In Place (tons)	Landfill Owner Organization	Project Status	Project Developer Organization	MW Capacity	LFG Flow to Project (mmscfd)
Birmingham	AL	6,394,527	City of Birmingham	Operational	DTE Biomass		0.970
Montgomery	AL	7,259,313	City of Montgomery	Operational	DTE Biomass		1.280
Mt. Vernon	AL	3,184,981	City of Mobile	Construction	Consortium Service Management/RTC		
Huntsville	AL	4,842,834	City of Huntsville S.W. Disposal Authority	Shutdown	Maguire Group		
Mobile	AL	967,998	Mobile County Commission	Shutdown			
Athens	AL	1,195,575	City of Athens	Candidate			
Albertville	AL	1,078,906	Bishop Landfill Company	Candidate			
Coker	AL	1,800,000	Black Warrior Solid Waste Disposal Authority	Candidate			
Anniston	AL	4,350,000	Calhoun County Commission	Candidate			
Dothan	AL	4,299,512	City of Dothan	Candidate			
Elba	AL	1,800,000	Coffee County Commission	Candidate			
	AL	1,000,000	Colbert County Commission	Candidate			
Birmingham	AL	2,750,000	City of Birmingham	Candidate			
Florence	AL	275,643	City of Florence Public Works	Candidate			
Florence	AL	1,461,912	City of Florence	Candidate			
Columbiana	AL	2,460,000	Shelby County	Candidate			
Magnolia Springs	AL	1,800,000	Baldwin County Commission	Candidate			
Monroeville	AL	1,205,385	Monroe County Commission	Candidate			
Trinity	AL	2,700,000	City of Decatur/Morgan County	Candidate			
Mt. Olive	AL	2,071,034	Jefferson County Commission	Candidate			
Ragland	AL	1,000,000	Onyx Waste Services, Inc.	Candidate			
Moody	AL	1,660,000	Onyx Waste Services, Inc.	Candidate			
Dora	AL	681,184	Allied Waste Industries	Candidate			
Opelika	AL	6,000,000	Waste Management, Inc.	Candidate			
			Jefferson County Solid Waste Admin.	Candidate			
Homewood	AL	9,324,502		Candidate			
Chickasaw	AL	4,550,396	Southside Storage, Inc.	Candidate			
	AL	1,200,000	Waste Management, Inc.	Candidate			
	AL	4,657,815	Jefferson County	Candidate			
Andalusia	AL		City of Andalusia	Potential			
Atmore	AL		City of Atmore	Potential			
Hayden	AL		Blount County	Potential			
Hartselle	AL		City of Hartselle	Potential			
	AL	912,793	Cullman County Commission	Potential			
Cullman	AL	248,686	Cullman Environmental, Inc.	Potential			
Fort Payne	AL	1,456,490	City of Fort Payne	Potential			
Geneva	AL		Southeast AL Solid Waste Disposal Auth.	Potential			
Haleyville	AL		TransAmerican Waste Industries Inc	Potential			
Addison	AL		Hyche Landfill (Mobile Home)	Potential			
Hillsboro	AL		Browning-Ferris Industries Inc	Potential			
Tuscaloosa	AL		Onyx Eagle Bluff Landfill	Potential			
Carrollton	AL		Pickens County	Potential			
Red Bay	AL		City of Red Bay	Potential			
	AL	850,705	City of Saraland	Potential			
Brewton	AL	716,007	Allied Waste Industries, Inc.	Potential			
	AL			Potential			
Tuskegee	AL	1,034,024	City of Tuskegee	Potential			
	AL	752,393	WMI	Potential			
Jasper	AL		Walker County	Potential			
Anchorage	AK	4,350,000	Municipality of Anchorage	Candidate			
Mobile	AZ	9,300,000	Waste Management Inc	Operational		0.1	
Chandler	AZ	2,100,000	City of Chandler	Operational	City of Chandler	0.3	
Tucson	AZ	10,000,000	City of Tucson	Operational	U.S. Energy Biogas Corp.	6.0	
Phoenix	AZ	12,000,000	Pima Maricopa Indian Community	Operational	DTE Biomass/LES	4.0	
Phoenix	AZ	3,000,000	Pima Maricopa Indian Community	Shutdown	DTE Biomass	0.0	
Phoenix	AZ	10,900,000	City of Phoenix	Candidate			
Apache Junction	AZ	1,000,000	Allied Waste Industries Inc.	Candidate			
Flagstaff	AZ	2,150,000	City of Flagstaff	Candidate			
Glendale	AZ	5,000,000	City of Glendale	Candidate			
			Waste Management of Arizona, Inc. (WMA)	Candidate			
Dewey	AZ	3,000,000		Candidate			
Huachuca	AZ	1,200,000	City of Huachuca	Candidate			
Phoenix	AZ	2,000,000	Pima Maricopa Indian Community	Candidate			
Phoenix	AZ	1,000,000	WMI	Candidate			
Joseph City	AZ	1,500,000	WMI	Candidate			
Queen Creek	AZ	1,500,000	Maricopa County	Candidate			
Rio Rico	AZ	1,000,000	Santa Cruz County	Candidate			
Phoenix	AZ	3,000,000	Pima Maricopa Indian Community	Candidate			
Phoenix	AZ	12,000,000	City of Phoenix	Candidate			
Tucson	AZ	2,100,000	Pima County	Candidate			
Somerton	AZ	2,200,000	WMI	Potential			
Tucson	AZ	2,250,000	City of Tucson	Potential			
Buckeye	AZ		Allied Waste Industries	Potential			
Tucson	AZ		City of Tucson	Potential			
Fort Smith	AR	2,453,458	City of Fort Smith	Operational	NEO		0.474
Little Rock	AR	799,244	City of Little Rock	Candidate			

Jonesboro	AR	1,600,000	Craighead County Solid Waste Disposal Authority	Candidate			
Springdale	AR	3,500,000	Waste Management of Arkansas, Inc.	Candidate			
North Little Rock	AR	3,825,000	Waste Management of Arkansas, Inc.	Candidate			
White Hall	AR	780,000	Waste Management of Arkansas, Inc.	Potential			
Little Rock	AR		Allied Waste	Potential			
Centerville	AR	780,000	Waste Management of Arkansas, Inc.	Potential			
Martinez	CA	10,800,000	Acme Fill Corporation	Operational	SCS Engineers	0.3	0.216
Martinez	CA	10,800,000	Acme Fill Corporation	Operational	Acme Fill	1.9	
Martinez	CA	10,800,000	Acme Fill Corporation	Operational	Acme Fill		1.500
Livermore	CA	26,450,512	WMI	Operational		2.5	1.440
Livermore	CA	26,450,512	WMI	Operational	Bio Energy Partners	6.6	
Napa	CA	4,230,000	South Napa SWA	Operational	GRS	1.5	1.000
Azusa	CA	7,900,000	Azusa Land Reclamation Company (Waste Management, Inc.)	Operational	Energy Developments, Inc.	3.2	
Moreno Valley	CA	4,081,500	Riverside County	Operational	Riverside County	1.1	0.576
Oxnard	CA	4,336,609	Ventura Regional Sanitation District	Operational	Covanta Energy	1.7	
West Covina	CA	46,000,000	BKK Corporation	Operational	NEO	4.3	6.700
West Covina	CA	46,000,000	BKK Corporation	Operational	NEO	6.0	7.780
Sun Valley	CA	31,000,000	WMI	Operational	WMI/Stewart and Stevenson	5.2	3.300
Sun Valley	CA	31,000,000	WMI	Operational	WMI	5.3	3.300
Burbank	CA	2,000,000	City of Burbank	Operational		0.3	
Agoura Hills	CA	21,000,000	County of Los Angeles	Operational	SCS Energy	0.3	0.216
Agoura Hills	CA	21,000,000	County of Los Angeles	Operational	LACSD	10.0	5.550
Redlands	CA	960,000	City of Redlands	Operational	City of Redlands	1.0	0.432
Antioch	CA		Contra Costa Solid Waste Inc. & GBF Com.	Operational		3.3	
Petaluma	CA	15,000,000	Sonoma County	Operational	LES/DTE Biomass	1.6	3.460
Petaluma	CA	15,000,000	Sonoma County	Operational	LES/DTE Biomass	3.2	
Petaluma	CA	15,000,000	Sonoma County	Operational	LES/DTE Biomass	3.2	
Corona	CA	4,000,000	City of Corona	Operational	City of Corona	0.6	0.432
Santa Clara	CA	3,500,000	City of Santa Clara	Operational	Covanta Energy	1.4	
Oxnard	CA	3,300,000	Ventura County	Operational	Covanta Energy	2.6	
Colton	CA	4,427,243	San Bernardino County	Operational	Montauk Energy Capital/NEO	1.2	0.700
Newport Beach	CA	25,000,000	Irvine Company	Operational	GRS	20.0	7.000
Salinas	CA	1,642,549	Salinas Valley Solid Waste Authority	Operational	Covanta Energy	1.4	
San Leandro	CA	9,700,000	East Bay Regional Parks District	Operational			
Manteca	CA	2,030,690	Forward, Inc. (Allied Waste Industries)	Operational	Covanta Energy	0.8	
San Jose	CA	5,500,000	Guadalupe Rubbish Disposal Company Co., Inc.	Operational	GRS		
Industry Hills	CA	1,500,000	City of Industry	Operational		2.5	1.500
Jamacha	CA		County of San Diego	Operational		0.5	0.190
Pittsburg	CA	3,500,000	Allied Waste Industries	Operational	County of San Diego	0.3	
Sacramento	CA	18,000,000	Allied Waste Industries	Operational	Energy Developments, Inc.	3.2	1.500
Sacramento	CA	18,000,000	Sacramento County	Operational	LES/DTE Biomass	9.1	
Lakeview Terrace	CA	16,600,000	Los Angeles Department of Public Works, Bureau of Sanitation	Operational	NEO	6.0	4.000
Lakeview Terrace	CA	16,600,000	Los Angeles Department of Public Works, Bureau of Sanitation	Operational	City of Los Angeles - Bureau of Sanitation	1.5	1.152
Menlo Park	CA	5,000,000	City of Menlo Park	Operational	GRS	2.0	1.500
Rialto	CA	2,934,655	San Bernardino County	Operational	Montauk Energy Capital/NEO	2.5	1.440
Ontario	CA	14,367,517	San Bernardino County	Operational	Montauk Energy Capital/NEO	2.5	2.880
San Diego	CA	34,000,000	City of San Diego	Operational	NEO	3.8	
San Diego	CA	34,000,000	City of San Diego	Operational	NEO	6.5	
Marina	CA	6,000,000	Monterey Regional Waste Management District	Operational	U.S. Energy Biogas Corp.	0.8	
Marina	CA	6,000,000	Monterey Regional Waste Management District	Operational	U.S. Energy Biogas Corp.	1.0	
Marina	CA	6,000,000	Monterey Regional Waste Management District	Operational	U.S. Energy Biogas Corp.	1.2	1.200
Marina	CA	6,000,000	Monterey Regional Waste Management District	Operational	U.S. Energy Biogas Corp.	0.8	
Los Angeles	CA	10,000,000	Barclay Hollander/American Golf Com.	Operational	GSF Energy	7.5	2.880
Fremont	CA	2,750,000	Allied Waste Industries	Operational	Gas Recovery Systems, Inc.	3.3	1.900
Fremont	CA	2,750,000	Allied Waste Industries	Operational	Gas Recovery Systems, Inc.		
Fremont	CA	2,750,000	Allied Waste Industries	Operational	Gas Recovery Systems, Inc.	2.0	1.100
Brea	CA	32,981,156	Orange County IWMD	Operational	GSF Energy	5.6	3.890
Monterey Park	CA		Operating Industries, Inc.	Operational	SCS Engineers/New Cure, Inc.	0.4	
Chula Vista	CA	8,099,138	Allied Waste Industries	Operational	Covanta Energy	3.7	
Palo Alto	CA	2,700,000	City of Palo Alto	Operational	EMCON/WPI	1.0	
Rolling Hills Estates	CA	23,000,000	LA County Sanitation District	Operational	LACSD	6.0	
Sun Valley	CA	9,000,000	Los Angeles By-Products, Inc.	Operational	Covanta Energy	4.5	
Lincoln	CA	2,000,000	County Of Placer Western Regional SLF	Operational		1.2	
San Juan Capistrano	CA	9,624,488	Orange County I.W.M.	Operational	NEO	6.0	2.700
Whittier	CA	62,500,000	LA County Sanitation District	Operational	LACSD		0.400
Whittier	CA	62,500,000	LA County Sanitation District	Operational	LACSD	3.9	5.300
Whittier	CA	62,500,000	LA County Sanitation District	Operational	LACSD	50.0	32.000
Novato	CA	1,000,000	Redwood Landfill, Inc. - Sanfill	Operational	RECO		
Sacramento	CA	3,900,000	City of Sacramento	Operational	GRS		1.600
San Marcos	CA	11,494,445	San Diego County	Operational	GRS	1.5	1.000
Oxnard	CA	2,700,000	City of Oxnard	Operational	Covanta Energy	0.9	
Santa Cruz	CA	2,040,427	City of Santa Cruz	Operational	GRS	0.7	0.500
Glendale	CA	23,340,000	City of Glendale	Operational	Palmer Capital Corporation	12.0	7.358
Sun Valley	CA	5,500,000	City of Los Angeles	Operational	Covanta Energy	2.3	0.432

Simi Valley	CA	12,085,000	WMI	Operational	WMI/Cryo Energy	2.7	1.296
Walnut	CA	12,500,000	CA Polytechnic of Pomona	Operational	LACSD	8.5	
Sunnyvale	CA	2,300,000	City Of Sunnyvale	Operational	City of Sunnyvale	1.6	0.656
Santee	CA	9,163,739	Allied Waste Industries	Operational	GRS	1.5	1.000
Goleta	CA	6,504,188	Santa Barbara County	Operational	NEO	3.0	1.500
Los Angeles	CA	16,000,000	City of Los Angeles	Operational	Covanta Energy	4.0	
El Dorado	CA	1,000,000	El Dorado County	Operational	El Dorado County	1.9	
Visalia	CA	3,000,000	Tulare County	Operational	NEO	1.9	1.000
Richmond	CA	10,105,056	Republic Services, Inc.	Operational	West County Energy Recovery Corp.	3.0	
Woodville	CA	2,500,000	Tulare County	Operational	NEO	0.6	0.400
Davis	CA	8,427,340	Yolo County	Operational	NEO	2.5	1.400
Whittier	CA	62,500,000	LA County Sanitation District	Construction	LACSD	3.0	
Wilmington	CA	4,000,000	Watson Energy Systems/Desser	Shutdown	Watson Biogas Systems		
Bakersfield	CA	13,000,000	City of Bakersfield	Shutdown	Cambrian Energy Development LLC		
	CA		County of San Diego	Shutdown	Cambrian Energy Development LLC		
Watsonville	CA	2,040,000	County of Santa Cruz	Shutdown	EMCON/Perennial Energy	0.3	
Burbank	CA	2,000,000	City of Burbank	Shutdown	Cambrian Energy Development LLC	0.8	
Lompoc	CA	1,056,000	City of Lompoc	Shutdown	Cambrian Development LLC		
Duarte	CA	1,600,000	City of Duarte	Shutdown	O'Brien Environmental Energy/NEO	1.6	
Industry Hills	CA	1,500,000	City of Industry	Shutdown	City of Industry		0.360
Brea	CA	32,981,156	Orange County IWMD	Shutdown	Ridgewood Power	2.5	1.440
Whittier	CA	62,500,000	LA County Sanitation District	Shutdown	LACSD		0.300
Sun Valley	CA	5,500,000	City of Los Angeles	Shutdown	Pacific Energy/Ogden	0.8	
Mountain View	CA	12,700,000	City of Mountain View	Shutdown	Laidlaw Gas Recovery Systems	3.5	
Sylmar	CA	27,000,000	Allied Waste Industries	Shutdown	GRS	6.0	
Upland	CA		City of Upland	Shutdown	Cambrian Energy Development LLC		
Jone	CA	1,013,553	Amador County	Candidate			
Kerman	CA		Fresno County	Candidate			
Palmdale	CA	2,678,000	Palmdale Disposal Company, Inc.	Candidate			
Arvin	CA	3,700,000	County Of Kern-Truxtun AV	Candidate			
	CA		County of San Diego	Candidate			
Modesto	CA	1,800,000	Rudi Bonzi Inc	Candidate			
Sun Valley	CA	31,000,000	WMI	Candidate			
Watsonville	CA	2,040,000	County of Santa Cruz	Candidate			
Castaic	CA	6,434,424	Republic Services, Inc.	Candidate			
Eureka	CA	1,260,000		Candidate			
Santa Maria	CA	1,809,995	City Of Santa Maria	Candidate			
Ukiah	CA	2,200,000	City Of Ukiah	Candidate			
Coachella	CA	3,238,000	Riverside County	Candidate			
San Luis Obispo	CA	2,254,575	Corral De Piedra Land Company	Candidate			
Cathedral City	CA	6,100,000	Riverside County	Candidate			
Corona	CA	12,500,000	WMI	Candidate			
Crows Landing	CA	1,142,398	Stanislaus County	Candidate			
Stockton	CA	2,100,000	San Joaquin County	Candidate			
Irvine	CA	18,000,000	Orange County	Candidate			
Hanford	CA	1,668,000	Kings County WM Authority	Candidate			
Beaumont	CA	3,000,000	Riverside County	Candidate			
Lodi	CA	8,000,000	San Joaquin County Public Works	Candidate			
Wheatland	CA	1,060,000	Yuba-Sutter Disposal Inc. (Norcal Waste Systems)	Candidate			
Half Moon Bay	CA	9,128,242	Allied Waste Industries	Candidate			
Suisun City	CA	2,538,000	Potrero Hills Landfill, Inc. (Republic Services, Inc.)	Candidate			
Ramona	CA	343,200	San Diego County	Candidate			
Ridgecrest	CA	1,300,000	Kern County	Candidate			
Redlands	CA	2,300,000	San Bernardino County	Candidate			
Santa Monica	CA			Candidate			
Irvine	CA	17,100,000	Orange County	Candidate			
Shafter	CA	1,400,000	Kern County	Candidate			
Mountain View	CA	12,700,000	City of Mountain View	Candidate			
Mountain View	CA	12,700,000	City of Mountain View	Candidate			
Mountain View	CA	12,700,000	City of Mountain View	Candidate			
Sylmar	CA	27,000,000	Allied Waste Industries	Candidate			
Porterville	CA	1,614,000	Tulare County, Solid Waste Div.	Candidate			
Ventura	CA	2,700,000	Ventura Regional Sanitation District	Candidate			
Fremont	CA	7,915,544	Waste Management Inc	Candidate			
Vandenberg Air Force	CA	1,500,000	30 SW/CEV	Candidate			
Livermore	CA	9,067,500	Republic Services, Inc.	Candidate			
Marysville	CA	2,200,000	Yuba-Sutter Disposal Inc.	Candidate			
Rancho Cordova	CA		Aerojet, Department 1520, Building 46010	Potential			
Rialto	CA	159,000	Agua Mansa Landfill, Inc.	Potential			
Alturas	CA	33,872	Modoc County Department of Public Works	Potential			
Napa	CA	482,541	South NAPA SWA	Potential			
Anderson	CA	1,554,000	Republic Industries, Inc	Potential			
Annapolis	CA	45,289	Sonoma County	Potential			
Anza	CA	381,600	Riverside County	Potential			
Apple Valley	CA	265,800	County of San Bernardino	Potential			
San Diego	CA			Potential			
Avenal	CA	300,000	City of Avenal	Potential			
Vacaville	CA	780,000	B&J Drop Box SLF Inc.	Potential			
Moreno Valley	CA	4,081,500	Riverside County	Potential			
Baker	CA	30,000	San Bernardino County	Potential			

