ARPA TRP TURBOALTERNATOR DEVELOPMENT

94-67459(5), Rev. 1

Quarterly Report for the
Period of July through September 1995

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CALSTART, Inc.
Burbank, California 91505
and
AlliedSignal Aerospace
Torrance, California 90509

Date Published—October 1995

PREPARED FOR THE UNITED STATES
DEPARTMENT OF ENERGY

Under Contract No. DE-FC36-946010016

Aerospace Equipment Systems
Torrance

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1. TASK 1, UNDER-ARMOR AUXILIARY POWER UNIT (UAAPU)
DEMONSTRATION ON AN M1A1 TANK

1.1 INTRODUCTION

The first task of the Advanced Research Projects Agency (ARPA) Technology Reinvestment Project (TRP) program under Contract DE-FC36-946010016 was to modify an existing APU and then demonstrate the AlliedSignal advanced oilless and gearless small gas turbine technology on an M1A1 tank. The intent of this phase of the program was to use some of the ARPA TRP funds to show the direct military benefit of advancing this revolutionary technology.

1.2 PROGRAM STATUS

This part of the program ended in September 1994. The M1A1 UAAPU development and field test program was described in detail in the first ARPA quarterly report, published in October 1994. Details on this successful demonstration will not be repeated in this, or future, quarterly reports. No additional work was conducted on this program task during the current reporting period. No additional work on this task is planned for the balance of the program.

2. TASK 2, LOW-COST TURBOGENERATOR DESIGN

2.1 INTRODUCTION

The low-cost turbogenerator design effort portion of the overall TRP program was established to assist in the development of a highly producible, low-cost turbogenerator. The low-cost turbogenerator is a viable candidate to become the heat engine power source in a hybrid electric vehicle (EV) propulsion system for a new generation of clean burning automobiles and buses. The hybrid EV vehicle configuration, in which the turbogenerator is typically used, is the series hybrid arrangement.

2.2 TURBOGENERATOR SYSTEM SPECIFICATION

As reported in the last quarterly report, system and vehicle trade studies were being conducted to determine what near-term engine core design and performance parameters should be used in the development of the low-cost turbogenerator. These trade studies are being performed outside of this ARPA contract and are driven by the turbogenerator electrical power and performance requirements of several major car manufacturers.

During this reporting period, the turbogenerator configuration was finalized for an early hardware build. The final, early build turbogenerator operating conditions are presented below in Tables 2-1 through 2-3. A new condition has been added to the standard-day conditions in the working internal specification which AES has entitled the design reference point (DRP). This condition was created to provide a fixed reference condition against which performance issues are measured such as aerodynamic, recuperator, and generator efficiencies, as well as pressure drop changes.

TABLE 2-1						
STANDARD DAY PERFORMANCE (75°F AMBIENT TEMPERATURE)						
Power	DRP	Percent				
		100	75	50	25	10
Power, kW	52.2	50	37.5	25	12.5	5
Cycle efficiency, percent	28.4	28.7	28.0	26.2	21.0	13.8
SFC, gm/kW-hr	300	297	304	325	406	618
Speed, rpm	78,414	76,595	70,089	61,830	52,079	42,862
Inlet airflow, lb/sec	0.965	0.927	0.792	0.631	0.467	0.342
Turbine inlet temperature, °F (4.0)	1850	1850	1800	1800	1768	1690
Exhaust temperature, °F (7.0)	552	542	500	464	422	384

TABLE 2-2					
HOT DAY PERFORMANCE (120°F AMBIENT TEMPERATURE)					
Power	Percent				
	82	75	50	25	10
Power, kW	41	37.5	25	12.5	5
Cycle efficiency, percent	26.2	25.5	24.7	20.3	13.3
SFC, gm/kW-hr	325	333	344	419	639
Speed, rpm	77,940	76,809	66,802	55,413	45,878
Inlet airflow, lb/sec	0.870	0.851	0.661	0.478	0.353
Turbine inlet temperature, °F (4.0)	1850	1800	1800	1781	1700
Exhaust temperature, °F (7.0)	583	568	520	472	430

TABLE 2-3					
COLD DAY PERFORMANCE (-22°F AMBIENT TEMPERATURE)					
Power	Percent				
	100	75	50	25	10
Power, kW	50	37.5	25	12.5	5
Cycle efficiency, percent	32.6	31.1	28.1	22.0	14.2
SFC, gm/kW-hr	261	274	303	387	598
Speed, rpm	64,451	59,047	52,595	44,620	36,496
Inlet airflow, lb/sec	0.856	0.728	0.595	0.450	0.328
Turbine inlet temperature, °F (4.0)	1800	1800	1800	1744	1672
Exhaust temperature, °F (7.0)	401	378	354	320	290

2.3 LOW-COST ENGINE CORE DESIGN AND ANALYSIS

2.3.1 Introduction

Design layout and analysis activities of the 50-kW, high-performance, low-cost engine core made significant progress during this last quarter. The problem statement

used to design the engine core was the internal specification noted above. The results of these activities are presented in the following sections.