Balance Rock	CA	9,300	Tulare County	Potential			
Barstow	CA	300,000	San Bernardino County Bureau of Land Management	Potential			
Johnstonville	CA		U.S. Bureau of Land Management	Potential			
Beale Air Force Base	CA	389,440	U.S. Air Force	Potential			
Ben Lomond	CA	1,200,000	Santa Cruz County	Potential			
Benton	CA	246,000	Real Estate Sec./LA City Department W&P	Potential			
Benton	CA	9,030	Bureau of Land Management	Potential			
Napa	CA	43,200	Gomez Boys	Potential			
Bieber	CA	25,410	Lassen County Department of Public Works	Potential			
Big Bear City	CA		San Bernadino County	Potential			
Groveland	CA	241,200	U.S. Bureau of Land Management	Potential			
Los Banos	CA			Potential			
Bishop	CA		City of Los Angeles Water Department	Potential			
Mount Shasta	CA		U.S. Department of Agriculture, USFS	Potential			
Blythe	CA	455,000	Riverside County	Potential			
Boron	CA		Kern County Waste Management Department	Potential			
Borrego Springs	CA	107,389	City of San Diego, D.P.W.	Potential			
Glendale	CA	408,000	City of Glendale	Potential			
Brawley	CA		Imperial County	Potential			
Bridgport	CA	54,000	Bureau of Land Management	Potential			
Buttontwillow	CA		County of Kern	Potential			
Crockett	CA	75,000	C & H Sugar	Potential			
San Andreas	CA		Calaveras Cement - Flintkote	Potential			
Calexico	CA		Imperial Irrigation District	Potential			
San Luis Obispo	CA		CA Valley Community Services District	Potential			
Camp Roberts (Mil Re	CA	865,443	CA National Guard/U.S. Army Corp Eng.	Potential			
Los Osos	CA		State Military Department	Potential			
Petaluma	CA		City of Petaluma Street Department	Potential			
Caspar	CA	365,215	Mendocino County Division of Solid Waste	Potential			
Cecilville	CA		U.S. Department of Agriculture, USFS	Potential			
Cedarville	CA		Bureau of Land Management	Potential			
Chalfant	CA	7,644	Bureau of Land Management	Potential			
Fresno	CA	12,451,139	Allied Waste Industries	Potential			
Chester	CA		Roseburg Resources	Potential			
Fresno	CA	459,600	Shubin Farms/Trifty Best Pumping Service	Potential			
Atascadero	CA	444,000	Johnson W.	Potential			
Bakersfield	CA	1,639,000	County Of Kern-Truxtun AV	Potential			
Fresno	CA		City of Clovis	Potential			
Palo Alto	CA		City of Palo Alto	Potential			
Paso Robles	CA	870,000	City of Paso Robles	Potential			
Redding	CA		City Public Works Department	Potential			
Rialto	CA			Potential			
Whittier	CA		City of Whittier	Potential			
Willits	CA		City of Willits	Potential			
Applegate	CA		Sutherland Construction, Inc.	Potential			
Cloverdale	CA	157,380	Louisiana Pacific Corporation	Potential			
Coalinga	CA	580,350	Chevron USA	Potential			
Chester	CA		Collins Pine Company Landfill	Potential			
Tracy	CA	885,000	San Joaquin County Public Works	Potential			
Covelo	CA	396,438	Louisiana Pacific Corporation	Potential			
Crescent City	CA	806,400	Del Norte County	Potential			
Big Pine	CA		Deep Springs College	Potential			
Brawley	CA	258,633	Desert Valley Company	Potential			
Red Bluff	CA		Larzibal J.	Potential			
Elk Grove	CA	214,800	West Coast Building - Wrecking, Inc.	Potential			
Winchester	CA	2,100,000	Riverside County	Potential			
China Lake NWC (Mil	CA		Cook	Potential			
Desert Center	CA	23,000	Bureau of Land Management	Potential			
Eagleville	CA		Bureau of Land Management	Potential			
Earlimart	CA	782,400		Potential			
Truckee	CA		U.S. Forest Service	Potential			
Clearlake	CA		Lake County	Potential			
Edwards Air Force Ba	CA		Department of the Air Force	Potential			
Edwards Air Force Ba	CA		U.S. Air Force	Potential			
Arbuckle	CA		Colusa County	Potential			
Exeter	CA		Tulare County	Potential			
Chowchilla	CA		Madera County Engineering Department	Potential			
Fort Bidwell	CA		Modoc County Department of Public Works	Potential			
Los Olivos	CA	120,000	Chamberlin Estate	Potential			
Stockton	CA		City of Stockton Public Works	Potential			
Furnace Creek Ranch	CA		Death Valley National Monument	Potential			
Modesto	CA			Potential			
Artois	CA	540,000	Glenn County	Potential			
Glennville	CA		U.S. Forest Service	Potential			
Fort Irwin (Mil Res)	CA			Potential			

Spanish Ranch	CA	283,500	Plumas National Forest Quincy Road	Potential			
Hanford	CA		Weaver's Tree Service	Potential			
Happy Camp	CA		Chambers G	Potential			
Lodi	CA	4,320,000	San Joaquin County Public Works	Potential			
Hanford	CA		Harold L. James, Inc.	Potential			
Herlong	CA	7,592	Bureau of Land Management	Potential			
Hesperia	CA	180,000	County of San Bernardino	Potential			
Highgrove	CA	3,493,000	Riverside County	Potential			
Merced	CA		Merced County	Potential			
Colma	CA		Cypress-Amlac Land Company, Inc.	Potential			
Rialto	CA		Holliday Trucking	Potential			
Holtville	CA		Bureau of Land Management	Potential			
Trona	CA		U.S. Bureau of Land Management	Potential			
Niland	CA		Imperial County	Potential			
Forks of Salmon	CA		U.S. Department of Agriculture, USFS	Potential			
Imperial	CA	802,060		Potential			
Imperial	CA	171,000	Imperial Irrigation District	Potential			
Independence	CA		City of Los Angeles Water Department	Potential			
Hat Creek	CA	40,500	Intermountain Landfill, Inc.	Potential			
Hollister	CA		San Benito County Public Works	Potential			
Gonzales	CA	430,000	Salinas Valley Solid Waste Authority	Potential			
King City	CA	169,200	Carmel Marina Corp.	Potential			
Sawyers Bar	CA		U.S. Department of Agriculture, USFS	Potential			
Johnsontdale	CA	6,600	U.S. Bureau of Land Management	Potential			
Kernville	CA	276,507	Bureau of Land Management	Potential			
San Jose	CA		Waste Management of California	Potential			
Sacramento	CA		Teichert Land Company	Potential			
Lake City	CA		Bureau of Land Management	Potential			
Landers	CA	279,000	Bureau of Land Management	Potential			
Camp Pendleton (Mil	CA		Johnson R	Potential			
Tulelake	CA		Dept. of Interior National Park Service	Potential			
Laytonville	CA	48,000	Mendocino County Division of Solid Waste	Potential			
Lebec	CA	584,435	County of Kern	Potential			
Lenwood	CA	120,000	County of San Bernardino	Potential			
Watsonville	CA	443,400	Monterey County Public Works	Potential			
Lone Pine	CA		City of Los Angeles Water Department	Potential			
Lost Hills	CA		County of Kern	Potential			
Red Bluff	CA		Louisiana-Pacific Corporation	Potential			
Loyalton	CA	56,400	Sierra County	Potential			
Lucerne Valley	CA	57,000		Potential			
Madeline	CA		Bureau of Land Management	Potential			
Mariposa	CA	720,000	Mariposa County	Potential			
McCloud	CA		McCloud Community Services District	Potential			
Grass Valley	CA	2,182,268	Co.Of Nevada, Dept. Of San. & Trans.	Potential			
Delano	CA	1,071,000	County Of Kern-Truxtun Av	Potential			
Perris	CA	2,312,837	Riverside County	Potential			
Mecca	CA	72,000	Riverside County	Potential			
Vidal	CA	2,400	Metro Water District of Southern CA	Potential			
Lucerne Valley	CA	312,240	Mitsubishi Cement Corporation	Potential			
Mojave	CA	523,170	County of Kern	Potential			
Colton	CA		Triscott W.	Potential			
Morongo Valley	CA	120,000	County of San Bernardino	Potential			
Lemoore	CA	591,600	Assistant Public Works Officer	Potential			
Chico	CA		Butte County	Potential			
Needles	CA	123,600	Bureau of Land Management	Potential			
New Cuyama	CA	23,400	Santa Barbara County	Potential			
Newberry Springs	CA	51,000	Bureau of Land Management	Potential			
Niland	CA		Bureau of Land Management	Potential			
McKittrick	CA	39,761	Shell Western E & P, Inc.	Potential			
Lodi	CA		San Joaquin County	Potential			
Arcadia	CA			Potential			
Oasis	CA			Potential			
Ocotillo	CA		Bureau of Land Management	Potential			
Olancho	CA		U.S. Bureau of Land Management	Potential			
Fresno	CA		Caglia F.	Potential			
Oro Grande	CA	3,000,000	Riverside Cement Co.	Potential			
	CA	4,590	Louisiana-Pacific Corporation	Potential			
San Jose	CA		Owens-Corning Fiberglas Corporation	Potential			
Gilroy	CA		South Valley Refuse	Potential			
Point Mugu (Mil Res)	CA		U.S. Naval Air Station	Potential			
Palo Verde	CA		Bureau of Land Management	Potential			
Phelan	CA		County of San Bernardino	Potential			
Winterhaven	CA	76,800	Quechan Indian Tribe Council	Potential			
Saugus (Santa Clarit	CA		Los Angeles County Sheriff's Department	Potential			
Brownsville	CA	4,128,000	Bureau of Land Management	Potential			

Portola	CA	135,000	City of Portola	Potential			
Lee Vining	CA	55,800		Potential			
Ravendale	CA	11,616	Lassen County Department of Public Works	Potential			
Vallecito	CA		Calaveras County	Potential			
Fort Irwin (Mil Res)	CA	7,577,621	US Dept Of Army-Fort Irwin	Potential			
Rio Vista	CA		State Reclamation Board	Potential			
Milton	CA		Calaveras County	Potential			
Somes Bar	CA		U.S. Department of Agriculture, USFS	Potential			
Salton City	CA		Bureau of Land Management	Potential			
Bradley	CA	18,000	Monterey County Parks Department	Potential			
Camp Pendleton (Mil)	CA	238,200	U.S.M.C.	Potential			
Fellows	CA		Santa Fe Energy Company	Potential			
Sonoma	CA		Bureau of Land Management	Potential			
Avalon	CA	46,710	Republic Services, Inc.	Potential			
Shoshone	CA	15,708	U.S. Bureau of Land Management	Potential			
Herlong	CA	236,055	Department of the Army	Potential			
Anderson	CA	187		Potential			
Samoa	CA	54,723	Louisiana Pacific Corporation	Potential			
San Diego	CA	4,750,000	City of San Diego	Potential			
Gualala	CA	54,600	Mendocino County D.P.W.	Potential			
Selma	CA			Potential			
Marvsville	CA		W. Jaxon, Baker Enterprises	Potential			
Stonyford	CA		Colusa County	Potential			
Taft	CA		County of Kern	Potential			
Tecopa	CA	48,707		Potential			
Tehachapi	CA	450,890	County of Kern	Potential			
Red Bluff	CA		Tehama County/Red Bluff Landfill Mgmt Agency	Potential			
Tennant	CA		U.S. Department of Agriculture, USFS	Potential			
Ventura San Buenave	CA	7,200		Potential			
Trona	CA	102,000		Potential			
Sonoma	CA		Vallejo Sanitation & Flood Control Dist.	Potential			
Tulelake	CA		City of Tulelake	Potential			
Jamestown	CA	986,400	Tuolumne County	Potential			
Twentynine Palms	CA	198,000	Bureau of Land Management	Potential			
Anderson	CA	85,200		Potential			
Avalon	CA		Santa Catalina Island Conservancy	Potential			
Carson	CA	44,231	Code 18 N.A.S. North Island	Potential			
Twentynine Palms	CA	612,000		Potential			
Davis	CA	534,000	Regents of University of California	Potential			
Napa	CA	570,000	Tamangei (Estate) W	Potential			
Bakersfield	CA	1,645	Valley Tree & Construction	Potential			
Ventucopa	CA		Santa Barbara County	Potential			
Victorville	CA	600,000	Bureau of Land Management	Potential			
Visalia	CA		Tulare County	Potential			
Walker	CA	8,610	Bureau of Land Management	Potential			
Lancaster	CA	3,185,000	Waste Management of California, Inc.	Potential			
Watsonville	CA		City of Watson D.P.W.	Potential			
Weaverville	CA	147,000	County of Trinity	Potential			
Weed	CA		Santa Fe Pac. Prop. Inc., Catellis Corp	Potential			
Redding	CA	632,098	Shasta County Public Works Department	Potential			
Point Reyes Station	CA	135,488	West Martin Sanitary Landfill, Inc.	Potential			
San Bernardino	CA	5,563	Donald C. Hubbs, Inc.	Potential			
Marvsville	CA			Potential			
Westwood	CA	48,162	W.M. Beaty & Associates, Inc.	Potential			
Yermo	CA	146,898	Bureau of Land Management	Potential			
Davis	CA	8,427,340	Yolo County	Potential			
Yreka	CA		City of Yreka	Potential			
Marvsville	CA	2,268,000	Barbieri, David	Potential			
San Jose	CA		Green Waste Recovery	Potential			
Commerce City	CO	18,052,817	Allied Waste Industries	Shutdown	ED1		2.7
Littleton	CO	30,000,000	Arapahoe County	Shutdown	Biomass Energy Partners		0.8
Erie	CO	13,000,000	Allied Waste (old LWS)	Shutdown			11.7
Pueblo	CO	5,163,653	Zupan Enterprises	Shutdown	Env. Technologies Int'l/Ultasystems		
Fountain	CO	5,218,297	Allied Waste Industries	Candidate			
Golden	CO	3,394,539	Allied Waste Industries	Candidate			
Aurora	CO	11,499,652	City of Denver	Candidate			
Erie	CO	7,036,984	Allied Waste Systems (old LWS)	Candidate			
Fort Collins	CO	12,558,321	Larimer County, Fort Collins, and Loveland	Candidate			
Milner	CO	1,125,701	Twin Landfill Company	Candidate			
Montrose	CO	2,054,466	Montrose County	Candidate			
Grand Junction	CO	8,688,255	Mesa County	Candidate			
Aspen	CO	2,465,911	Pitkin County	Candidate			
Glenwood Springs	CO	1,798,154	City of Glenwood Springs	Candidate			
Dillon	CO	2,680,410	Summit County	Candidate			
Fountain	CO	1,059,137	Landfill Services, Inc.	Candidate			
Colorado Springs	CO	15,562,598	Waste Management, Inc.	Candidate			
Pagosa Springs	CO		Allied Waste Industries	Potential			
	CO	4,689,328	Allied Waste Industries	Potential			

Pueblo	CO	393,114	Mid American Waste	Potential			
Milliken	CO	6,246,093	Waste Services, Inc.	Potential			
	CO			Potential			
Erie	CO		Republic Services, Inc.	Potential			
Longmont	CO	2,711,778	City of Longmont	Potential			
	CO	25,725,664	Waste Management, Inc.	Potential			
Fort Morgan	CO	915,183		Potential			
Ault	CO	831,063	Waste Management, Inc.	Potential			
Penrose	CO	180,000		Potential			
	CO			Potential			
	CO	2,058,908	Templeton Gap Landfill, Inc.	Potential			
	CO	867,423		Potential			
Hartford	CT	2,187,000	Connecticut Resources Recovery Authority	Operational	Minnesota Methane/Hudson Bank/American Gas Co.	2.9	1,300
New Milford	CT	6,700,000	WM	Operational	Bio Energy Partners	3.3	
Groton	CT	2,000,000	Town of Groton	Shutdown	IFC	0.2	
Shelton	CT	2,000,000		Shutdown	Resource Technology Corporation	1.8	
East Windsor	CT	837,000	Boticello, Inc.	Candidate			
Manchester	CT	1,000,000	City of Manchester	Candidate			
New Haven	CT	2,500,000	City of New Haven	Candidate			
Waterbury	CT	1,600,000	Town of Waterbury	Candidate			
Windsor	CT	1,268,000	Town of Windsor	Candidate			
Bridgeport	CT		Bridgeport	Potential			
Bristol	CT	65,000	Bristol	Potential			
Canterbury	CT	546,000	Yaworski, Inc.	Potential			
Danbury	CT	65,000	Danbury	Potential			
East Hartford	CT			Potential			
Ellington	CT		CT Resources Recovery Authority	Potential			
Groton	CT	2,000,000	Town of Groton	Potential			
Manchester	CT			Potential			
Danielson	CT			Potential			
Meriden	CT			Potential			
Montville	CT			Potential			
North Haven	CT			Potential			
Putnam	CT	214,000	Putnam	Potential			
Shelton	CT	2,000,000		Potential			
Torrington	CT			Potential			
Wallingford	CT		Town of Wallingford	Potential			
Windham	CT			Potential			
Wilmington	DE	6,850,000	Delaware Solid Waste Authority	Operational	Cereza Energy		4,600
Wilmington	DE	6,000,000	Delaware Solid Waste Authority	Shutdown	The Bantech Group		
Sandtown	DE	2,310,000	Delaware Solid Waste Authority	Candidate			
Wilmington	DE	6,000,000	Delaware Solid Waste Authority	Candidate			
Jones Crossroads	DE	2,430,000	Delaware Solid Waste Authority	Candidate			
Okeechobee	FL	4,375,800	WMI	Operational	Organic Waste Technologies, Inc.		0.432
Pompano Beach	FL	23,288,000	WMI	Operational	Bio Energy Partners	12.0	
Jacksonville	FL	4,122,058	City of Jacksonville	Operational	Jacksonville City & Electric Authority/Earth Tech	2.4	0.800
Tampa	FL	3,436,000	Hillsborough County, Solid Waste	Operational	Tampa Electric Company (TECO)	0.0	0.029
			Polk County Board of County Commissioners	Operational	Wheelabrator Ridge Energy (WRE)/DTE Biomass		
Winter Haven	FL	5,200,000		Operational	Earth Tech	2.9	1,000
Jacksonville	FL	4,900,000	City of Jacksonville	Operational			
Orlando	FL	14,165,000	Orange County Utilities Division	Operational	DTE Biomass Energy, Inc.	16.3	5,050
Cantonment	FL	4,050,400	County Gov't BOCC	Operational	DTE Biomass		
Archer	FL	849,420	Alachua County Public Works Department	Operational	Jones, Edmunds & Associates, Inc.	2.3	
Daytona Beach	FL	7,569,140		Operational	NEO	3.8	1,300
Cocoa	FL	8,139,201	Brevard County, Solid Waste Department	Construction			
Tampa	FL	2,061,000	Hillsborough County, Solid Waste	Construction			
Jacksonville	FL	2,250,000	Brownfield Properties	Shutdown	Ring Power	0.8	
Ocala	FL	3,432,000	Marion County	Candidate			
Sarasota	FL	4,978,200	Sarasota County, Solid Waste Department	Candidate			
St. Petersburg	FL	2,850,000	Pinellas County, Solid Waste Management	Candidate			
Lecanto	FL	1,000,000	Citrus County Solid Waste Management Division	Candidate			
Sebring	FL	413,432	Highlands County Solid Waste	Candidate			
West Palm Beach	FL	15,598,440	Palm Beach County	Candidate			
Fort Myers	FL	2,125,340	WMI	Candidate			
Vero Beach	FL	3,266,617	Indian River County Solid Waste Disposal District	Candidate			
Miami	FL	18,018,000	WMI	Candidate			
Naples	FL	4,992,000	Collier County	Candidate			
West Palm Beach	FL	4,418,356	Palm Beach County	Candidate			
Geneva	FL	4,800,000	Seminole County	Candidate			
Fort Pierce	FL	1,287,000	St. Lucie County	Candidate			
Nokomis	FL	1,400,000	Sarasota County	Candidate			
Goulds	FL	10,000,000	Metro-Dade Solid Waste Management	Candidate			
Osceola	FL	1,029,000	Osceola County Solid Waste	Candidate			
Campbellton	FL	3,245,441	Waste Management, Inc.	Candidate			
Maxville	FL	4,677,128	Duval County Public Utilities	Candidate			
Tallahassee	FL	1,680,000	Leon County	Candidate			
Winter Garden	FL		Republic Services, Inc.	Potential			
Greenville	FL	204,490	Madison County Solid Waste	Potential			
	FL		Solid Waste Authority of Palm	Potential			

			Beach				
Tropical Gulf Acres	FL	638,509	Charlotte County Mosquito Control	Potential			
	FL			Potential			
Brooksville	FL	1,215,500	Hernando County, Utilities Department	Potential			
	FL	895,573	Putnam County, Sanitation Department	Potential			
Kennedy Space Center	FL	264,030	Kennedy Space Center (NASA)	Potential			
	FL		Pinellas County, Solid Waste Management	Potential			
Manatee	FL	660,660	Manatee County	Potential			
Madison	FL	950,092	Madison County Solid Waste	Potential			
	FL	2,059,200	Bay County, Solid Waste Division	Potential			
Palm City	FL	772,200	Martin County, Solid Waste Management	Potential			
Raiford	FL			Potential			
St. Augustine	FL		Republic Services, Inc.	Potential			
Brooksville	FL	950,000	Hernando County	Potential			
Palatka	FL	434,720	Putnam County, Sanitation Department	Potential			
Palatka	FL	3,103,623	Putnam County	Potential			
New Port Richey	FL	680,000		Potential			
Green Cove Springs	FL	545,688	Clay County Public Works	Potential			
Saint Cloud	FL	514,800	Department of Solid and Hazardous Waste	Potential			
	FL		Public Utilities	Potential			
Bagdad	FL	806,520	Santa Rosa County	Potential			
Kennedy Space Center	FL		Kennedy Space Center (NASA)	Potential			
Miami	FL			Potential			
	FL		Metro-Dade Solid Waste Management	Potential			
Key West	FL	343,200	Monroe County, Environmental Management	Potential			
	FL	1,389,682	Taylor County Public Works	Potential			
St. Augustine	FL	434,720	St. Johns County Solid Waste Management	Potential			
St. Petersburg	FL		Pinellas County, Solid Waste Management	Potential			
Callahan	FL	240,240		Potential			
Crestview	FL	6,542,657		Potential			
Fayetteville	GA	4,214,375	Allied Waste Industries	Operational	Energy Developments, Inc.	1.0	0.800
Macon	GA	3,500,000	City of Macon	Operational	Cherokee Brick & Tile		
Blythe	GA	2,326,908	City of Augusta-Richmond County	Operational	LandGas		1.000
Mauk	GA	2,738,405	Allied Waste Industries	Operational	Energy Developments, Inc.	4.0	2.200
Norcross	GA	4,627,157	WMI	Operational	Bio Energy Partners	2.4	
LaGrange	GA		City of LaGrange	Construction	Interface Flooring Systems, City of LaGrange		0.576
Buford	GA	3,491,863	Allied Waste Industries	Construction	Energy Developments, Inc.	4.0	2.900
Atlanta	GA	543,050	WMI	Shutdown	WMI		0.089
Atlanta	GA	1,667,108	WMI	Shutdown	WMI		0.240
	GA	1,000,000		Shutdown	WMI		
Lawrenceville	GA	1,282,907	WMI	Candidate			
Atlanta	GA	2,371,617	City of Atlanta	Candidate			
Atlanta	GA	2,341,823	City of Atlanta	Candidate			
Grovetown	GA	2,395,800	Columbia County SWM	Candidate			
Cartersville	GA	1,800,000	Bartow County Department of Solid Waste	Candidate			
Conley	GA	982,305	Allied Waste Industries	Candidate			
Atlanta	GA	543,050	WMI	Candidate			
Columbus	GA	2,354,569	City of Columbus	Candidate			
Albany	GA	1,700,000	Dougherty County	Candidate			
Savannah	GA	643,608	Waste Management of Georgia, Inc.	Candidate			
Chickamauga	GA	1,050,000	Walker County	Candidate			
Dalton	GA	2,695,000	Dalton-Whitfield Regional Solid Waste Management Authority (DWRSWMA)	Candidate			
Valdosta	GA	2,657,359	Onyx Waste Services, Inc.	Candidate			
Ball Ground	GA		Waste Management, Inc.	Candidate			
Ellenwood	GA	7,194,141	DeKalb County Public Works Dept Sanitation Division	Candidate			
Atlanta	GA	1,667,108	WMI	Candidate			
Winder	GA	2,838,500	Republic Services, Inc.	Candidate			
Atlanta	GA	6,000,000	Allied Waste Industries	Candidate			
Conley	GA	5,266,000	WMI	Candidate			
	GA	272,750		Potential			
Jesup	GA		Republic Services, Inc.	Potential			
	GA			Potential			
Forsyth	GA	1,297,000	City of Forsyth	Potential			
	GA			Potential			
	GA			Potential			
	GA		Clarke County	Potential			
Fayetteville	GA		Fayette County	Potential			
	GA			Potential			
Gainesville	GA	1,149,551	Hall County, Public Works Department	Potential			
	GA			Potential			
Kathleen	GA	815,788	Houston County	Potential			
Midway	GA		Liberty County SWA	Potential			
Dunwoody	GA		Fulton County	Potential			

Winder	GA		Republic Services, Inc.	Potential			
	GA			Potential			
Griffin	GA		Republic Services, Inc.	Potential			
Cedartown	GA		Polk County Board of Commissioners	Potential			
	GA			Potential			
	GA			Potential			
Savannah	GA		Republic Services, Inc.	Potential			
	GA	902,256	Bartow County	Potential			
Lovejoy	GA	598,122		Potential			
Macon	GA		Republic Services, Inc.	Potential			
	GA			Potential			
	GA	118,389		Potential			
Kailua	HI	4,500,000	City and County of Honolulu	Shutdown	GRS	3.3	2,500
Puunene	HI	2,450,000	Maui County	Candidate			
Kealahou	HI	500,000	Hawaii County	Candidate			
Waimanalo	HI	1,310,000	City and County of Honolulu	Candidate			
	HI	2,122,147	City and County of Honolulu	Candidate			
Kekaha	HI	1,492,886	Kauai County	Candidate			
Hilo	HI	2,000,000	Hawaii County	Candidate			
Kapolei	HI	5,070,000	City and County of Honolulu	Candidate			
Waikoloa	HI	900,000	Hawaii County	Candidate			
Hana	HI		Maui County	Potential			
Kaneohe Bay	HI		United States Marine Corps	Potential			
Lanai City	HI		Maui County	Potential			
Kaunakakai	HI		Maui County	Potential			
Nanakuli	HI		Private Land Company	Potential			
Princeville	HI		Kauai County	Potential			
Coeur d'Alene	ID	1,000,000	County of Kootenai	Operational			
Pocatello	ID	2,000,000	Bannock County Solid Waste Dept.	Candidate			
Dayton	ID		Franklin County	Candidate			
Boise	ID		Ada County	Potential			
Weippe	ID		Aitco, Inc.	Potential			
Mountain Home	ID		Elmore County	Potential			
Shellev	ID		Bingham County	Potential			
Blackfoot	ID		Bingham County	Potential			
Idaho City	ID		Garden Valley Non-Municipal Landfill	Potential			
Idaho Falls	ID		Bonneville County	Potential			
Bonniers Ferry	ID		Boundary County	Potential			
Arco	ID		Butte County	Potential			
Howe	ID		Butte County	Potential			
Nampa	ID		Canyon County	Potential			
Terreton	ID			Potential			
Council	ID		Adams County	Potential			
Boise	ID		Garden Valley Non-Municipal Landfill	Potential			
Glenns Ferry	ID		Elmore County	Potential			
Twin Falls	ID		Twin Falls County	Potential			
Island Park	ID		Fremont County	Potential			
Mud Lake	ID		Jefferson County	Potential			
Moscow	ID		Latah Sanitation Inc	Potential			
Salmon	ID		Lemhi County	Potential			
Bannock	ID		Bannock County	Potential			
Montpelier	ID		Bear Lake County	Potential			
Salmon	ID		Lemhi County	Potential			
Molad City	ID		Southern Idaho Regional Solid Waste	Potential			
Pavette	ID		Pavette County	Potential			
Burley	ID		Southern Idaho Regional Solid Waste	Potential			
St Anthony	ID		Fremont County	Potential			
Driggs	ID		Teton County	Potential			
Belleville	IL	1,000,000	Allied Waste	Operational	DTE Biomass	0.8	0.522
Danville	IL	2,301,000	Brickyard Disposal and Recycling (Allied Waste Systems)	Operational	U.S. Energy Biogas Corp.	4.0	
Rockdale	IL	1,596,000	CDT Landfill Corporation	Operational	KMS Power Income Trust Fund	1.6	
Chicago	IL	21,690,122	WMI	Operational	Bio Energy Partners	6.6	
Grayslake	IL	6,500,000	WMI	Operational	U.S. Energy Biogas Corp.	7.8	
Dixon	IL	1,140,780	Allied Waste Industries, Inc.	Operational	U.S. Energy Biogas Corp.	1.5	
Charleston	IL	824,600	Environmental Reclamation Company (Allied Waste Industries)	Operational	DTE Biomass	2.0	
Morris	IL	6,387,000	Allied Waste Industries	Operational	U.S. Energy Biogas Corp.	3.9	
Naperville	IL	18,707,345	Forest Preserve District of DuPage County	Operational	Bio Energy Partners	6.0	
Antioch	IL	2,000,000	Waste Management of Illinois, Inc.	Operational	RMT, Inc.	0.4	1,296
Chebanse	IL	2,198,000	WMI	Operational	Bio Energy Partners	1.6	
Northbrook	IL	10,000,000	WMI	Operational	Bio Energy Partners	9.9	
Park Ridge	IL	11,183,620	Land & Lakes Company	Operational	U.S. Energy Biogas Corp.	1.5	
Chicago	IL	15,000,000	Stoney Island Reclamation Co. (Land and Lakes Company)	Operational	U.S. Energy Biogas Corp.	3.5	
Dolton	IL	4,202,000	Land and Lakes Company	Operational	U.S. Energy Biogas Corp.	1.5	
Lansing	IL	1,300,000	Prarie Preservation	Operational	Resource Technology Corporation	2.2	
Dixon	IL	4,030,000	Allied Waste Industries, Inc.	Operational	U.S. Energy Biogas Corp.	1.5	
Pontiac	IL	15,132,820	American Disposal Services of Illinois (Allied Waste Industries)	Operational	Resource Technology Corporation	15.6	
McCook	IL	7,000,000	American Grading	Operational	Resource Technology Corporation	3.6	
Hanover Park	IL	24,468,450	Forest Preserve District of DuPage	Operational	GRS	17.4	14,400

			County				
East St. Louis	IL	16,000,000	WMI	Operational	Waste Management, Inc.	2.4	
Davis Junction	IL	1,858,300	Onyx Orchard Hills Landfill	Operational	GRS	1.8	
Decatur	IL	8,141,092	Onyx Valley View Landfill, Inc.	Operational	KMS Power Income Trust Fund	1.6	
Zion	IL	10,000,000	Onyx Zion Landfill	Operational	Energy Developments, Inc.	5.4	2,900
New Milford	IL	4,552,694		Operational			
Brimfield	IL	4,200,000	Peoria City/County	Operational	Resource Technology Corporation	1.8	
Milan	IL	4,565,450	Millenium Waste, Inc.	Operational	GRS	2.0	0.750
Roxana	IL	3,252,000	Roxana Landfill Authority, Inc. (Allied Waste Industries)	Operational	DTE Biomass	2.7	0.464
Springfield	IL	1,123,500	Sangamon Valley Landfill, Inc. (Allied Waste Industries)	Operational	Resource Technology Corporation	1.4	
Batavia	IL	13,000,000	Kane County Department of Environmental Management	Operational	Bio Energy Partners	6.0	3,600
Hillside	IL	740,200	Congress Development Company (CDC)	Operational	Resource Technology Corporation	15.6	
South Barrington	IL	1,000,000	Allied Waste Industries	Operational	GRS	1.6	0.800
Streator	IL	2,500,000	Streator Area Landfill, Inc. (Allied Waste Industries)	Operational	U.S. Energy Biogas Corp.	1.9	
East Peoria	IL	4,049,900	WMI	Operational	Bio Energy Partners	1.6	
East Moline	IL	2,052,000	Upper Rock Island County LF, Inc. (Allied Waste Industries)	Operational	U.S. Energy Biogas Corp.	4.0	
	IL		Land & Lakes Company	Operational	U.S. Energy Biogas Corporation	1.0	
South Elgin	IL	7,800,000	Oak Brook Bank, Trust #8-1735 (Waste Management, Inc.)	Operational	Bio Energy Partners	1.6	
Blue Island	IL			Shutdown	GSF Energy		
Morris	IL	1,022,000	City of Morris	Shutdown	KMS		
Westchester	IL	1,922,000	Catholic Charities/Sexton	Candidate			
Beecher	IL	5,500,000	Beecher Development Company	Candidate			
Bloomington	IL	1,000,000	Sexton	Candidate			
Clinton	IL	1,223,000	Peoria Disposal Company	Candidate			
Greenville	IL	1,581,904	D & L Landfill, Inc.	Candidate			
Dekalb	IL	4,176,117	WMI	Candidate			
Hoopeston	IL		Allied Waste Industries	Candidate			
Macomb	IL	235,000	WMI	Candidate			
Taylorville	IL	10,200,000	Waste Management, Inc.	Candidate			
Milford	IL	13,736,372	Allied Waste Industries	Candidate			
Kewanee	IL	1,000,000	City of Kewanee	Candidate			
Oneida	IL	1,256,000	Knox County Landfill Committee	Candidate			
	IL	1,699,012	Laidlaw Waste Systems	Candidate			
Ottawa	IL	1,674,800	LandComp Corporation (Allied Waste Industries)	Candidate			
Effingham	IL	1,152,200	Landfill 33 Ltd.	Candidate			
Elwood	IL	693,240	WMI	Candidate			
Litchfield	IL	2,037,840	Envotech Illinois (Allied Waste Industries)	Candidate			
Des Plaines	IL	1,000,000	Catholic Charities/Sexton	Candidate			
Villa Grove	IL	1,307,103		Candidate			
Ottawa	IL	910,000	States Land Improvement	Candidate			
Chicago	IL	9,766,782	Stryker International	Candidate			
Brimfield	IL	1,415,680	Peoria City/County	Candidate			
Baylis	IL	1,862,000	Pike County Landfill, Inc.	Candidate			
Morrison	IL	1,827,000	Whiteside County	Candidate			
Jerseyville	IL	584,700	RCS, Inc. (Allied Waste Industries)	Candidate			
Harrisburg	IL	1,249,000	Saline County Landfill, Inc.	Candidate			
Granite City	IL	5,850,000	WMI	Candidate			
Desoto	IL	1,491,000	Southern Illinois Regional Landfill, Inc. (Allied Waste Industries)	Candidate			
Fairview	IL	11,237,000	BFI (Allied Waste Industries)	Candidate			
Taylor Ridge	IL	4,500,000	ESG Watts	Candidate			
Viola	IL	900,000	ESG Watts, Inc.	Candidate			
Fairfield	IL	1,309,600	Wayne County Landfill, Inc. (Allied Waste Industries)	Candidate			
Plainfield	IL	2,068,400	Waste Management, Inc.	Candidate			
Rockford	IL	10,845,380	Winnebago Reclamation Services, Inc.	Candidate			
	IL	789,236		Potential			
Bloomington	IL	887,000	American Disposal Services of Illinois, Inc.	Potential			
	IL	1,480,204		Potential			
Belvidere	IL	1,000,000	Dwane	Potential			
Belleville	IL	6,014,100	BFI Waste Systems of North America	Potential			
Marissa	IL	900,000	Waste Management, Inc.	Potential			
White Hall	IL	12,919,592		Potential			
Freeport	IL	780,000	City of Freeport	Potential			
Herrin	IL	1,980,816	City of Herrin	Potential			
Donovan	IL	8,887,255	Allied Waste Industries	Potential			
Sumner	IL	400,000	Waste Management, Inc.	Potential			
Olney	IL		Eastern Environmental Services, Inc.	Potential			
Pekin	IL	2,059,340	Floyd and Frances Simpson	Potential			
Peoria	IL	889,200	Peoria Disposal Company	Potential			
DuQuoin	IL	0	GERE Properties, Inc.	Potential			
Wilmington	IL	0	Will County Land Use Department	Potential			
Quincy	IL	2,187,359	Adams County	Potential			
Rochelle	IL	711,740	City of Rochelle	Potential			
Salem	IL	107,200	City of Salem	Potential			
Washington	IL	1,529,502	Peoria Disposal Company	Potential			
	IL	2,083,875		Potential			

Thompsonville	IL	3,000	Landfill L.L.C.	Potential			
Argos	IN	2,628,000	Allied Waste Industries	Operational	Allied Waste Industries	0.7	
Michigan City	IN	10,196,460	WMI	Operational	Bio Energy Partners	2.4	
Elkhart	IN	4,091,598	WMI	Operational	OWT		
Indianapolis	IN	5,903,986	Allied Waste Industries	Operational	GRS	4.4	
Evansville	IN	5,000,000	Allied Waste	Operational		0.2	0.144
Monticello	IN	1,400,000	WMI	Operational	OWT		
Fort Wayne	IN	9,115,011	Republic Industries, Inc.	Operational	Toro Energy		2.520
Wyatt	IN	5,425,098	WMI	Operational	Bio Energy Partners	3.2	
Modoc	IN	3,454,330	Randolph Farms, Inc	Operational	Randolph Farms, Inc.		
Indianapolis	IN	15,000,000	Balkema	Operational	Granger Energy		2.400
Indianapolis	IN	15,000,000	Balkema	Operational	Granger Energy/Indianapolis Power & Light	0.0	0.200
Indianapolis	IN	15,000,000	Balkema	Operational	Granger Energy		0.300
Indianapolis	IN	15,000,000	Balkema	Operational	Granger Energy	5.0	0.800
Danville	IN	5,336,140	WMI	Operational	Bio Energy Partners	3.2	
Hobart	IN	6,692,406	WMI	Operational	WMI	2.4	
Richmond	IN	1,900,000	Richmond Sanitary District	Construction		1.6	
Fort Wayne	IN	9,115,011	Republic Industries, Inc.	Shutdown	Republic	0.4	0.200
Fort Wayne	IN	9,115,011	Republic Industries, Inc.	Shutdown	Toro Energy		1.575
Morristown	IN	2,437,688	Caldwell	Candidate			
Clarksville	IN	4,033,944	Clark-Floyd	Candidate			
Goshen	IN	2,624,888	Elkhart County	Candidate			
Carthage	IN	1,500,000	Caldwell's Gravel Company	Candidate			
	IN	1,801,536	Hayes Landfill, Inc.	Candidate			
Portland	IN	1,770,714	WMI	Candidate			
Medora	IN	1,594,000	Rumpke Waste Inc.	Candidate			
	IN	3,712,324		Candidate			
Logansport	IN	3,930,000	Indiana Waste Systems	Candidate			
Winslow	IN	1,021,300	Onyx Blackfoot LP	Candidate			
Crothersville	IN	1,639,263		Candidate			
	IN	1,538,482		Candidate			
Pimento	IN	595,000	Republic Services, Inc.	Candidate			
Fort Wayne	IN	7,125,327	National Serv-All, Inc.	Candidate			
Terre Haute	IN	5,000,000	Republic Services, Inc.	Candidate			
Wabash	IN	4,488,770	Republic Services, Inc.	Candidate			
Worthington	IN	2,115,000	Republic Services of IN	Candidate			
Decatur	IN	2,774,000	Adams County SWMD	Potential			
	IN		Alcoa-Warrick Operation	Potential			
Columbus	IN	1,785,030	Bartholomew County	Potential			
Vincennes	IN		Eastern Environmental Services, Inc.	Potential			
	IN	201,058	Blackford County	Potential			
	IN	288,653	Brown County	Potential			
Eidenburgh	IN	16,526	U.S. Government	Potential			
Center Point	IN	2,529,693	Mid-American Waste Systems	Potential			
Terre Haute	IN	2,145,000	Laidlaw Waste Systems	Potential			
Crane	IN	53,568	U.S. Government	Potential			
Montgomery	IN	452,122	Daviess County Commissioners	Potential			
	IN	640,418	Decatur County	Potential			
Modoc	IN	229,817	Decatur Hills, Incorporated	Potential			
	IN	1,286,743		Potential			
	IN	196,596	Knox County Commissioners	Potential			
Covington	IN	412,082	Fountain County Commissioners	Potential			
South Bend	IN	268,000	Weaver Boos & Gordon, Inc.	Potential			
	IN	101,760		Potential			
Gary	IN	4,813,700	City of Gary	Potential			
	IN	255,029	Gibson County	Potential			
	IN	89,407		Potential			
	IN	11,492		Potential			
	IN	129,466	Griffith City	Potential			
Huntington	IN	253,110	City of Huntington	Potential			
	IN	26,325	U.S. Government	Potential			
	IN		Inland Container Corporation	Potential			
	IN	346,484	Jasper City	Potential			
	IN	410,322		Potential			
Clinton	IN	110,040	Kanizer	Potential			
	IN	99,220	LaGrange County Commissioners	Potential			
	IN	534,384		Potential			
Anderson	IN			Potential			
	IN	912,240	McCarty	Potential			
Frankfort	IN	397,870	MidWest Disposal, LLC	Potential			
Bloomington	IN		Monroe County Solid Waste Mgmt. District	Potential			
Munster	IN	246,048	Munster City	Potential			
	IN		Allied Waste Industries	Potential			
	IN	4,319	U.S. Government	Potential			
	IN	927,741		Potential			
	IN			Potential			
	IN	1,331,427	North Wells	Potential			
Spencer	IN	98,626	Owen County Commissioners	Potential			
	IN	165,243	Pike County Commissioners	Potential			
Claypool	IN	978,190	Ransbottom	Potential			
	IN			Potential			