2.3.2 Engine Core-Level Design/Analysis Activities

2.3.2.1 Revised Layout for Early Hardware Builds

A revised layout, shown in Figure 2-1, was nearly completed during the current reporting period. This design is an alteration of the baseline/trade study machine presented in the last few quarterly reports. This new layout is the design basis for a functional engine core to be built with development-type tooling, etc. This layout also addresses some design details not finalized at the end of the initial trade study phase and incorporates some low-risk technical solutions in lieu of higher risk ones. Close cost tracking is continuing and a new bill of materials has been generated. Once preliminary detail design drawings are issued, new costing will be performed and the overall engine core price reviewed. If cost has been added to the to the baseline engine core design, parallel development plans will be made in the areas of increased cost. Actions will be taken to address removing these extra costs through further analysis and/or component testing.

2.3.2.2 Critical Speed

2.3.2.2.1 Analysis

Critical speed analytical efforts continued in support of the layout work presented above. To explain the unique rotor design approach taken in the engine core design, a rotor dynamic bode plot is presented as Figure 2-2. Modeling results to date are presented in Table 2-4 and in Figures 2-3 and 2-4. These results show acceptable speed margins between the first and second bending criticals and an ideal mode shape for traversing the first bending critical. In addition, the journal bearing length did not require any reduction for obtaining critical speed margin, therefore bearing sizing has not been compromised.

Further rotor design changes are expected to be small. The current rotor design is presented in Figure 2-5.

2.3.2.2.2 Super Critical Test Unit No. 2 Test Results

As presented in previous reports, early development testing has been ongoing to ensure that an air-bearing turbomachine could be operated super critically as discussed in the previous paragraph. Current development test efforts have focused on the second super critical test rig. During this last quarter, substantial testing was performed on this unit. A cross section of this unit is presented in Figure 2-6. Testing to date has been very successful. In Figure 2-7, bode and nyquist (polar) plots of the rotor dynamic behavior, as the rotor goes through the first bending traverse, is presented. The mode shape can be correlated with the analysis because of the use of shaft motion probes in the three planes noted.

INVERTED FLARED SEAT
SAE J512
.875-18 UNS THREAD
(.625 DIA TUBE)

OIL INLET

AlliedSignal
AEROSPACE

Aerospace Equipment Systems
Torrance

Figure 2-1. Updated 50-kw Turbogenerator
Engine Core Layout

Typical Air Bearing Turbomachine Rotor Dynamic Design Approach

Engine Core Rotor Dynamic Design Approach

IG-26864

Figure 2-2. Unique Super-Critical Rotor Design Approach

TABLE 2-4

CRITICAL SPEED MODELING RESULTS

- **Bending Natural Frequencies**
 - **First Bending = 34,300 rpm: 15% Below Min. Speed (40 krpm)**
 - **Second Bending = 106,500 rpm: 27% Above Max. Transient Speed (84 krpm)**
- **Mode Shape**
 - **Optimum: Maximum First Bending Rotor Displacements Occur at Journal Locations**
- **First Bending Displacement Levels with Moderate Unbalance Applied:**
 - **@ Turbine Wheel Nose = +0.0014-In.**
 - **@ Compressor Bearing (Ave) = -0.0017-In.**
 - **@ Thrust Rotor = -0.0018-In. (Max. Mid-Plane Displacement)**
 - **@ Outboard Bearing (Ave) = +0.0023-In.**
 - **@ Max. Cooling Air Pump Nose = +0.0035-In.**

IW-32175

Figure 2-3. First Bending Mode Shape at Natural Frequency (34,300 rpm)

IG-26861

IG-26862

Figure 2-4. Second Bending Mode Shape at Natural Frequency (106,500 rpm)

Total Rotor Weight = 17.65 lbs

Key Rotor Dynamic Parameters

$$I_p = 4.179 \times 10^{-2} \text{ lb-sec}^2\text{-in.}$$

$$I_d = 1.5588 \text{ lb-sec}^2\text{-in.}$$

IG-27102

Figure 2-5. Current Rotor Design

IG-26874

Figure 2-6. Super-Critical Test Rig No. 2 Cross Section

Results Match Critical Speed Model

IG-26873

Figure 2-7. Typical Polar Plot at 3 Measurement Locations During Traverse Through First Bending

Additional testing is planned in the upcoming quarter for a modification to this second test rig. The major difference will be a new compressor slug that will bring the second bending critical speed down to a similar value as the real rotor. This will allow additional investigations on engine core speed range capability, and will verify that the current assumptions on critical speed margins are adequate. In addition, trim balance information will be generated to achieve the best rotor dynamic compromise between the first and second bending criticals.