	IN			Potential			
	IN	967,974	Wells County	Potential			
	IN	208		Potential			
Huntingburg	IN	3,191,122		Potential			
	IN	262,710	American Disposal	Potential			
	IN	2,900,286		Potential			
Mt. Vernon	IN		Republic Services, Inc.	Potential			
Cass	IN	315,370	Sullivan County Commissioners	Potential			
	IN	1,359,893		Potential			
	IN		Continental Waste	Potential			
	IN	287,672	Tipton County Commissioners	Potential			
	IN	185,900		Potential			
	IN	31,950	Warren County	Potential			
Warrick	IN	229,306	Warrick County	Potential			
Salem	IN	260,357	Washington County Commissioners	Potential			
Clinton	IN	659,792	West Clinton	Potential			
	IN			Potential			
Cedar Rapids	IA	3,800,000	City of Cedar Rapids	Operational	EnviroGas		2.300
Mitchellville	IA	22,213,605	Des Moines Metropolitan Area Solid Waste Agency	Operational	Bio Energy Partners	6.4	
Buffalo	IA	2,277,600	Scott Area Solid Waste Management Commission	Operational	NST Landfill Gas of Iowa		
Waterloo	IA	1,821,820	Black Hawk County Solid Waste Mgmt. Comm.	Candidate			
Lake Mills	IA	4,528,000	WMI	Candidate			
Clinton	IA	3,697,808	Clinton County Area Solid Waste Agency	Candidate			
West Burlington	IA	2,597,642	Des Moines County Regional Solid Waste Comm.	Candidate			
Dubuque	IA	1,600,000	Dubuque Metro Area Solid Waste Agency	Candidate			
Fayette	IA	275,484	Fayette County Solid Waste Management Commission	Candidate			
Ft. Dodge	IA	1,000,000	City of Ft. Dodge	Candidate			
Fort Madison	IA	1,099,900	Lee County Solid Waste Management Commission	Candidate			
Tracy	IA	1,127,560	South Central Iowa Solid Waste Agency	Candidate			
Sheldon	IA	1,092,500	Northwest Iowa Solid Waste Agency	Candidate			
Sioux City	IA	1,144,000	Siouxland Environmental Services, Inc.	Candidate			
Decorah	IA	1,000,000	Winnebago County	Candidate			
Marion	IA	858,000	Blairstown Solid Waste Agency	Potential			
Boone	IA	718,624	Black Hawk County Solid Waste Mgmt. Comm.	Potential			
Spirit Lake	IA		Sanfill of Iowa	Potential			
Dillon	IA	726,000	Marshall County Solid Waste Management Comm.	Potential			
	IA	1,212,640	Muscatine County Solid Waste Management Agency	Potential			
Newton	IA	840,000	Department of Public Works	Potential			
Ottumwa	IA	828,000	Ottumwa-Wapello Solid Waste Commission	Potential			
Malcolm	IA	326,040	South Central Iowa Solid Waste Mgmt. Assn	Potential			
Douds	IA	104,866	Van Buren Solid Waste Management Agency	Potential			
Wichita	KS	10,737,610	City of Wichita	Operational	DTE Biomass Energy		3.600
Kansas City	KS	9,935,717	Allied Waste Industries, Inc.	Operational	Wyandotte County		0.030
Shawnee	KS	18,000,000	Deffenbaugh Disposal	Operational	South Texas Treaters		
Garden City	KS	1,339,005	Finney County	Candidate			
Newton	KS	1,511,401	Harvey County	Candidate			
	KS	1,904,204	N.R. Hamm Quarry	Candidate			
Hutchinson	KS	2,624,021	Reno County	Candidate			
Salina	KS	2,797,917	City of Salina	Candidate			
Columbus	KS	1,514,421	Allied Waste Industries, Inc.	Candidate			
Iola	KS	607,729	Allen County	Potential			
Great Bend	KS	566,862	Barton County	Potential			
El Dorado	KS		City of Augusta	Potential			
Chanute	KS		City of Chanute	Potential			
	KS	175,463		Potential			
Clay Center	KS		Clay County	Potential			
Burlington	KS	178,752	Coffee County	Potential			
	KS	1,095,150	Cowley County	Potential			
	KS	1,661,063	Oak Grove Landfill	Potential			
Dodge City	KS	706,013	Ford County	Potential			
Hill City	KS		Graham County	Potential			
Greensburg	KS	76,076	Kiowa County	Potential			
	KS	217,268	Grosse Construction Company, Inc.	Potential			
Beloit	KS	217,750	Mitchell County	Potential			
Arcadia	KS		Crawford County	Potential			
Phillipsburg	KS	148,412	Phillips County	Potential			
Pratt	KS	379,181	Pratt County	Potential			
	KS		Refuse Service, Inc.	Potential			
	KS	296,015	Resource Recovery, Inc.	Potential			
Stockton	KS	270,635	Rooks County	Potential			
Russell	KS	230,165	Russell County	Potential			
Liberal	KS	870,365	Seward County	Potential			
Topeka	KS		Waste Management of Kansas, Inc.	Potential			

Wakeeny	KS	133,956	Trego County	Potential			
Leoti	KS	111,490	Wichita County	Potential			
Walton	KY	4,582,464	Bavarian Waste Services, Inc.	Operational	Owen Electric Cooperative/ East Kentucky Power Cooperative (EKPC)	3.2	
	KY	394,995	Allied Waste	Operational	East Kentucky Power Cooperative (EKPC)	3.2	
Lily	KY	2,870,686	Waste Management of Kentucky, Inc.	Operational	East Kentucky Power Cooperative (EKPC)	4.0	
Louisville	KY	35,046,257	Waste Management of Kentucky, Inc.	Operational	Toro Energy		0.720
Frankfort	KY	4,142,152	Allied Waste Industries	Candidate			
Hopkinsville	KY	2,439,142	City of Hopkinsville	Candidate			
Owensboro	KY	2,599,178	City of Owensboro	Candidate			
Ashland	KY	4,500,516	Cooksey Brothers Disposal Company, Inc.	Candidate			
Owensboro	KY	1,900,000	Daviess County	Candidate			
Morganfield	KY	1,235,835	Dorzi Company, Inc. (Republic Services, Inc.)	Candidate			
Williamstown	KY	1,857,438	Epperson Waste Disposal (Republic Services, Inc.)	Candidate			
Glasgow	KY	1,414,608	City of Glasgow	Candidate			
Morehead	KY	782,676	Local Sanitation, Inc.	Candidate			
Maysville	KY	1,500,000	Mason County	Candidate			
Hazard	KY		Perry County Fiscal Court	Candidate			
	KY	1,255,031	Pulaski Grading, Inc.	Candidate			
Jeffersonville	KY	1,266,307	Rumpke Waste, Inc.	Candidate			
Butler	KY	1,423,548	Rumpke Waste, Inc.	Candidate			
	KY	2,000,000	Waste Management, Inc.	Candidate			
Stanford	KY	3,502,499	Republic Services, Inc.	Candidate			
Sulphur	KY	4,664,272	Republic Industries, Inc.	Candidate			
Taylorsville	KY	1,295,153	Williams Sanitation Service	Candidate			
	KY	5,691,119	Waste Management of Kentucky, Inc.	Candidate			
Bowling Green	KY	2,574,000	City of Bowling Green	Potential			
	KY		Madison County	Potential			
Martin	KY	346,320	Floyd County Fiscal Court	Potential			
	KY	1,576,656	Hardin County- Fiscal Court	Potential			
	KY	728,973	Jones Sanitation, Inc.	Potential			
Louisville	KY			Potential			
Calvert City	KY	980,525	LWD Sanitary Landfill, Inc.	Potential			
	KY	1,718,758	McCracken County	Potential			
Beaver Dam	KY	785,803	Republic Services, Inc.	Potential			
Pikeville	KY	800,000	Pike County Fiscal Court	Potential			
Stanford	KY	429,000		Potential			
Irvine	KY	562,051	Waste Management of Kentucky, Inc.	Potential			
Keithville	LA	1,194,765	City of Shreveport	Operational	Renovar Energy Corp.		2.592
Baton Rouge	LA	10,174,562	East Baton Rouge Parish	Construction	Renovar		
Keithville	LA	1,194,765	City of Shreveport	Shutdown	Renovar Energy Corp.		1.700
Sorrento	LA	1,609,759	Allied Waste Industries	Candidate			
West Avondale	LA	3,066,630	Southern Recovery Management, Inc.	Candidate			
Welsh	LA	1,757,000	Jefferson Davis Parish LF Commission	Candidate			
Avondale	LA	7,153,000	Jefferson Parish	Candidate			
Jena	LA	257,455	IESI, Inc.	Candidate			
Monroe	LA	3,600,000	Waste Management of Louisiana, LLC	Candidate			
Lafayette	LA	2,000,000	City of Lafayette	Candidate			
Zachary	LA	5,564,250	City of Baton Rouge/Parish of East Baton Rouge	Candidate			
Westwego	LA	3,300,000	River Birch, Inc.	Candidate			
Washington	LA	1,000,000	St. Landry Parish Solid Waste Commission	Candidate			
Independence	LA	1,000,000	Tangipahoa Parish Police Jury	Candidate			
Walker	LA	1,126,745	Livingston Parish Council	Candidate			
Egan	LA	600,249	Acadia Parish Police Jury	Potential			
Avondale	LA	7,407,540	Allied Waste Industries	Potential			
Franklinton	LA	284,020	Washington Parish Police Jury	Potential			
	LA	2,690,454	Allied Waste Industries	Potential			
Mansfield	LA	176,724	Desoto Parish Police Jury	Potential			
Franklin	LA	553,808	St. Mary Parish Police Jury	Potential			
Dubach	LA	90,079	Lincoln Parish Police Jury	Potential			
New Orleans	LA	3,567,455	City of New Orleans	Potential			
Torbert	LA		Western Waste	Potential			
Many	LA	305,415	Sabine Parish Police Jury	Potential			
	LA	51,628	St. James Parish Council	Potential			
Covington	LA	408,980	St. Tammany Parish Police Jury	Potential			
St. Joseph	LA	171,664	Tensas Parish Police Jury	Potential			
Houma	LA	963,058	Terrebonne Parish Consolidated Govt.	Potential			
Farmerville	LA	414,621	Union Parish Police Jury	Potential			
Venice	LA	134,856	South Louisiana Environmental Control	Potential			
Abbeville	LA	461,500	Vernillion Parish Police Jury	Potential			
Dixie Inn	LA	500,000	Webster Parish Police Jury	Potential			
Oak Grove	LA	128,513	West Carroll Police Jury	Potential			
Monroe	LA	276,438	Allied Waste Industries	Potential			
Sulphur	LA	973,484	Allied Waste Industries	Potential			
Norridgewock	ME	2,500,000	WMI	Candidate			
Augusta	ME	677,167	City of Augusta	Candidate			

Hampden	ME	2,820,619	Casella Waste Systems, Inc.	Candidate			
Bath	ME	72,000	City of Bath	Potential			
Brunswick	ME	377,400	Town of Brunswick	Potential			
Lewiston	ME	1,250,000	City of Lewiston	Potential			
Forks	ME	6,000	Caratunk, West Forks	Potential			
Fort Fairfield	ME	750,000	Tri-Community LF	Potential			
Greenville	ME	60,000	Town of Greenville	Potential			
Presque Isle	ME	252,000	City of Presque Isle	Potential			
Portland	ME	300,000	Regional Waste Systems	Potential			
Upper Marlboro	MD	8,900,000	Prince George's County Government, MD	Operational	Prince George's County Govt., MD	3.5	2,110
Upper Marlboro	MD	8,900,000	Prince George's County Government, MD	Operational	Prince George's County Govt., MD	2.8	1,332
Upper Marlboro	MD	8,900,000	Prince George's County Government, MD	Operational	Prince George's County Govt., MD		1,332
Rockville	MD	4,800,000	Montgomery County	Operational	Covanta Power Pacific, Inc.	2.0	
Bowie	MD	5,125,946	Prince George's County	Operational	Toro Energy		1,900
Laytonsville	MD	6,874,060	Montgomery County	Shutdown	PEPCO		2,592
Baltimore	MD	1,400,000	City of Baltimore	Shutdown			
Marriottsville	MD	3,518,784	Howard County	Candidate			
Newark	MD	900,000	Worcester County	Candidate			
White Marsh	MD	3,800,000	Baltimore County	Candidate			
Fort Meade	MD	1,074,376		Candidate			
Elkton	MD	1,286,642	Cecil County	Candidate			
Severn	MD	6,300,000	Anne Arundel County	Candidate			
Salisbury	MD	2,765,002		Candidate			
Laytonsville	MD	6,874,060	Montgomery County	Candidate			
Curtis Bay	MD	7,640,000	Baltimore City Bureau of Solid Waste	Candidate			
Frederick	MD	3,400,000	Board of County Commissioners	Candidate			
Hagerstown	MD	2,055,978	Washington County	Candidate			
California	MD	2,250,266	St. Mary's County	Candidate			
Annapolis	MD	858,000		Potential			
Lusby	MD	103,996	Calvert County	Potential			
	MD	595,000		Potential			
Beulah	MD	476,190	Dorchester County	Potential			
Baltimore	MD	490,000		Potential			
	MD	700,000		Potential			
Eastern Waldorf	MD	160,000	Charles County	Potential			
Fort Detrick	MD		U.S. Army	Potential			
	MD	40,963		Potential			
Street	MD	843,502		Potential			
	MD	1,160,000	Caroline County	Potential			
Perry Hall	MD		Republic Services, Inc.	Potential			
	MD	480,000	Carroll County Environmental Services	Potential			
Easton	MD	433,930	Maryland DNR	Potential			
Frostburg	MD	501,140	Chambers Waste Systems	Potential			
	MD	610,000		Potential			
Westminster	MD	496,052	Carroll County Environmental Services	Potential			
	MD	1,500,000		Potential			
	MD	28,000		Potential			
Oakland	MD	800,000		Potential			
	MD			Potential			
	MD	671,000		Potential			
	MD	1,470,000		Potential			
	MD	447,610		Potential			
Baltimore	MD			Potential			
Springfield	MA	2,200,000	City of Springfield	Operational	U.S. Energy Biogas Corp.		
Chicopee	MA	1,000,000	Allied Waste Industries	Operational	GRS	1.8	1,000
Attleboro	MA	1,000,000	Attleboro Landfill, Incorporated	Operational	Highland Power Co.	1.6	0,720
Barre	MA	1,671,070	WMI	Operational	U.S. Energy Biogas Corp.	1.0	
Braintree	MA	1,000,000	Town of Braintree	Operational	IFC/Braintree Electric & Light	0.2	
Chicopee	MA	4,901,700	WMI	Operational	Ameresco	5.7	
Fall River	MA	5,400,200	Allied Waste Industries	Operational	GRS	5.2	
Fall River	MA	5,400,200	Allied Waste Industries	Operational	GRS	2.1	
Westminster	MA	4,254,300	WMI	Operational	NEO	1.8	1,700
Granby	MA	2,995,829	WMI	Operational	Industrial Power Services Corp.	1.7	1,224
Halifax	MA	2,900,000	Allied Waste Industries	Operational	GRS	1.8	1,200
Lowell	MA	1,335,800	City of Lowell	Operational	Minnesota Methane/Hudson Bank; American Gas Co.	0.8	0,200
East Bridgewater	MA	2,970,729	Allied Waste Industries	Operational	GRS	3.6	2,500
East Bridgewater	MA	2,970,729	Allied Waste Industries	Operational	GRS	1.8	1,300
Plainville	MA	7,095,200	Allied Waste Industries	Operational		5.6	3,600
Randolph	MA	1,812,096	Allied Waste Industries	Operational	GRS	2.7	2,200
Taunton	MA	1,611,694	City of Taunton	Operational	NEO	1.9	1,000
Plainville	MA	7,095,200	Allied Waste Industries	Shutdown	DTE Biomass		
Bourne	MA	1,000,000	Town of Bourne	Candidate			
Dartmouth	MA	268,000	New Bedford Refuse District	Candidate			
Dartmouth	MA	1,259,650	Town of Dartmouth	Candidate			
Northampton	MA	2,565,237	City of Northampton	Candidate			
Clinton	MA	1,500,000	Town of Clinton	Candidate			
Westfield	MA		City of Westfield	Candidate			
Acushnet	MA	900,000	City of Acushnet	Potential			
Amherst	MA	525,096	Town of Amherst	Potential			

Billerica	MA		PRP Group	Potential			
Chatham	MA	230,354	Town of Chatham	Potential			
Peabody	MA		GCR	Potential			
Worcester	MA	2,000,000	City of Worcester, Department of Public Works	Potential			
Leicester	MA		Town of Leicester	Potential			
Marion	MA		Town of Marion	Potential			
Marshfield	MA	774,189	Town of Marshfield	Potential			
Nantucket	MA	890,568	Town of Nantucket	Potential			
New Bedford	MA	9,516,000		Potential			
Buzzards Bay	MA		Mass Air National Guard	Potential			
Palmer	MA		Town of Palmer	Potential			
Hudson	MA	711,360	United Waste Systems	Potential			
South Hadley	MA		Town of South Hadley	Potential			
Winchendon	MA		Town of Winchendon	Potential			
Adrian	MI	1,000,000	Great Lakes Waste Services/Allied Waste	Operational	DTE Biomass/LES	2.4	1,300
Ann Arbor	MI	2,740,000	City of Ann Arbor	Operational	CTR/LES	1.6	
Montrose	MI	2,000,000	Republic Services, Inc.	Operational	Granger Electric Company	1.6	
Marshall	MI	6,876,300	Allied Waste, Inc.	Operational	GRS	2.7	3,800
New Boston	MI	1,000,000	Republic Services, Inc.	Operational	LES - Sumpter Energy	6.4	
Grand Blanc	MI	4,000,000	Allied Waste Industries	Operational	Granger Electric Company	3.2	
Belleville	MI	13,600,000	WMI	Operational	LES - Sumpter Energy	8.0	
Orion	MI	5,700,000	WMI	Operational	Toro Energy		
Three Oaks	MI	5,101,348	Republic Services, Inc.	Operational	EMCON/Organic Waste Technologies, Inc.		
Grand Ledge	MI	4,300,000	Granger Waste Management Corp.	Operational	Granger Electric Company	4.0	
Lansing	MI	4,000,000	Granger Waste Management Corp.	Operational	Granger Electric Company	3.2	
New Hudson	MI	1,000,000	Allied Waste Industries	Operational	GRS	4.5	2,600
Auburn Hills	MI	3,000,000		Operational	Toro Energy		
Northville	MI	25,000,000	Onyx Arbor Hills Landfill	Operational	GRS	17.4	11,000
Coopersville	MI	9,000,000	Allied Waste Industries	Operational	Granger Electric Company	4.8	
Montrose	MI	2,000,000	WMI	Operational	NANR	2.4	
Lenox	MI	5,000,000	WMI	Operational	LES - Sumpter Energy	4.0	
Riverview	MI	15,000,000	City of Riverview	Operational	DTE Biomass	6.6	4,200
Canton	MI	3,000,000	Allied Waste Industries	Operational	EMCON/Organic Waste Technologies, Inc.		2,160
Montrose	MI	1,200,000	WMI	Operational	Granger Electric Company	0.8	
Birch Run	MI	1,000,000	Republic Services, Inc.	Operational	Granger Electric Company	0.8	
Lennon	MI	4,000,000	WMI	Operational	NANR	0.8	0,400
Lennon	MI	4,000,000	WMI	Operational	Bio Energy Partners	1.6	0,800
Erie	MI	1,000,000	Allied Waste Industries	Operational	GRS		1,080
Belleville	MI	1,000,000	Allied Waste Industries	Operational	EQ Energy Recovery, Inc.	0.1	
Belleville	MI	1,000,000	Allied Waste Industries	Operational	EQ Energy Recovery, Inc.	1.5	
Whitehall	MI	2,300,000	WMI	Operational	North American Natural Resources, LLC (NANR)	0.8	
Whitehall	MI	2,300,000	WMI	Operational	North American Natural Resources, LLC (NANR)	0.2	
Wayne	MI	2,944,975	WMI	Operational	Ameresco		0,160
Canton	MI	6,295,758	WMI	Operational	Ameresco		0,570
Wayne	MI	13,800,000	WMI	Operational	Ameresco		1,270
Belleville	MI	1,000,000	Allied Waste Industries	Shutdown	EQ Energy Recovery, Inc.		
Three Rivers	MI	3,000,000		Shutdown	To be sold		
Zeeland	MI	3,000,000	WMI	Candidate			
Pierson	MI	1,620,000	Allied Waste Industries	Candidate			
Midland	MI	1,700,000	City of Midland Utilities Dept.	Candidate			
Byron Center	MI	2,850,000	Kent County	Candidate			
Menominee	MI	1,500,000		Candidate			
Ravenna	MI	1,360,000	Muskegon County	Candidate			
Harrison	MI	1,260,000		Candidate			
Saginaw	MI	1,440,000	WMI	Candidate			
Smiths Creek	MI	2,880,000	St. Clair County	Candidate			
Buchanan	MI	2,794,000	Southeast Berrien County Authority	Candidate			
Pinconning	MI	1,868,900	Republic Services, Inc.	Candidate			
Frederic	MI	867,692	WMI	Potential			
Bad Axe	MI	462,000	Cove Landfill of Bad Axe Inc.	Potential			
Onaway	MI		Republic Services, Inc.	Potential			
Maple City	MI	661,743	Glen's Sanitary Landfill Inc.	Potential			
On-tonagon	MI	2,000,000	Onyx Waste Services, Inc.	Potential			
Manistee	MI	912,000		Potential			
Watervliet	MI	1,055,251	Landfill Management Co.	Potential			
Burnsville	MN	1,000,000	WMI	Operational	NEO	4.2	1,600
Elk River	MN	1,464,188	WMI	Operational	Power Strategies	3.2	1,250
Eden Prairie	MN	4,799,040	Allied Waste Industries	Operational	NEO/GRS	4.8	1,500
Inver Grove Heights	MN	18,362,736	Allied Waste Industries	Operational	GRS	12.0	6,000
Mankato	MN	1,165,580	Ponderosa of Blue Earth County	Candidate			
Brainerd	MN	337,671		Candidate			
New London	MN	1,300,000		Candidate			
	MN	1,125,000		Candidate			
Dudas	MN	1,555,267		Candidate			
Glencoe	MN	2,920,000	WMI	Candidate			
Hamel	MN	2,100,000	Minnesota Pollution Control Agency	Candidate			
	MN	1,060,161		Potential			
Anoka	MN	3,419,316	State of Minnesota	Potential			
Detroit Lakes	MN	420,000		Potential			
Sleepy Eye	MN	480,000		Potential			

Moorhead	MN			Potential		
Brainerd	MN	224,332		Potential		
Mora	MN	164,592		Potential		
	MN	3,000,000		Potential		
Hopkins	MN	1,500,000		Potential		
Lake Elmo	MN	1,540,000		Potential		
Marshall	MN	800,000		Potential		
	MN	310,640		Potential		
Buffalo	MN		Onyx Forest City Road LF	Potential		
Crookston	MN	600,000		Potential		
	MN	1,439,112		Potential		
St. Cloud	MN			Potential		
Blooming Prairie	MN	621,000		Potential		
Andover	MN	2,400,000		Potential		
Rice Lake	MN			Potential		
Leland	MS	3,500,000	Allied Waste Industries	Candidate		
McNeill	MS	562,201	Trans American Waste Central Landfill, Inc.	Candidate		
Lake	MS	1,538,740	Waste Management of Mississippi, Inc.	Candidate		
Starkville	MS	725,000	Golden Triangle Regional Solid Waste Management Authority	Candidate		
Ridgeland	MS	12,000,000	Allied Waste Industries	Candidate		
			Waste Management of Mississippi, Inc.			
Pass Christian	MS	4,009,297		Candidate		
Ovett	MS	752,000	Pine Belt Regional SW Authority	Candidate		
			Waste Management of Mississippi, Inc.			
Meridian	MS	1,754,865		Candidate		
			Waste Management of Mississippi, Inc.			
Sibley	MS	1,005,696		Candidate		
			Waste Management of Mississippi, Inc.			
Houston	MS	3,465,834		Candidate		
			Three Rivers Regional Solid Waste Management Authority			
Pontotoc	MS	1,471,872		Candidate		
Robinsonville	MS	744,600	Trash Hunters of Tunica, Inc.	Candidate		
Jackson	MS	2,032,800	City of Jackson	Potential		
Canton	MS	500,000	City of Canton	Potential		
			Trans American Waste Central Landfill, Inc.			
McNeill	MS	746,805		Potential		
Durant	MS	650,000	Transhunters of Mississippi, Inc.	Potential		
Louisville	MS	125,000	City of Louisville	Potential		
Columbus	MS	4,000,000		Potential		
Biloxi	MS	1,000,000	Allied Waste Industries	Potential		
Kiln	MS		Hancock County	Potential		
Holcomb	MS		J. Ryan Corporation	Potential		
			Southern Landfill Management, Inc.			
	MS	121,200		Potential		
De Kalb	MS	16,000	Safe Disposal, Inc.	Potential		
			Leflore County Board of Supervisors			
Greenwood	MS	368,350		Potential		
	MS		MacLand Ash Disposal Facility	Potential		
			Northeast Mississippi Solid Waste Management Authority			
Ripley	MS	96,000		Potential		
	MS			Potential		
Maryland Heights	MO	5,406,271	Fred Weber, Inc.	Operational	Fred Weber, Inc.	0.300
Maryland Heights	MO	5,406,271	Fred Weber, Inc.	Operational	Fred Weber, Inc.	1.000
Ballwin	MO	8,000,000	Onyx Oak Ridge LF	Operational	Toro Energy, Inc.	2.304
Sugar Creek	MO	3,743,617	WMI	Operational	WMI	
Hartville	MO	1,994,000	Hartville Sanitary Landfill	Candidate		
Poplar Bluff	MO	1,950,000	Allied Waste Industries	Candidate		
Sedalia	MO	1,417,442	WMI	Candidate		
Columbia	MO	1,600,000	City of Columbia	Candidate		
Jefferson City	MO	1,600,000	Allied Waste Industries Inc.	Candidate		
Joplin	MO	1,324,368	Joplin City Hall	Candidate		
Lees Summit	MO	1,423,910	City of Lee's Summit	Candidate		
Dexter	MO	1,200,000	Allied Waste Industries Inc.	Candidate		
Maryland Heights	MO	11,939,306	Allied Waste Industries	Candidate		
Joplin	MO	900,500	Newton-McDonald LF Board	Candidate		
Washington	MO	900,000	Allied Waste Industries Inc.	Candidate		
Macon	MO	833,524	Onyx Maple Hill Landfill	Candidate		
Lamar	MO	5,889,917	Allied Waste Industries	Candidate		
Warrensburg	MO	1,017,600	Allied Waste Industries	Candidate		
Kansas City	MO	8,071,606	Allied Waste (old LWS)	Candidate		
Springfield	MO	3,600,000	City of Springfield	Candidate		
Park Hills	MO	894,820	St. Francois Environmental Corporation	Candidate		
St. Joseph	MO	2,995,346	City of St. Joseph	Candidate		
Mineral Point	MO	1,091,900	Washington County	Candidate		
Bridgeton	MO	9,692,739	Allied Waste	Candidate		
	MO	98,520	3M Company	Potential		
	MO	56,987		Potential		
	MO	60,944	Atchison County Court	Potential		
Warrensburg	MO	5,965,432	Allied Waste Industries, Inc.	Potential		
	MO		City of Ava	Potential		
La Grange	MO	890,000	Allied Waste Industries, Inc.	Potential		
	MO			Potential		
	MO	324,335	City of Boonville	Potential		
	MO			Potential		
	MO	93,704	City of Cape Girardeau	Potential		

	MO	135,643	City of Centralia	Potential			
	MO	141,570	Callaway Mining Company	Potential			
	MO			Potential			
Sugar Creek	MO		Allied Waste Industries, Inc.	Potential			
	MO	107,440	Crawford County	Potential			
	MO	22,708		Potential			
	MO			Potential			
	MO	88,725	Niangua Excavating	Potential			
Clinton	MO	455,000	Allied Waste Industries Inc.	Potential			
	MO			Potential			
Mooresville	MO	430,122		Potential			
	MO	181,696	U.S. Army Directorate of Eng. & Hsg.	Potential			
	MO	75,863	City of Fredricktown	Potential			
Fulton	MO	360,725	City of Fulton	Potential			
	MO			Potential			
	MO	2,610	City of Morrison	Potential			
	MO	257,014		Potential			
	MO			Potential			
	MO	406,709	City of Hannibal	Potential			
Galt	MO	393,394	Henderson Sanitary Landfill	Potential			
	MO	132,017	Childers Construction Company	Potential			
	MO	37,482	City of Hermann	Potential			
	MO			Potential			
	MO	231,826	City of Jackson	Potential			
	MO	202,800		Potential			
Owensville	MO	500,176	Waste Management Inc.	Potential			
	MO			Potential			
	MO	8,551	U.S. Army	Potential			
Dexter	MO	1,881,554	Allied Waste Industries Inc.	Potential			
	MO	374,210		Potential			
Maryville	MO	595,867	City of Maryville Sanitary Landfill	Potential			
	MO	424,464	City of Mexico	Potential			
	MO		Midwest Method Land Management, Inc.	Potential			
St. Clair	MO	404,586	Allied Waste Industries	Potential			
	MO		Mississippi County Commission	Potential			
	MO			Potential			
Moberly	MO	446,863	City of Moberly	Potential			
	MO	190,440	Modern Sanitation Inc.	Potential			
	MO	382,910	J.Z. Disposal, Inc.	Potential			
	MO		Morgan County Courthouse	Potential			
	MO		National Refractories & Mining Corp.	Potential			
	MO	18,992	Nelanco Ltd.	Potential			
	MO	337,755	City of Nevada	Potential			
	MO	545,099	New Madrid County	Potential			
	MO	152,280		Potential			
	MO	215,398	Pemiscot County Commission	Potential			
	MO	1,000,000	Perry County	Potential			
Rolla	MO	291,908	County of Phelps	Potential			
Arnold	MO	2,000,000	Allied Waste Industries	Potential			
	MO	500,000	Renfro's Refuse Service, Inc.	Potential			
Kirksville	MO	171,483	Rye Creek Corporation	Potential			
	MO	108,015	City of Salem	Potential			
	MO	20,842	City of Savannah	Potential			
	MO		City of Sedalia	Potential			
	MO	2,560	St. Jude Industrial Park	Potential			
	MO		Sunray of Joplin, Inc.	Potential			
	MO		Sutton & Sons Refuse & Disposal	Potential			
Bowling Green	MO	106,438	SLE	Potential			
	MO	661,536		Potential			
	MO	24,788	City of Viburnum	Potential			
	MO			Potential			
	MO	20,280		Potential			
	MO	3,298	City of Washington	Potential			
	MO	176,288	Wat-Park Sanitary Landfill	Potential			
No reference available	MO	75,653	Allied Waste Industries	Potential			
	MO			Potential			
	MO	171,124		Potential			
	MO	191,779	City of West Plains	Potential			
	MO			Potential			
	MO	203,948	City of Willow Springs	Potential			
	MO	1,578,000	Deffenbaugh Industries, Inc.	Potential			
	MO	94,640		Potential			
Bozeman	MT	945,000	City of Bozeman	Candidate			
Kalispell	MT	1,000,000	Flathead County Solid Waste District	Candidate			
Great Falls	MT	6,665,766	Montana Waste Systems	Candidate			
Missoula	MT	3,900,000	BFI Waste Systems of North America	Candidate			
Billings	MT		City of Billings	Potential			
Bennington	NE	7,500,000	Waste Management of Nebraska, Inc.	Operational	Omaha Public Power District	3.2	6.658
Fremont	NE	1,106,263	Allied Waste Industries/Community Refuse Disposal	Candidate			

Jackson	NE	2,292,240	L.P. Gill	Candidate			
Lincoln	NE	3,039,054	City of Lincoln	Candidate			
Norfolk	NE		Allied Waste Industries	Candidate			
Papillion	NE	1,320,587	Sarpy County	Candidate			
Alliance	NE	677,472	City of Alliance	Potential			
Beatrice	NE	279,024	Beatrice County, D.P.P.	Potential			
David City	NE	550,794	Butler County Landfill, Incorporated	Potential			
Norfolk	NE	1,446,633	Community Refuse Disposal	Potential			
Milford	NE	250,165		Potential			
Grand Island	NE	319,848		Potential			
Hastings	NE	377,445		Potential			
Holdrege	NE	781,296	Holdrege City	Potential			
Ogallala	NE			Potential			
Kearney	NE	175,893	City of Kearney	Potential			
Kearney	NE	260,000	City of Kearney	Potential			
Geneva	NE		Nebraska Ecology Systems, Incorporated	Potential			
Clarkson	NE	183,000	N.N.S.W.C.	Potential			
North Platte	NE	407,491	City of North Platte	Potential			
Nebraska City	NE	218,822	Osceola County	Potential			
Plattsmouth	NE	777,276		Potential			
	NE	529,074	City of Scottsbluff	Potential			
York	NE	254,240	York Area Solid Waste Agency	Potential			
Apex	NV	5,211,913	Republic Services, Inc.	Candidate			
Boulder City	NV	2,319,518	Boulder Disposal Company	Candidate			
Elko	NV	1,528,208	Elko City	Candidate			
Sparks	NV	9,200,000	Waste Management, Inc.	Candidate			
Carson City	NV	5,276,832	City of Carson City	Candidate			
	NV		Douglas Disposal, Inc.	Potential			
Laughlin	NV		Republic Services, Inc.	Potential			
Pahrump	NV	25,980	Nye County	Potential			
	NV	28,200,000	Clark County Department of Public Works	Potential			
Winnemucca	NV	174,642	Humboldt County	Potential			
Keene	NH	800,000	City of Keene	Operational	Vermont Energy Recovery	1.0	
Manchester	NH	1,200,000	Dunbarton Energy Partners	Operational	U.S. Energy Biogas Corp.	0.6	
Nashua	NH	2,500,000	City of Nashua Solid Waste Department	Operational	NEO/Ziegler	2.4	1.300
Bethlehem	NH	2,000,000	Casella Waste Systems, Inc.	Operational	Commonwealth Resource Mgmt / Montauk		
Rochester	NH	8,750,000	Waste Management of NH	Operational	Bio Energy Partners	3.2	
Rochester	NH	8,750,000	Waste Management of NH	Operational	Bio Energy Partners	6.6	
Alton	NH		Town of Alton	Potential			
Bedford	NH		Town of Bedford	Potential			
Bennington	NH		Town of Bennington	Potential			
Chester	NH	30,000	Town of Chester	Potential			
Farmington	NH		Town of Farmington	Potential			
Gilmanston	NH		Town of Gilmanston	Potential			
Groton	NH		Town of Groton	Potential			
Harrisville	NH		Town of Harrisville	Potential			
Hinsdale	NH			Potential			
Kingston	NH		Town of Kingston	Potential			
Lebanon	NH		City of Lebanon	Potential			
Lisbon	NH		Town of Lisbon	Potential			
Conway	NH		Town of Conway	Potential			
Merrimack	NH		Town of Merrimack	Potential			
Success	NH		Mount Carberry I.F. LLC	Potential			
New Ipswich	NH		Town of New Ipswich	Potential			
Tamworth	NH		Town of Tamworth	Potential			
Troy	NH		Town of Troy	Potential			
Center Tuftonboro	NH		Town of Tuftonboro	Potential			
Rochester	NH	8,750,000	Waste Management of NH	Potential			
Wakefield	NH		Town of Wakefield	Potential			
North Arlington	NJ	28,206,601	Hackensack Meadowlands Development Commission	Operational	NEO	3.8	1.600
Mansfield	NJ	3,875,822	Burlington County Board of Chosen Freeholders	Operational	Burlington County, Rutgers University		0.140
Woodbine	NJ	2,000,000	Cape May County	Operational	U.S. Energy Biogas Corp.		0.600
East Brunswick	NJ	45,437,841	Middlesex County Utility Authority	Operational	NERC	6.7	
Edison	NJ	2,500,000	Edison Township	Operational	NERC	6.6	
Lafayette	NJ	1,700,000		Operational	U.S. Energy Biogas Corp.	1.2	
Edison	NJ	4,302,213	JLR	Operational	NERC	6.7	
North Arlington	NJ	20,995,801	Hackensack Meadowlands Development Commission	Operational	NEO	1.9	0.900
Sewell	NJ	7,259,985	Kinsley's	Operational		2.4	
Tinton Falls	NJ	9,381,853	County of Monmouth	Operational	GSF Energy	10.0	4.900
Manchester	NJ	16,203,683	Ocean County Landfill Corp.	Operational	LES	4.8	2.016
North Arlington	NJ	28,206,601	Hackensack Meadowlands Development Commission	Shutdown	NEO		0.600
	NJ	1,000,000	Dover Township	Shutdown	Wehran Energy Corporation	0.4	
Kearny	NJ	5,998,800	HMDC	Shutdown	GSF Energy		
Kearny	NJ	26,994,601	HMDC	Shutdown	GSF Energy		
Kearny	NJ	11,997,600	Town of Kearny	Shutdown	GSF Energy		
Mount Holly	NJ	3,290,000		Shutdown	Bio Energy Partners	3.3	
Egg Harbor Township	NJ	900,000	Atlantic County Utilities Authority	Candidate			
Deerfield	NJ	2,970,590	Cumberland County	Candidate			

Woodbury	NJ	2,889,947	Gloucester County	Candidate			
Pennsauken	NJ	3,011,986	Pollution Control Financing Auth. of Camden County	Candidate			
	NJ	1,000,000	Browning Ferris Industries	Candidate			
Galloway	NJ			Potential			
Linden	NJ	1,155,339	City of Linden	Potential			
Bordentown	NJ			Potential			
Egg Harbour	NJ	882,917	Newco Waste Systems	Potential			
Alloway	NJ	963,756	Salem County Utilities Authority	Potential			
Lafayette	NJ	858,576	Sussex County Municipal Utilities Authority	Potential			
Oxford	NJ	960,000	Pollution Control Financing Authority	Potential			
Albuquerque	NM	2,800,000	City of Albuquerque	Shutdown		0.1	
Albuquerque	NM	3,700,000	City of Albuquerque	Candidate			
Las Vegas-New Mexico	NM	1,000,000	City of Las Vegas-New Mexico	Candidate			
Albuquerque	NM	2,800,000	City of Albuquerque	Candidate			
	NM		Los Alamos County	Potential			
Albuquerque	NM	6,000,000	City of Albuquerque	Potential			
Goshen	NY	4,051,529		Operational	Ameresco	5.0	
Brookhaven	NY	8,100,000	Town of Brookhaven	Operational	Palmer Capital Corporation/Wehran Energy	6.0	1,522
Maine	NY	3,500,000	Broome County	Operational	Meridian Resources	1.6	
Staten Island	NY	108,361,626	New York City	Operational	DQE Financial / Montauk Energy Capital		14,000
Albany	NY	4,100,000	City of Albany	Operational	NEO	1.9	1,000
Fairport	NY	6,366,723	WMI	Operational	Bio Energy Partners	3.2	
Lancaster	NY	2,352,773	Gunville Energy	Operational	Innovative Energy	5.2	
Model City	NY	5,173,578	Modern Disposal Services, Inc.	Operational	Innovative Energy	5.6	2,600
Scottsville	NY	5,212,000	WMI	Operational	Bio Energy Partners	2.4	
Hempstead	NY	6,500,000	Town of Hempstead	Operational	U.S. Energy Biogas Corp.	3.0	
Stanley	NY	2,623,871	Ontario County	Operational	Innovative Energy	3.2	0,936
Goshen	NY	4,000,000		Operational	GRS	3.2	2,000
Waterloo	NY	19,000,000	IESI	Operational	Innovative Energy	11.2	6,480
Smithtown	NY	1,700,000		Operational	U.S. Energy Biogas Corp.	1.2	
Camillus	NY	1,700,000		Operational	U.S. Energy Biogas Corp.	1.2	
Troy	NY	1,500,000	City of Troy	Operational	Siemens Building Technologies, Inc.	0.8	
Maine	NY	3,500,000	Broome County	Construction	Meridian Resources	0.7	
Hauppauge	NY	8,000,000	Town of Islip	Shutdown	JWP Energy & Environment	4.0	
Huntington	NY	3,800,000		Shutdown	Wehran Energy Corporation	1.0	
Utica	NY	1,190,000		Shutdown	Bio Energy Partners	1.6	
Oyster Bay	NY	2,800,000		Shutdown	U.S. Energy Biogas Corp.	1.2	
Riverhead	NY	3,000,000	Town of Riverhead	Shutdown	United Environmental Services, Inc.		
Saratoga Springs	NY	2,129,596	City of Saratoga Springs	Shutdown	Saratoga Springs	1.0	
Auburn	NY	700,000	City of Auburn	Candidate			
Middlegrove	NY	1,406,308	Austro Brothers, Inc.	Candidate			
Sardinia	NY	4,026,200	WMI	Candidate			
Elmira	NY	1,500,000	Chemung County SWMD	Candidate			
Morrisonville	NY	1,500,000	Clinton County	Candidate			
Rodman	NY	1,500,000	Development Authority of the North Country (DANC)	Candidate			
Walton	NY	795,236	Delaware County Department of Public Works	Candidate			
Ellery	NY	3,200,000	Chautauqua County	Candidate			
Perry	NY	2,000,000	Formso, Incorporated	Candidate			
Johnstown	NY	998,486	Fulton County	Candidate			
	NY	1,200,000	Rockland County	Candidate			
Fairport	NY	2,000,000	WMI	Candidate			
Angelica	NY	500,000	Casella Waste Systems, Inc.	Candidate			
Lincoln	NY	1,260,000	Madison County	Candidate			
Bergen	NY	1,200,000	Monroe County D.E.S.	Candidate			
Niagara Falls	NY	2,633,473	Niagara County	Candidate			
North Sea	NY	1,102,714	Town of Southampton	Candidate			
Thompson	NY	1,359,000	Sullivan County	Candidate			
Tonawanda	NY	1,000,000	Allied Waste Industries	Candidate			
Cohoes	NY	3,112,000	Town of Colonie	Candidate			
Lake George	NY	403,000	Village of Lake George	Candidate			
Goshen	NY	4,051,529		Potential			
	NY	360,199	Allegany County D.P.W.	Potential			
Auburn	NY	51,895	City of Auburn	Potential			
	NY	6,000,000		Potential			
Bath	NY	549,466	Steuben County	Potential			
	NY	125,813	Oswego County	Potential			
	NY			Potential			
Root	NY	1,050,620	MOSA Solid Waste Management Authority	Potential			
Pharsalia	NY			Potential			
	NY			Potential			
	NY			Potential			
	NY	1,451,997	Town of Clifton Park	Potential			
Cohoes	NY			Potential			
McGraw	NY		Cortland County Solid Waste Department	Potential			
	NY	10,000,000	Westchester County	Potential			
	NY			Potential			
	NY			Potential			
	NY			Potential			
	NY	1,000,000	Town of East Hampton	Potential			

Fort Johnson	NY	733,505	MOSA	Potential			
	NY		NYC Department of Sanitation	Potential			
	NY			Potential			
	NY	15,000		Potential			
Staten Island	NY	108,361,626	New York City	Potential			
Holtsville	NY		Town of Brookhaven	Potential			
	NY		Essex County	Potential			
Felts Mills	NY	1,380,000	Laidlaw Waste Systems	Potential			
	NY		Town of Manlius	Potential			
	NY			Potential			
Lockport	NY	696,421	Niagara County Refuse Disposal	Potential			
Tonawanda	NY	5,998,800	Niagara Landfill, Incorporated	Potential			
Niskayuna	NY			Potential			
	NY			Potential			
	NY	1,734,043	Orleans Sanitary Landfill, Incorporated	Potential			
	NY	141,790	Town of Philinstown	Potential			
North Hempstead	NY	2,161,000	North Hempstead D.P.W.	Potential			
Queensbury	NY	1,597,197	Town of Queensbury	Potential			
	NY			Potential			
Waterloo	NY	19,000,000	IESI	Potential			
Bath	NY	615,000		Potential			
Ithaca	NY		Tompkins County	Potential			
	NY			Potential			
Naponechle	NY	52,500	Ulster County RRA	Potential			
Woodstock	NY	100,000	Ulster County RRA	Potential			
Hickory	NC	2,681,250	Catawba County	Operational	Enerdyne Power Systems, Inc.	3.0	1,584
Woodfin	NC	3,300,000	Buncombe County	Operational	Enerdyne Power Systems, Inc.		0.198
Greensboro	NC	7,616,175	City of Greensboro	Operational	Duke Solutions		
Winston-Salem	NC	5,258,571	City of Winston-Salem	Operational	DTE Biomass/Enerdyne Power Systems, Inc.	4.6	
Concord	NC	4,500,000	Allied Waste Industries	Operational	GRS	4.0	2,300
Fayetteville	NC	1,651,776	Cumberland County	Operational	DTE Biomass/Enerdyne Power Systems, Inc.		0.725
Hendersonville	NC	2,100,000	Henderson County	Operational	Enerdyne Power Systems, Inc.		0.820
Raleigh	NC	5,000,000	Wake County Solid Waste Department	Operational	DTE Biomass Energy		1,440
Greenville	NC	3,064,206	Pitt County	Operational	Enerdyne Power Systems, Inc.		1,000
Raleigh	NC	5,600,000	City of Raleigh Solid Waste Services Dept.	Operational	Natural Power/Palmer Capital Corporation		1,440
Burnsville	NC	384,804	Yancey County	Operational	Natural Power		0.029
Newland	NC	249,000	Avery County	Construction	Natural Power/Blue Ridge RC&D	0.0	0.014
Newton	NC	1,600,000	Catawba County	Shutdown	Enerdyne Power Systems, Inc.	0.8	0.300
Raleigh	NC	50,000	Rowland Landfill, Inc.	Shutdown	Natural Power	0.2	
Sparta	NC	237,899	Alleghany County	Candidate			
Crumpler	NC	438,069		Candidate			
Supply	NC	1,063,573		Candidate			
Asheville	NC	850,000	Buncombe County	Candidate			
Concord	NC	1,368,822	Cabarrus County	Candidate			
Hudson	NC	1,214,795	Caldwell County	Candidate			
Albemarle	NC	1,523,736	City of Albemarle	Candidate			
Shelby	NC	1,135,140	Cleveland County	Candidate			
New Bern	NC	1,300,000	Coastal Regional Solid Waste Management Authority	Candidate			
Holly Grove	NC	856,860	Davidson County	Candidate			
Aulander	NC	3,272,685	Republic Services, Inc.	Candidate			
Tarboro	NC	1,870,499	Edgecombe County	Candidate			
Dallas	NC	1,634,226	Gaston County	Candidate			
Dunn/Erwin	NC	1,154,018	Harnett County	Candidate			
Statesville	NC	2,280,900	Iredell County	Candidate			
Dillsboro	NC	419,054		Candidate			
Selma	NC	1,780,000	Johnston County	Candidate			
La Grange	NC	1,200,000		Candidate			
Wilmington	NC	1,500,000	New Hanover County	Candidate			
Newton	NC	1,600,000	Catawba County	Candidate			
Chapel Hill	NC	1,600,000	Orange County	Candidate			
Kernersville	NC	959,464	WMI	Candidate			
St. Pauls	NC	1,009,134		Candidate			
Madison	NC	1,357,483	Rockingham County	Candidate			
Mt. Airy	NC	1,243,631	Surry County	Candidate			
Union	NC	30,000	Republic Services, Inc.	Candidate			
Rougemont	NC	1,000,000	Republic Services	Candidate			
Fort Bragg	NC	763,800		Candidate			
Jacksonville	NC	1,299,515	U.S. Marine Corps	Candidate			
Mount Gilead	NC	3,600,000	Republic Services, Inc.	Candidate			
Boone	NC		Watauga County	Candidate			
Dudley	NC	2,416,380	Wayne County	Candidate			
Wilkesboro	NC	1,127,341	Wilkes County	Candidate			
Roaring River	NC	500,000	Wilkes County	Candidate			
Wilson	NC	2,080,436	Wilson County	Candidate			
Graham	NC	500,000	Alamance County	Potential			
Burlington	NC	2,278,000	Alamance County	Potential			
Taylorsville	NC	433,130		Potential			
Anderson Creek	NC	206,497	Harnett County	Potential			
Lilesville	NC	303,941	Anson County	Potential			
Saxapahaw	NC	159,740		Potential			

	NC	554,620	Weyerheuser County	Potential			
	NC	289,647	Bertie County	Potential			
Elizabethtown	NC	636,269	Bladen County	Potential			
Morganton	NC	649,695		Potential			
Canton	NC	889,647	Town of Canton	Potential			
Beaufort	NC	799,920	Carteret County	Potential			
Yanceyville	NC	111,731	Caswell County	Potential			
	NC	559,020	Chatham County	Potential			
Marble	NC	423,438		Potential			
Durham	NC	4,957,416	City of Durham	Potential			
	NC	52,189	Clay County	Potential			
Cliffside	NC	209,084	Rutherford County	Potential			
Whiteville	NC	1,285,344	Chambers Waste Systems	Potential			
	NC	985,028	Craven County	Potential			
	NC	251,370	Currituck County LF	Potential			
Elkin	NC	82,099	Dan Johnson	Potential			
Mocksville	NC	431,466	Davie County	Potential			
Kenansville	NC	595,645	Duplin County	Potential			
East Lake	NC	643,568	Dare County	Potential			
Feltonville	NC	2,087,201	Wake County	Potential			
Lenoir	NC	650,000	Republic Services, Inc.	Potential			
Franklin	NC	624,283	Franklin County	Potential			
Robbinsville	NC	117,713	Graham County	Potential			
Butner	NC	379,005		Potential			
Oxford	NC	808,605		Potential			
Snow Hill	NC	147,289		Potential			
Aurelian Spring	NC	856,932		Potential			
Haywood	NC	787,116	Haywood County	Potential			
Haywood	NC	89,804		Potential			
Winton	NC	305,291	Hartford County	Potential			
Highlands	NC	78,788	U.S. Forest Service Highlands District	Potential			
Raeford	NC	354,726	Hoke County	Potential			
Grover	NC		Republic Services, Inc.	Potential			
Trenton	NC	156,630	Jones County	Potential			
High Point	NC	950,000	City of High Point Department of Public Services	Potential			
Lemon Springs	NC	935,481	Lee County	Potential			
Lincolnton	NC	491,739		Potential			
Franklin	NC	46,753		Potential			
Franklin	NC	184,096	Macon County	Potential			
Marshall	NC	181,770	Madison County	Potential			
Marshall Township	NC	37,055		Potential			
	NC	735,300	McDowell County	Potential			
Charlotte	NC	3,710,642	Mecklenburg County	Potential			
Carthage	NC	1,315,238	Moore County	Potential			
Nashville	NC	1,669,484	Nash County	Potential			
Jackson	NC	363,180	Northampton County	Potential			
Jacksonville	NC	923,543		Potential			
Bayboro	NC	133,583	Pamlico County	Potential			
Elizabeth City	NC	892,763	Pasquotank County	Potential			
Buraw	NC	351,401	Pender County	Potential			
Belvidere	NC	315,619	Perquimans-Chowan Counties	Potential			
Mill Springs	NC	214,002	Polk County	Potential			
Asheboro	NC	1,815,420	Randolph County	Potential			
Rockingham	NC	523,271	Richmond County	Potential			
High Point	NC	2,330,000	City of High Point Department of Public Services	Potential			
Madison	NC	170,000	Rockingham County	Potential			
Salisbury	NC	807,044		Potential			
	NC	598,794	City of Roxboro	Potential			
Rutherfordton	NC	1,600,000	Rutherford County	Potential			
Roseboro	NC	2,500,000	Allied Waste Industries, Inc.	Potential			
Maxton	NC	788,677		Potential			
	NC	613,676	Sorrells Grading Company	Potential			
Raleigh	NC	1,786,691	Wake County	Potential			
	NC	478,368	Stokes County	Potential			
Elkin	NC	645,506	Surry County	Potential			
Mt. Airy	NC		Surry County	Potential			
Bryson City	NC	127,170	Swain County	Potential			
	NC		City of Thomasville	Potential			
Rosman	NC	628,320	NC State Division of Highways	Potential			
Rosman	NC	67,337		Potential			
Henderson	NC	970,899		Potential			
Warrenton	NC	230,029	Warren County	Potential			
Plymouth	NC	362,728	Washington County	Potential			
Cullowhee	NC	9,312	West Carolina University	Potential			
Charlotte	NC		City of Charlotte	Potential			
Fargo	ND	1,500,000	City of Fargo	Operational			0.576
Grand Forks	ND	2,600,000	City of Grand Forks	Candidate			
Waynesburg	OH	14,157,332	WMI	Operational	Toro Energy		
Columbus	OH	2,664,190		Operational	Network Energy		
Lowellville	OH	18,575,447	Allied Waste Industries	Operational	Energy Developments, Inc.	14.9	
Bellevue	OH	6,000,000	Allied Waste (old LWS)	Operational	DTE Biomass		

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Solon	OH	10,000,000	WMI	Operational	MM-Cuyahoga Energy	3.8	1,728
Solon	OH	10,000,000	WMI	Operational	EMCON/OWT		1,728
Cincinnati	OH	9,699,780	WMI	Operational	WMI		
Northwood	OH	6,000,000	WMI	Operational	Toro Energy		2,200
Northwood	OH	6,000,000	WMI	Operational	Toro Energy		1,100
Akron	OH	6,014,112	City of Akron	Operational	Granger Energy		
Painesville	OH	1,000,000	Lake County DPW	Operational	Granger Energy		
Oberlin	OH	12,529,697	Allied Waste Industries	Operational	Energy Developments, Inc.	10.8	4,500
Grove City	OH		SWACO	Operational	Energy Developments, Inc.	4.1	
Port Clinton	OH	6,032,659	Allied Waste Industries	Operational	Energy Developments, Inc.	2.7	
Dayton	OH	6,150,000	WMI	Operational	DTE Biomass		1,122
Cincinnati	OH	11,500,000	Rumpke Sanitary Landfill, Inc.	Operational	GSF / DQE Financial / Montauk Energy Capital		9,000
Canton	OH	1,000,000	WMI	Operational	Toro Energy		
Dayton	OH	7,500,000	WMI	Operational	DTE Biomass		1,550
Springfield	OH	4,720,924	Davis	Operational	Springfield Gas/Custer Services		
Nelsonville	OH	1,200,000	Kilbarger Construction, Inc.	Candidate			
New Springfield	OH	1,228,735	Waste Management, Inc.	Candidate			
Morrow	OH	10,107,852	Allied Waste Industries	Candidate			
Georgetown	OH	2,784,000	Rumpke Waste, Inc.	Candidate			
Celina	OH	684,000	Allied Waste Industries	Candidate			
Alliance	OH	1,053,492		Candidate			
East Liverpool	OH	1,037,357	East Liverpool LF, Inc.	Candidate			
Carey	OH	3,700,000	Allied Waste Industries	Candidate			
East Sparta	OH	8,238,436	Republic Services, Inc.	Candidate			
Bucyrus	OH	1,562,265	Crawford County Board of Commissioners	Candidate			
Defiance	OH	1,320,000		Candidate			
Huron	OH	2,920,000	Erie County	Candidate			
Amanda	OH	3,014,323	WMI	Candidate			
Grove City	OH	7,632,104	Solid Waste Authority of Central Ohio	Candidate			
Geneva	OH	1,927,180	WMI	Candidate			
Solon	OH	4,188,783	Allied Waste Industries	Candidate			
Findlay	OH	2,295,000	Hancock County Commissioners	Candidate			
Toledo	OH	5,416,000	City of Toledo	Candidate			
Willard	OH	1,464,341	Huron County Commissioners	Candidate			
Dover	OH	11,739,500	Kimble Clay & Limestone Company	Candidate			
Painesville	OH	3,742,878	Lake County	Candidate			
Navarre	OH	3,484,027	Norton Environmental Company	Candidate			
Shiloh	OH	1,440,000	Allied Waste Systems	Candidate			
Waverly	OH	1,438,000	Pike Sanitation Company	Candidate			
Amanda	OH	1,544,082	Republic Services, Inc.	Candidate			
Broadview Heights	OH	5,627,995	Norton Environmental Company	Candidate			
Wellston	OH	1,600,000	Rumpke	Candidate			
Bryan	OH	2,362,036	Allied Waste	Candidate			
Glenford	OH	5,088,000	Waste Management, Inc.	Candidate			
Bowling Green	OH	858,259	Wood County Solid Waste District	Candidate			
Ashland	OH	1,099,797	Ashland County Commission	Potential			
Atwater	OH	16,141,779		Potential			
	OH			Potential			
	OH			Potential			
Brooklyn	OH			Potential			
St. Marys	OH	1,142,214	City of Saint Marys Refuse Department	Potential			
	OH	235,750		Potential			
	OH	430,008		Potential			
Brooklyn Heights	OH	2,985,302		Potential			
	OH	596,840		Potential			
Fowler Township	OH	5,167,297		Potential			
Bidwell	OH		Gallia County Commissioners	Potential			
	OH			Potential			
Malinta	OH	720,000	Henry County Commissioners	Potential			
Millersburg	OH	648,641	Holmes County Commissioners	Potential			
Findlay	OH			Potential			
Twinsburg	OH			Potential			
Springfield	OH			Potential			
Middletown	OH	2,800,000	City of Middletown	Potential			
	OH			Potential			
Dayton	OH		Montgomery County Board of Commissioners	Potential			
	OH			Potential			
Eaton	OH		Preble County Commissioners	Potential			
Ottawa	OH	531,308	Putnam County Commissioners	Potential			
	OH			Potential			
	OH	104,814		Potential			
	OH	347,907		Potential			
South Salem	OH	1,332,511	Triangle Landfill, Inc.	Potential			
	OH			Potential			
Newcastle	OK	3,739,670	All-American Waste/Allied Waste Industries	Operational			
Oklahoma City	OK	10,401,150	Allied Waste Industries	Operational	DTE Biomass		3,000
	OK	3,736,131	Allied Waste Industries	Candidate			
Broken Arrow	OK	3,736,131	Allied Waste Industries	Candidate			
El Reno	OK	1,800,000	Canadian County	Candidate			

Oklahoma City	OK	6,589,455	WMI	Candidate			
Enid	OK	2,392,871	City of Enid	Candidate			
Lawton	OK	2,950,368	City of Lawton	Candidate			
Alex	OK	1,849,055	Waste Management Inc.	Candidate			
Muskogee	OK	2,093,541	Waste Management, Inc.	Candidate			
Bartlesville	OK	1,000,000	Waste Connections, Inc.	Candidate			
Shawnee	OK	1,526,225	Allied Waste Industries	Candidate			
Tulsa	OK	4,354,126	WMI	Candidate			
Ardmore	OK	1,195,397	Southern Oklahoma Regional Disposal, Inc.	Candidate			
Alex	OK	1,201,200	Waste Connections, Inc.	Candidate			
	OK	6,865,818	Allied Waste Industries	Potential			
Tecumseh	OK	162,847	Absolute Waste Solutions, Inc.	Potential			
Tahlequah	OK	877,185	Cherokee Nation of Oklahoma	Potential			
Chicasha	OK	424,500	Sanifill of Oklahoma, Inc.	Potential			
			Northwest Oklahoma Solid Waste Authority	Potential			
Woodward	OK	982,633		Potential			
	OK	800,800	Waste Connections, Inc.	Potential			
Corvallis	OR	4,500,000	Allied Waste Industries	Operational	Pacific Northwest Generating Cooperative	2.4	
McMinnville	OR	2,500,000	WMI	Operational	RECO		
Eugene	OR	3,410,000	Lane County	Operational	Emerald People's Utility District	3.2	
Portland	OR	12,000,000	Portland Metro	Operational	Palmer Capital Corporation		
Arlington	OR	20,000,000	WM Disp. Services of OR	Candidate			
Medford	OR	1,650,000		Candidate			
Boardman	OR	4,000,000	Waste Connections	Candidate			
Bonanza	OR	1,000,000	Klamath County	Candidate			
Bend	OR	700,000	Deschutes County	Candidate			
The Dalles	OR	1,600,000	Waste Connections	Candidate			
Roseburg	OR	1,200,000	Douglas County Southwest Administration	Candidate			
Milton-Freewater	OR	125,000	City of Milton-Freewater	Potential			
	OR	500,000	Sanitary Service Company	Potential			
Taylor	PA	3,000,000	Taylor Energy Partners	Operational	U.S. Energy Biogas Corp.	2.0	
Millersburg	PA	1,200,000	WMI	Operational	RECO		
Plainfield Township	PA	8,000,000	WMI	Operational	Green Knight Economic Development Corporation/Waste Management, Inc.	9.9	
Lebanon	PA	1,000,000	Greater Lebanon Refuse Authority	Operational	Lebanon Methane Recovery Inc.	1.2	
Scottdale	PA	6,000,000	Allied Waste Services	Operational	Organic Waste Technologies, Inc./TT Group		
Morrisville	PA	17,000,000	WMI	Operational	WMI/PECO Energy		
Morrisville	PA	17,000,000	WMI	Operational	WMI/PECO Energy	6.6	
Dunmore	PA	14,000,000	Keystone Landfill, Inc.	Operational	PG Energy, PEI Power	7.0	4.900
Dunmore	PA	14,000,000	Keystone Landfill, Inc.	Operational	Keystone Recovery	5.6	
Erie	PA	7,000,000	WMI	Operational	Bio Energy Partners	6.1	
Montgomery	PA	5,000,000	Lycoming County Solid Waste Authority	Operational	Lycoming County	1.3	
York	PA	15,000,000	Republic Services, Inc.	Operational	Bio Energy Partners	9.1	
Greencastle	PA	3,000,000	WMI	Operational	INGENCO	1.2	2.880
Burlington	PA	1,197,066	Northern Tier SW Authority	Operational	NTSWA & PPL	0.8	0.504
Kersey	PA	12,000,000	Onyx Greentree LF	Operational	Enerdyne		
Pottstown	PA	11,000,000	WMI	Operational	Bio Energy Partners	6.6	3.000
Davidsville	PA	2,644,903	WMI	Operational	Organic Waste Technologies, Inc.		0.300
Tullytown	PA	12,500,000	WMI	Operational	WMI/PECO Energy	11.0	
Irwin	PA	6,000,000	WMI	Operational	Beacon Generating/Magellan EnviroGas Partners LLC		
Lock Haven	PA	1,500,000	Clinton County Solid Waste Authority	Operational	Wayne Township LF/Jersey Shore Steel		0.864
Zelienople	PA	500,000	J.J. Brunner	Construction	EMCON/OWT		
Monroeville	PA	2,000,000	WMI	Construction	Beacon Generating/Magellan EnviroGas Partners LLC		
Taylor	PA	14,000,000	WMI	Shutdown	Dominion Energy, Inc.		
Swedeland	PA	3,600,000	U.S. Gypsum/Montgomery County	Shutdown	NEO	2.0	0.000
East Pennsboro	PA	400,000	East Pennsboro Township	Shutdown	East Pennsboro Township	0.1	
Imperial	PA	3,000,000	Mazzaro Coal Company	Shutdown	NEO	0.5	0.000
Birdsboro	PA	5,100,000	J.P. Mascaro & Sons	Shutdown	O'Brien Energy Products, Inc. / Resource Management International	0.3	
Washington	PA	2,347,954	WMI	Candidate			
Bethlehem	PA	3,115,000	IESI, Inc.	Candidate			
Chambersburg	PA	5,500,000	IESI, Inc.	Candidate			
Easton	PA	5,000,000	Chrin Brothers, Inc.	Candidate			
Hegins	PA	2,500,000	DeNaples Brothers	Candidate			
Morgantown	PA	10,800,000	New Morgan Landfill Company (Allied Waste Industries)	Candidate			
Clarion	PA	929,812	Allied Waste Industries	Candidate			
Newburg	PA	4,000,000	WSI International	Candidate			
Crestwell	PA	3,300,000	Lancaster County Solid Waste Management Authority	Candidate			
Imperial	PA	8,400,000	Allied Waste Industries	Candidate			
Elizabeth	PA	7,187,741	WMI	Candidate			
Caernarvon Township	PA	10,003,394	Chester County Solid Waste Authority	Candidate			
West Sunbury	PA	2,000,000	WMI	Candidate			
McClellandtown	PA	570,000	Onyx CBF, Inc. LF	Candidate			
Pine Grove	PA	2,000,000	WMI	Candidate			
Birdsboro	PA	5,100,000	J.P. Mascaro & Sons	Candidate			
Media	PA	5,500,000	Delaware County	Candidate			
West Grove	PA	1,000,000	Southeastern Chester County Refuse Authority	Candidate			
Evans City	PA	3,000,001	Vogel Disposal Service Incorporated	Candidate			

Cairnbrook	PA	3,500,000	WMI	Candidate			
Libravv	PA	1,500,000	WMI	Candidate			
Mars	PA		Vogel Disposal Services Incorporated	Candidate			
Greensburg	PA	1,000,000	Westmoreland County	Candidate			
Hanover	PA	22,078,368	Adams Sanitation, Inc.	Potential			
	PA		Center Township	Potential			
Manor Township	PA			Potential			
Drumore	PA	352		Potential			
Coral	PA	350,225		Potential			
Ford City	PA			Potential			
New Bloomfield	PA	5,907	Fred D. Thebes	Potential			
Chambersburg	PA			Potential			
Lemont Furnace	PA		A.M. Kennedy, Inc.	Potential			
Hanover	PA		Kenneth Noel	Potential			
Kittanning Borough	PA			Potential			
Hollsopple	PA			Potential			
Johnstown	PA	683,559	Chambers Development (WMI)	Potential			
Slickville	PA			Potential			
Liberty	PA		M.C. Aroni Company	Potential			
Lemont Furnace	PA			Potential			
Greensburg	PA		Westmoreland Waste Company	Potential			
Hutchins	PA	590,891	McKean County Solid Waste Authority	Potential			
Lewistown	PA	400,677	Mifflin County Solid Waste Authority	Potential			
Charleroi	PA			Potential			
Graysville	PA		Morris Township	Potential			
Somerset	PA		WSI International	Potential			
Marion Center	PA			Potential			
Beach Lake	PA	1,000,000	Sexton	Potential			
Hopewell	PA	311,509	WSI International	Potential			
Greensburg	PA	398,041	Municipal Authority Westmoreland County	Potential			
Blossburg	PA			Potential			
West Newton	PA			Potential			
Parker	PA			Potential			
Birdsboro	PA	943,550	Western Berks Refuse Authority	Potential			
York	PA	754,137	York County Solid Waste Authority	Potential			
Cagsey	PR	1,000,000	Commonwealth of Puerto Rico	Candidate			
Fajardo	PR	1,000,000	Commonwealth of Puerto Rico	Candidate			
Guaynabo	PR	1,100,000	Commonwealth of Puerto Rico	Candidate			
Majaguez	PR	1,000,000	Commonwealth of Puerto Rico	Candidate			
San Juan	PR	5,000,000	Commonwealth of Puerto Rico	Candidate			
Toa Baja	PR	1,000,000	Commonwealth of Puerto Rico	Candidate			
Johnston	RI	12,000,000	Rhode Island Resource Recovery Corporation	Operational	Palmer Capital Corporation	2.0	
Johnston	RI	12,000,000	Rhode Island Resource Recovery Corporation	Operational	Palmer Capital Corporation	12.0	
Cranston	RI	1,000,000	WMI	Operational	Highland Power Co.	0.5	
Johnston	RI	12,000,000	Rhode Island Resource Recovery Corporation	Construction	Rhode Island Resource Recovery Corp.	2.5	
Bristol	RI	400,000		Potential			
Tiverton	RI			Potential			
Westerly	RI			Potential			
Conway	SC	3,037,000	Horry County Solid Waste Authority, Inc.	Operational	SCS Engineers/Enerdyne/Santee Cooper	3.0	
Wellford	SC	2,700,000	WMI	Operational	Ameresco	4.8	5.760
Bishopville	SC	3,674,775	Lee County L.E. LLC	Construction	Ameresco/Santee Cooper	5.4	
West Columbia	SC	2,485,571	Lexington County	Shutdown			
Columbia	SC	1,000,000	Greenville County	Shutdown	Cargan Resources/Alterstone		
Belton	SC	1,260,000	Allied Waste Industries	Candidate			
Johns Island	SC	2,300,000	Charleston County	Candidate			
Moncks Corner	SC	1,215,200	Berkeley County Water and Sanitation Authority	Candidate			
Moncks Corner	SC	467,700	Berkeley County Water and Sanitation Authority	Candidate			
Spartanburg	SC	1,100,000	County of Spartanburg	Candidate			
Gaffney	SC	2,400,000	Allied Waste Industries, Inc.	Candidate			
Greer	SC		Greenville County	Candidate			
Greer	SC	2,800,000	Greenville County	Candidate			
Georgetown	SC	971,000	Georgetown County Public Works Dept.	Candidate			
Greenwood	SC	710,899	Greenwood County Public Works	Candidate			
Greenwood	SC	752,734	Greenwood County Public Works	Candidate			
Ridgeland	SC	2,867,263	Waste Management, Inc.	Candidate			
Camden	SC	1,048,021	Kershaw County	Candidate			
Eastover	SC	1,987,890	Allied Waste Industries	Candidate			
Dorchester	SC	3,183,595	WMI	Candidate			
Orangeburg	SC	1,093,022	Orangeburg County	Candidate			
Pendleton	SC	600,000	Anderson County	Candidate			
Elgin	SC	15,250,000	WMI	Candidate			
Seneca	SC	749,266	Oconee County	Candidate			
Starr	SC	1,120,000	Anderson County	Candidate			
Jackson	SC	900,000	Three Rivers Solid Waste Authority	Candidate			
Spartanburg	SC	1,824,500	County of Spartanburg	Candidate			
York	SC	2,955,847	York County	Candidate			

Abbeville	SC		Abbeville County	Potential			
Abbeville	SC		Abbeville County	Potential			
Langley	SC	1,273,000	Aiken County	Potential			
Aiken	SC		Aiken County	Potential			
Allendale	SC		WasteMasters of South Carolina, Inc.	Potential			
Bamberg	SC		Bamberg County	Potential			
Barnwell	SC		Barnwell County	Potential			
Belton	SC	1,802,000	Anderson County	Potential			
Chester	SC	615,600	Chester County	Potential			
Manning	SC		Clarendon County	Potential			
Pickens	SC		Pickens County	Potential			
Florence	SC	1,114,000	Florence County	Potential			
Fort Jackson	SC		Richland County	Potential			
Georgetown	SC	208,565	Georgetown County	Potential			
Greenville	SC		City of Greenville	Potential			
Lancaster	SC		Lancaster County Public Works Department	Potential			
Laurens	SC		Laurens County Public Works Department	Potential			
Lexington	SC	500,000	Lexington County	Potential			
North Charleston	SC		Republic Services, Inc.	Potential			
Columbia	SC	1,626,000	Richland County	Potential			
	SC	310,908	Waste Management of SC, Inc.	Potential			
Sumter	SC	786,000	Sumter County	Potential			
Union	SC	530,000	Union County	Potential			
Aiken	SC		Westinghouse Savannah River Co.	Potential			
Kingsree	SC		Williamsburg County	Potential			
Sioux Falls	SD	3,410,463	City of Sioux Falls	Candidate			
Rapid City	SD	3,000,000	City of Rapid City	Candidate			
Brookings	SD	322,471		Potential			
Aberdeen	SD			Potential			
Mitchell	SD	746,864	City of Mitchell	Potential			
Pierre	SD	2,000		Potential			
Pukwana	SD	100,000	Tri-County Landfill Association	Potential			
Vermillion	SD	250,000	City of Vermillion	Potential			
Watertown	SD	353,235	City of Watertown	Potential			
Nashville	TN	3,850,150	Metropolitan Gov. of Nashville Department of Public Works	Operational	NEO	1.3	0.936
Murfreesboro	TN	4,800,000	Allied Waste Industries	Operational	Energy Developments, Inc.	2.7	1.380
Raus	TN	1,820,801	WMI	Operational	RECO		0.135
Hieskell	TN	5,031,388	WMI	Shutdown	Bio Energy Partners	3.2	1.250
Ooltewah	TN	7,300,000	City of Chattanooga	Shutdown	City of Chattanooga		
Alcoa	TN	2,027,267	City of Alcoa, City of Maryville, Blount County	Candidate			
Millington	TN	5,500,000	Allied Waste Industries	Candidate			
Memphis	TN	13,303,800	Allied Waste Industries	Candidate			
Knoxville	TN	1,670,000	Allied Waste Industries	Candidate			
Clarksville	TN	1,000,000	Bi-County Solid Waste System	Candidate			
McDonald	TN	1,595,853	Bradley County	Candidate			
Church Hill	TN	3,400,000	Allied Waste Industries	Candidate			
Lewisburg	TN	2,500,000	Waste Management	Candidate			
Johnson City	TN	2,358,439	Johnson City Solid Waste	Candidate			
Dandridge	TN	1,065,797	Jefferson County	Candidate			
Union City	TN	2,000,000	Republic Services, Inc.	Candidate			
Millington	TN	2,300,000	Shelby County	Candidate			
Ooltewah	TN	7,300,000	City of Chattanooga	Candidate			
Camden	TN	1,200,000	Waste Management, Inc.	Candidate			
Franklin	TN	1,454,272	Williamson County	Candidate			
	TN		Allied Waste Industries	Potential			
	TN		Patterson Waste Control	Potential			
	TN	154,440		Potential			
Bolivar	TN		City of Bolivar	Potential			
Bolivar	TN	390,301	Bolivar City/Hardeman County	Potential			
	TN		City of Brownsville	Potential			
Blountville	TN	1,144,000	Waste Resources of Tennessee	Potential			
Elizabethton	TN	468,565	Carter County	Potential			
	TN	81,510	Carthage City	Potential			
	TN	42,900		Potential			
Ashland City	TN	262,518	Cheatam County	Potential			
Henderson	TN	187,603	City of Hendersonville	Potential			
Harrison	TN	800,000	City of Chattanooga	Potential			
New Tazewell	TN	441,316	Claiborne County	Potential			
	TN	96,525	Clay County	Potential			
	TN		Town of Clinton	Potential			
Newport	TN	379,459	Cocke County	Potential			
	TN		Coffee County	Potential			
	TN	302,280	City of Crossville	Potential			
Crossville	TN	470,845	Cumberland County	Potential			
	TN		Gentry, Tipton, Kizer & Little PC	Potential			
	TN		D&S Land Company	Potential			
Dayton	TN		Rhea County	Potential			
Smithville	TN	302,668	De Kalb County	Potential			
Decaturville	TN	78,101	Decatur County	Potential			
	TN		Decatur County	Potential			

Dickson	TN	724,581	Dickson County	Potential			
	TN		City of Dyer	Potential			
Dyersburg	TN	396,621	City of Dyersburg	Potential			
	TN			Potential			
	TN			Potential			
Somerville	TN	171,514	Fayette County	Potential			
	TN		Fayette County Public Works	Potential			
	TN	176,450	Fayetteville/Lincoln County	Potential			
Jamestown	TN	127,893	Fentress County	Potential			
Fort Campbell	TN	1,000,000	U.S. Army	Potential			
Winchester	TN	410,018	Franklin County	Potential			
Sevier	TN	718,146	City of Gatlinburg	Potential			
Rutledge	TN	245,197	Grainger County	Potential			
Obion	TN	89,356		Potential			
Greeneville	TN	872,958	Greenville County & Town	Potential			
Greenfield	TN	55,034	Greenfield City	Potential			
	TN	96,732	Grundy County	Potential			
	TN	269,749	Hamblen County	Potential			
Chattanooga	TN	1,361,735	Hamilton County	Potential			
	TN	13,801	Hancock County	Potential			
	TN	88,458	Hartsville City/Trousdale County	Potential			
Brownsville	TN	400,926	Haywood County Solid Waste	Potential			
	TN	69,529	Henderson County	Potential			
Centerville	TN	354,497	Hickman County	Potential			
Hohenwald	TN	156,283	City of Hohenwald	Potential			
Erin	TN	69,002	Houston County	Potential			
	TN	137,966	City of Humboldt	Potential			
Waverly	TN	351,072	Humphrey's County	Potential			
	TN	464,424	Town of Huntington	Potential			
Jackson	TN	1,700,000	City of Jackson/Madison County	Potential			
Dyersburg	TN	755,275		Potential			
	TN		Commercial Waste Disposal System	Potential			
	TN			Potential			
	TN	1,500,000	City of Johnson City	Potential			
	TN	152,241	Johnson County	Potential			
	TN	173,888	City of Lafayette	Potential			
	TN	592,844	Lauderdale County	Potential			
Lawrenceburg	TN	822,071	Lawrenceburg County	Potential			
Lowland	TN		Tidi Waste Systems	Potential			
	TN	79,966	City of Lewisburg	Potential			
Lexington	TN	288,555	City of Lexington	Potential			
			Loudon County Solid Waste Commission	Potential			
Loudon	TN	903,195		Potential			
Jasper	TN	623,292	Marion County	Potential			
	TN	528,608	City of Martin	Potential			
Columbia	TN	845,293	Maury County	Potential			
McKenzie	TN	282,284	City of McKenzie	Potential			
Athens	TN	231,996	McMinn County	Potential			
	TN	311,397	City of McMinnville	Potential			
Selmer	TN	700,637	McNairy County	Potential			
Success	TN	9,387	U.S. Army - Fort Johnson	Potential			
Milan	TN	198,457	City of Milan	Potential			
Athens	TN	340,814	Waste Connections	Potential			
	TN	200,000	Monroe County	Potential			
	TN	234,569	Morgan County	Potential			
Morristown	TN	799,699	Morristown Solid Waste Authority	Potential			
Newbern	TN	49,335	Town of Newbern	Potential			
Rogersville	TN	336,369		Potential			
	TN	180,219		Potential			
Paris	TN	292,337	City of Paris	Potential			
	TN	194,088	Perry County	Potential			
Byrdstown	TN	60,435	Pickett County	Potential			
Pulaski	TN	285,586	City of Pulaski	Potential			
Cookeville	TN	1,256,907	Putnam County	Potential			
	TN	145,159	City of Red Boil Springs	Potential			
	TN		Tellico Area Disposal	Potential			
Davton	TN	616,688	Rhea County	Potential			
Harriman	TN	1,600,000	Roane County Solid Waste Authority	Potential			
Springfield	TN	542,882	Robertson County	Potential			
Murfreesboro	TN	886,600	Rutherford County	Potential			
	TN	882,395	Savannah City/Hardin County	Potential			
Helenwood	TN	340,028	Scott County	Potential			
Dunlap	TN	338,738	Sequatch/Bledsoe City	Potential			
Sevierville	TN	253,402	Sevier Solid Waste Authority	Potential			
Carthage	TN	132,992	Smith County	Potential			
	TN	42,358	City of South Fulton	Potential			
Blountville	TN	2,000,000	Baker Construction	Potential			
Brighton	TN	782,737	Tipton County	Potential			
Dresden	TN	59,837	Sharon, Dresden	Potential			
Luttrell	TN	384,572	Union County	Potential			
Moss	TN	133,987		Potential			
	TN		UT Knoxville	Potential			

	TN	186,025	Van Buren County	Potential			
	TN		Barker Brothers Incorporated	Potential			
	TN			Potential			
Sparta	TN	214,429	White County	Potential			
Lebanon	TN	594,158	Wilson County	Potential			
Arlington	TX	13,981,144	City of Arlington	Operational	Renovar Energy Corp.	9.0	2,160
Austin	TX	8,080,500	Waste Management of Texas	Operational	Toro Energy		
Houston	TX		WMI	Operational	Reliant Energy	2.0	
Garland	TX	3,858,050	City of Garland	Operational	U.S. Energy Biogas Corp.		
Brownwood	TX	1,000,000	City of Brownwood	Operational	Fenn Tech		
Conroe	TX	3,146,000	City of Conroe	Operational	Reliant Energy	2.0	
Alvin	TX	2,689,200	WMI	Operational	Reliant Energy	6.8	
San Antonio	TX	3,728,209	WMI	Operational	Energy Developments, Inc.	7.0	
Dallas	TX	15,000,000	WMI	Operational	Bio Energy Partners	6.6	
Houston	TX	28,918,718	Allied Waste Industries	Operational	DQE Financial / Montauk Energy Capital		
Dallas	TX	20,000,000	City of Dallas	Operational	Pacific Natural Energy LLC		
Rosenberg	TX	1,000,000	Fort Bend County	Operational	Landfill Gas Production		1,000
Barttown	TX	5,142,461	Waste Management of TX	Operational	Reliant Energy	6.0	1,730
Securty	TX	572,000	WMI	Operational	Reliant Energy	3.4	
Austin	TX	9,600,000	Allied Waste Industries	Operational	Gas Recovery Systems	3.0	1,500
San Antonio	TX	9,337,518	Allied Waste Industries	Operational	Energy Developments, Inc.	6.8	2,900
Humble	TX	6,049,524	Waste Management of Texas, Incorporated	Operational	Reliant Energy	11.7	
Waco	TX	2,000,000	City of Waco	Construction	CROMECCO	2.0	1,500
Corpus Christi	TX	7,830,840	City of Corpus Christi	Construction	Resource Technology Corporation		
Abilene	TX	7,020,000	Allied Waste Industries	Candidate			
Amarillo	TX	6,132,600	City of Amarillo	Candidate			
Beaumont	TX	1,123,882	Allied Waste Industries	Candidate			
Houston	TX	1,910,676	Allied Waste Industries	Candidate			
Angleton	TX	3,919,000	Republic Services, Inc.	Candidate			
Brian & College Station	TX	1,980,000	Brazos Valley SWMA	Candidate			
Linn	TX	3,844,000	Republic Waste Services of Texas	Candidate			
Austin	TX	4,734,096	City Of Austin	Candidate			
Beaumont	TX	2,251,245	City of Beaumont	Candidate			
Hebron	TX	4,381,844	City of Farmers Branch	Candidate			
Fort Worth	TX	4,996,994	City of Fort Worth	Candidate			
Grand Prairie	TX	2,098,584	City of Grand Prairie	Candidate			
Irving	TX	1,800,000	City of Irving	Candidate			
Laredo	TX	2,273,083	City of Laredo	Candidate			
Lubbock	TX	1,629,232	City of Lubbock	Candidate			
McKinney	TX	1,626,644	City of McKinney	Candidate			
Midland	TX	2,620,672	City of Midland	Candidate			
Nacogdoches	TX	1,064,534	City of Nacogdoches	Candidate			
Pampa	TX	1,157,820	City of Pampa	Candidate			
Perrvton	TX	1,606,966	City of Perrvton	Candidate			
Port Arthur	TX	1,516,481	City of Port Arthur	Candidate			
Sweetwater	TX	1,262,948	City of Sweetwater	Candidate			
Temple	TX		City of Temple	Candidate			
Bloomington	TX	2,280,369	City of Victoria	Candidate			
Weatherford	TX	1,016,400	IESI	Candidate			
Wichita Falls	TX	3,495,847	City of Wichita Falls	Candidate			
Clint	TX	4,185,278	City of El Paso	Candidate			
Colorado City	TX	1,528,956	City of Colorado	Candidate			
New Braunfels	TX	2,000,000	WMI	Candidate			
Avalon	TX	4,254,250	Republic Services, Inc.	Candidate			
Denton	TX	2,266,664	City of Denton	Candidate			
Fort Worth	TX		WMI	Candidate			
Galveston	TX	6,618,319	Allied Waste Industries	Candidate			
Tyler	TX	2,247,294	Allied Waste Industries	Candidate			
Sherman	TX	2,008,657	WMI	Candidate			
Waco	TX	936,358	WMI	Candidate			
El Paso	TX	3,653,342	City of El Paso	Candidate			
Fort Worth	TX	2,303,364	Allied	Candidate			
San Antonio	TX	3,000,000	City of San Antonio	Candidate			
Plano	TX	6,064,067	North Texas Municipal Water District	Candidate			
Kingston	TX		WMI	Candidate			
Lonaview	TX	11,250,000	Allied Waste Industries	Candidate			
Jacksonville	TX	720,000	Allied Waste Industries	Candidate			
Ferris	TX	28,200,000	WMI	Candidate			
Canyon	TX	2,760,000	Allied Waste Industries	Candidate			
Houston	TX	1,664,372	Sprint Fort Bend County	Candidate			
Sugarland	TX	1,716,000	Laidlaw Waste Systems Inc.	Candidate			
Austin	TX	2,043,245	Texas Disposal Systems	Candidate			
Altair	TX	1,841,556	Safety Clean (formerly Laidlaw)	Candidate			
Mesquite	TX	4,692,025	Allied Waste Industries, Inc.	Candidate			
Alvarado	TX	2,324,631	Allied Waste Industries	Candidate			
Fort Worth	TX	1,823,250	Waste Management Inc	Candidate			
Aledo	TX		Waste Management of Texas, Incorporated	Candidate			
Houston	TX	6,005,991	Allied Waste Industries	Candidate			
Hutto	TX	1,516,800	WMI	Candidate			
Belton	TX	343,200	Bell County	Potential			
Wichita Falls	TX		Bell Processing Inc	Potential			

Pattison	TX		Waste Management, Inc.	Potential			
Abilene	TX	745,888	Pine Street Salvage Company	Potential			
Lubbock	TX	200,200	Allied Waste Industries	Potential			
Odessa	TX		Republic Services, Inc.	Potential			
Elroy	TX			Potential			
Cleburne	TX	1,512,000		Potential			
Corsicana	TX	343,200		Potential			
Richardson	TX	825,218	City of Richardson	Potential			
Palmer	TX	892,320	WMI	Potential			
Kingston	TX	1,479,894		Potential			
	TX			Potential			
Hutchins	TX	1,000,000	Allied Waste Industries	Potential			
Altair	TX	9,009,000	Clean Harbors	Potential			
Wilmer	TX	686,400	Laidlaw Waste Systems (Dallas)	Potential			
Campbell	TX		Republic Services, Inc.	Potential			
League City	TX		Republic Services, Inc.	Potential			
Elroy	TX		T.C. Steiner	Potential			
San Angelo	TX		Republic Services, Inc.	Potential			
Cut-N-Shoot	TX	2,005,452		Potential			
Alvin	TX	858,000	Waste Management of Texas, Incorporated	Potential			
Layton	UT	3,300,000	Wasatch Energy Systems	Construction	Exelon Services Federal Group	1.0	0.648
Salt Lake	UT	11,000,000	Salt Lake Valley Solid Waste Management Council	Construction	DTE Energy		1.008
West Bountiful	UT	2,171,531	City of Bountiful	Candidate			
Logan	UT	1,400,000	City of Logan	Candidate			
Utah	UT	1,100,000	South Utah Valley Solid Waste District	Candidate			
South Jordan	UT	5,600,000	Trans-Jordan Cities	Candidate			
Vernal	UT	2,773,000	Uintah County and Vernal City	Candidate			
Washington	UT	1,292,000	Washington County	Candidate			
Beaver	UT	107,648	Beaver City	Potential			
Blanding	UT	50,780	Blanding City	Potential			
Brigham	UT	693,000	Box Elder County	Potential			
	UT			Potential			
Price	UT	280,000	Carbon County	Potential			
Monticello	UT	36,000	Max Dalton	Potential			
Duchesne	UT		Duchesne & Wasatch Counties	Potential			
East Carbon	UT			Potential			
Castledale	UT	212,184	Emery County	Potential			
	UT			Potential			
Garfield	UT	16,200	Garfield County	Potential			
Garfield	UT	8,852	Garfield County	Potential			
Garrison	UT	660	Millard County	Potential			
Moab	UT	76,300	Grand County	Potential			
Green River	UT	79,205	Green River City	Potential			
Grouse Creek	UT	1,760	Box Elder County	Potential			
Cedar	UT	44,962	Iron County	Potential			
Glendale	UT	6,560	Western Kane County SSD# 1	Potential			
Kanab	UT	39,680	Western Kane County SSD# 1	Potential			
Manila	UT	32,120	Manila City/Daggett County	Potential			
Delta	UT	67,650	Millard County	Potential			
Nephi	UT	18,300	Nephi City	Potential			
	UT	1,707,965		Potential			
Park Valley	UT	2,400	Box Elder County	Potential			
	UT			Potential			
Payson	UT	616,029	Payson City Corporation	Potential			
Provo	UT	1,131,000	City of Provo	Potential			
Laketown	UT	40,688	Rich County	Potential			
Bluff	UT	1,600	San Juan County	Potential			
Halls Crossing	UT	9,464	San Juan County	Potential			
Mexican Hat	UT	1,600	San Juan County	Potential			
Spring City	UT	108,396	Sanpete SLF Coop	Potential			
Santaquin	UT	262,080	Santaquin City	Potential			
Glenwood	UT	70,200	Sevier County	Potential			
Snowville	UT	8,100	Snowville Town	Potential			
Wanship	UT	358,896	Summit County	Potential			
Tooele	UT		Commander Tead	Potential			
Tooele	UT		Commander Tead	Potential			
Tremonton	UT	184,600	Box Elder County	Potential			
Ordan	UT	3,500,000	Weber County	Potential			
Wendover	UT	118,260	Wendover City	Potential			
Yost	UT	800	Box Elder County	Potential			
Brattleboro	VT	1,000,000	Vermont Energy Recovery	Operational	Total Energy Technologies/VER	0.5	
Burlington	VT	863,000	Biomass Energy Partners	Operational	U.S. Energy Biogas Corp.	0.7	
Newport	VT	3,600,000	New England Waste Services of Vermont (Casella Waste)	Candidate			
Moretown	VT	1,950,000		Candidate			
Morrisville	VT		Lamoille Regional SWMD	Potential			
Bristol	VT		Town of Bristol	Potential			
Randolph	VT	190,000	Town of Randolph	Potential			
Salisbury	VT		Town of Salisbury	Potential			
Shaftsbury	VT		Town of Shaftsbury	Potential			
Christiansied	VI	2,200,000		Candidate			
St. Thomas	VI	1,500,000		Candidate			

Amelia	VA	3,800,000	Waste Management of Virginia, Inc.	Operational	INGENCO	4.2	1,800
Sussex	VA	6,000,000	WMI	Operational	Enerdyne		
Hampton	VA	4,400,000	WMI	Operational	RECO		2,200
Charles City	VA	6,406,400	WMI	Operational	RECO		0,000
Winchester	VA	3,160,300	Frederick County	Operational	Frederick County		
Fairfax	VA	18,072,000	Fairfax County	Operational	LES		1,728
Fairfax	VA	18,072,000	Fairfax County	Operational	LES	3.2	1,728
Fairfax	VA	18,072,000	Fairfax County	Operational	LES	3.2	1,728
Prince William	VA	5,000,000	Prince William County	Operational	NEO	1.9	0,800
Lynchburg	VA	2,500,000	Lynchburg City	Operational	Enerdyne		
Richmond	VA	1,000,000	Allied Waste Industries	Operational	GRS	2.7	1,500
Suffolk	VA	8,344,022	SPSA	Operational	U.S. Energy Biogas Corp.		1,000
Suffolk	VA	8,344,022	SPSA	Operational	U.S. Energy Biogas Corp.	3.2	1,300
Virginia Beach	VA	7,500,000	Virginia Beach	Operational	INGENCO	3.6	0,770
Charles City	VA	6,406,400	WMI	Construction	INGENCO	3.0	3,168
Warrenton	VA	2,500,000	Fauquier County	Construction	Commonwealth Green Energy	1.0	
Chester	VA	7,000,000	Shoosmith Brothers	Construction	INGENCO	4.8	
Chester	VA	1,250,000	Chesterfield County	Shutdown		0.4	
Dinwiddie	VA	450,000	Dinwiddie County	Shutdown		0.8	
Suffolk	VA	8,344,022	SPSA	Shutdown	U.S. Energy Biogas Corp.		0,140
Charlottesville	VA	2,519,088		Candidate			
Midlothian	VA	858,000	Chesterfield County	Candidate			
Danville	VA	2,745,600		Candidate			
King George	VA	4,100,000	King George County	Candidate			
Leesburg	VA	1,487,200	Loudon County	Candidate			
Martinsville	VA	2,323,750	City of Martinsville	Candidate			
Glenns	VA	1,500,000	WMI	Candidate			
Newport News	VA	1,800,000		Candidate			
Northampton	VA		Northampton County	Candidate			
Glen Allen	VA	2,400,000	Henrico County	Candidate			
Petersburg	VA	1,144,000	City of Petersburg	Candidate			
Harrisonburg	VA	1,200,000	Rockingham County	Candidate			
Roanoke	VA	5,262,400	Roanoke Valley RA	Candidate			
Tazewell	VA	873,841	Tazewell County Board of Supervisors	Candidate			
	VA			Potential			
	VA	92,664		Potential			
Bedford	VA	14,300		Potential			
Rocky Mount	VA	38,610		Potential			
	VA			Potential			
Standardsville	VA	1,000,000	Greene County	Potential			
Lynchburg	VA		Campbell County	Potential			
Madison	VA			Potential			
Christianburg	VA		Montgomery County	Potential			
Dublin	VA		Pulaski County	Potential			
Newport News	VA	1,250,000	City of Newport News	Potential			
Richmond	VA			Potential			
	VA	429,000		Potential			
Roanoke	VA		Roanoke Valley Resource Authority	Potential			
	VA	715,000		Potential			
	VA			Potential			
	VA		Loudon County	Potential			
	WA	1,769,760	Lewis County Public Services	Operational			
Puyallup	WA	17,425,280	Land Recovery, Inc.	Operational		1.9	
Spokane	WA	4,000,000	City of Spokane	Operational	NEO	0.9	0,400
Port Orchard	WA	7,004,248	Kitsap County	Operational	OWT		
Roosevelt	WA	13,000,000	Allied Waste Systems	Operational	Klickitat County Public Utility District No. 1	2.1	
Roosevelt	WA	13,000,000	Allied Waste Systems	Operational	Klickitat County Public Utility District No. 1	8.4	4,600
Tacoma	WA	5,610,756	City of Tacoma, Refuse Utility Division	Operational	NEO	1.9	0,800
Longview	WA	950,000		Construction			
Kelso	WA	1,984,762	Cowlitz County	Construction			0,648
Snohomish	WA	3,258,989	Snohomish County Solid Waste Division	Candidate			
Maple Valley	WA	24,135,629	King County	Candidate			
Yakima	WA	1,198,976	Yakima County Solid Waste Division	Candidate			
	WA	2,008,248	WMI	Candidate			
Olympia	WA	750,000	Thurston County	Candidate			
Kent	WA	8,000,000	City of Seattle	Candidate			
Walla Walla	WA	1,102,317	City of Walla Walla	Candidate			
Yakima	WA	3,727,219	Yakima County	Candidate			
Aberdeen	WA	430,714		Potential			
Clarkston	WA	848,342	Asotin County	Potential			
Ritzville	WA	220,609		Potential			
	WA			Potential			
	WA	250,000		Potential			
Grand Coulee	WA			Potential			
	WA			Potential			
Ephrata	WA			Potential			
Fort Lewis	WA			Potential			
	WA			Potential			
	WA	599,880		Potential			

	WA	413,697		Potential			
Holden Village	WA			Potential			
	WA			Potential			
Vancouver	WA	5,000,000	Disposal Group, Inc.	Potential			
	WA			Potential			
	WA	655,454		Potential			
	WA			Potential			
	WA			Potential			
	WA			Potential			
Okanogan	WA	700		Potential			
Okanogan	WA			Potential			
	WA			Potential			
Pasco	WA			Potential			
	WA			Potential			
Port Angeles	WA			Potential			
	WA	473,243		Potential			
Raymond	WA			Potential			
	WA			Potential			
Roosevelt	WA	13,000,000	Allied Waste Systems	Potential			
	WA	464,000		Potential			
	WA			Potential			
	WA			Potential			
	WA			Potential			
Kettles Falls	WA			Potential			
Vashon Island	WA	281,554		Potential			
Pulman	WA			Potential			
Yakima	WA		U.S. Army - Fort Johnson	Potential			
Colliers	WV	1,325,118	J.P. Mascaro & Sons	Candidate			
Bridgeport	WV	2,250,000	WMI	Candidate			
Parkersburg	WV	4,100,000	WMI	Candidate			
Beckley	WV	1,600,000	Raleigh County Solid Waste Authority	Candidate			
New Martinsville	WV	1,000,000	J.P. Mascaro & Sons	Candidate			
Charleston	WV	519,163	City of Charleston	Potential			
Hurricane	WV		WMI	Potential			
Elkins	WV		City of Elkins	Potential			
Lewisburg	WV		Greenbrier County	Potential			
Peterstown	WV		Ham Sanitary Landfill Inc.	Potential			
Hedgesville	WV	916,600	WMI	Potential			
Princeton	WV		Mercer County	Potential			
Hinton	WV	250,000	Midwest Disposal Inc.	Potential			
Calvin	WV		Nicholas County	Potential			
West Liberty	WV		Rohrig Sanitation Service Inc.	Potential			
Bluefield	WV			Potential			
Dunmore	WV		Pocahontas County	Potential			
Beckley	WV	672,000	Raleigh County	Potential			
Clarksburg	WV	660,000	WMI	Potential			
Short Creek	WV	2,985,349	American Disposal Services of West Virginia	Potential			
Hurricane	WV		Onyx Waste Services, Inc.	Potential			
Thomas	WV		Tucker County Solid Waste Authority	Potential			
Camden-on-Gauley	WV		Webster County	Potential			
	WI	1,050,000	Dane County Public Works	Operational	Dane County	0.9	
Madison	WI	2,094,058	Dane County Public Works	Operational	Dane County	1.6	
Racine	WI	9,425,000	Republic Services, Inc.	Operational	Northern Power Systems	3.2	2.080
Delavan	WI	3,434,413	Republic Services, Inc.	Operational	Republic Services, Inc.	2.4	
	WI		Marathon County Solid Waste Department	Operational	Becher Hoppe Assoc/Conestoga-Rovers		0.140
Ringle	WI	1,532,612		Operational	Bio Energy Partners		
Franklin	WI	3,666,667	WMI	Operational	Bio Energy Partners	6.5	
Franklin	WI	3,666,667	WMI	Operational	Bio Energy Partners	3.3	
Menomonee Falls	WI	9,000,000	WMI	Operational	Bio Energy Partners	2.0	
Horicon	WI	3,309,000	Onyx Glacier Ridge LF	Operational		2.0	1.100
Horicon	WI	3,309,000	Onyx Glacier Ridge LF	Operational		0.2	
Eau Claire	WI	1,200,000	Onyx Seven Mile Creek LF #2	Operational	Ameresco	3.0	
Menomonee Falls	WI	4,805,000	WMI	Operational	Bio Energy Partners	2.0	
Appleton	WI	3,994,902	Outagamie County	Operational	Outagamie County	1.2	
Menomonee Falls	WI	4,250,000	WMI	Operational	Bio Energy Partners	2.0	
Bristol	WI	11,000,000	WMI	Operational	Bio Energy Partners	5.6	1.296
Bristol	WI	11,000,000	WMI	Operational	Bio Energy Partners	3.3	1.728
Bristol	WI	11,000,000	WMI	Operational	Bio Energy Partners	3.2	1.728
Franklin	WI	3,957,234	WMI	Operational	BioEnergy Partners	2.4	
Baraboo	WI	625,000	Sauk County	Operational	Alliant Energy/RMT, Inc.	0.2	0.864
Berlin	WI	5,118,050	WMI	Operational	WMI		0.576
Oshkosh	WI	5,000,000	Winnebago County	Operational	Winnebago County	0.9	0.504
Oshkosh	WI	1,356,000	County of Winnebago	Operational	Winnebago County	2.1	1.190
Janesville	WI	1,540,000	City of Janesville	Construction	Ameresco, Inc.	3.1	2.000
Racine	WI	9,425,000	Republic Services, Inc.	Shutdown	Republic		0.600
Sarona	WI	469,302	BEI Waste Systems of N.A., Inc.	Candidate			
Green Bay	WI	5,400,000	Brown County Port & Solid Waste	Candidate			
Greenbay	WI	3,800,000	Brown County Port & Solid Waste	Candidate			
Medary	WI	697,680	La Crosse County	Candidate			
Delavan	WI	742,428	Republic Services, Inc.	Candidate			
Ringle	WI	1,060,000	Marathon County Solid Waste Department	Candidate			

Rhineland	WI	456,458	Oneida County	Candidate			
Wisconsin Rapids	WI	890,000	Onyx Cranberry Creek LF	Candidate			
Muskego	WI	3,500,000	Onyx Emerald Park LF	Candidate			
Chilton	WI	1,100,000	Onyx Hickory Meadows Landfill	Candidate			
Eau Claire	WI	960,000	Onyx Seven Mile Creek LF #1	Candidate			
Fort Atkinson	WI	1,800,000	Onyx Valley Meadows LF	Candidate			
Friendship	WI	43,626	Adams County	Potential			
Watertown	WI	308,640		Potential			
Sturgeon Bay	WI	348,089	Door County	Potential			
Easton	WI	3,833		Potential			
Berlin	WI	239,952	WMI	Potential			
Hayward	WI	133,120		Potential			
Hustisford Town	WI	5,150		Potential			
Black River Falls	WI	498,202	Jackson County	Potential			
Kewaunee	WI	146,359	Kewaunee Solid Waste Department	Potential			
LaCrosse	WI	86,304	La Crosse County	Potential			
Merrill	WI	84,520	Lincoln County	Potential			
Marinette	WI	199,631		Potential			
Mackford	WI	7,807		Potential			
Middleton	WI	574,518		Potential			
Sparta	WI	301,784	Peterson Construction	Potential			
Montello	WI	2,600,000	Onyx Waste Services, Inc.	Potential			
Wisconsin Rapids	WI	800,000	Onyx Waste Services, Inc.	Potential			
Greenbush	WI	10,260		Potential			
Custer	WI	699,000	Portage County Solid Waste Department	Potential			
Prentice	WI	3,627		Potential			
Princeton	WI	5,848		Potential			
Franklin	WI		WMI	Potential			
Lowville	WI	2,258		Potential			
Round Lake	WI	1,508		Potential			
	WI	446,623		Potential			
Sheboygan Falls	WI	146,700	Sheboygan Falls	Potential			
Superior	WI	826,738	City of Superior Public Works	Potential			
East Troy	WI	3,072,144	Allied Waste Industries	Potential			
Berlin	WI	5,118,050	WMI	Potential			
Cheyenne	WY	4,189,263	City of Cheyenne	Candidate			

(USEPA, LMOP)