2.3.2.3 Thermal Modeling

During the current reporting period, the initial trade study thermal analysis was upgraded. As the layout progressed and detail design/analysis information became available on the generator, air bearings, and power head components, the thermal models were revised and new analytical results presented. These results were then reiterated back into the mechanical rotor and structure finite element models and loadings were investigated. This iterative process was still ongoing at the close of the quarter and the designs were being fine tuned to achieve acceptable thermal/stress solutions.

The thermal analysis activity is split into two parts to keep the finite element models manageable. Figure 2-8 presents the thermal model results of the generator end analysis, and Figure 2-9 presents the power head thermal analysis. In addition, the critical generator end maximum allowable temperatures are compared to the current thermal predictions in Table 2-5.

2.3.2.4 Rotor—Mechanical Design

A finite element modeling task for the overall rotor was just initiated at the close of this reporting period, with results to be presented in the next quarterly report. This rotor model will be used primarily to determine the necessary rotor lockup preload (to transmit the torque between components), radial fits, and all the rotor joints and shaft stresses.

2.3.2.5 Rotor/Bearing Cooling Flow Circuit

Following the detailed generator end thermal analysis, design work and a cooling flow analytical model were generated to determine how best to achieve the necessary bearing and rotor cooling. Results of this task are presented below.

In Figure 2-10, an overall schematic of the current cooling flow scheme with key flow rates is presented. In Figure 2-11, the front end of the engine core is shown with the cooling air takeoff and tubing, main cooling impeller, outboard journal bearing, and entrance to rotor gap noted. Pressures, temperatures, and flow rates are shown into and out of each of these engine core components. Note that there is a flow bypass leg around the journal bearing. Relatively high flows are required to keep the rotor cool enough to use the neodymium iron boron magnet material. However, the journal bearings do not require this amount of cooling, and if the entire flow was forced through the journal, excessive pressure drop would occur.

Figure 2-9. Turbogenerator Steady-State Temperature — Power Head

<p>TABLE 2-5</p> <p>GENERATOR END THERMAL PREDICTIONS AS COMPARED TO MAXIMUM ALLOWABLES</p>				
Component	Node Number	Temp - °F (Flow Rate)	Max. Allowable - °F	Comments
Stator	60	307	428	
Magnet	12	240	250	
Journal Bearing	78	319	600	
Thrust Bearing	48	449	600	
Cooling Oil Inlet	600	158	---	
Cooling Oil Outlet	613	172	350	
Cooling Air Inlet	200	175	---	Incl. 50 °F Rise Across Impeller
Cooling Air Outlet	214	271	---	
Stator Cooling Oil Flow	600	21 lb/min (9.5 kg/min)	---	
Turbine-End Bearings Cooling Flow	206	0.53 lb/min (0.25 kg/min)		
Rotor Cooling Air Flow	200	2.6 lb/min (1.2 kg/min)	---

Figure 2-10. Overall Schematic of the Rotor/Bearing Cooling Air Flow System (with Key Flow Rates Noted)

IG-27534

Figure 2-11. Cooling Flow Circuit on Front End of Engine Core

In Figure 2-12, the approximate center of the engine core is shown with the rotor exit and loaded side thrust-bearing cooling flow circuit. The loaded side thrust-bearing section also requires a bypass leg because the thrust bearing cooling requirements are significantly less than the 2.6 lb/min (required for rotor cooling) available, and high pressure drops would occur if a bypass were not incorporated.

In Figure 2-13, the other side of the thrust disk is shown and the inboard journal and unloaded side thrust bearing cooling flows are highlighted. This cooling flow circuit has a unique feature in that a jet pump effect is used at the compressor inlet plane to induce the small amount of flow needed through the inboard journal bearing.

2.3.3 Component Design/Analysis Activities—Generator End

2.3.3.1 Permanent-Magnet Motor/Generator

The analytical design of the generator was finalized during the last reporting period and detail design work was started. This effort was closely coordinated with the power electronics design (being performed under a separate contract), and the current generator design will produce the necessary power over an acceptable voltage range. In Table 2-6, a summary of the key generator electrical design parameters, as well as performance predictions, are presented. In Figure 2-14, the generator power loss breakdown is graphically presented to show the components that provide the performance predictions shown in Table 2-6.

2.3.3.2 Journal Bearing Analysis/Design

During the current reporting period, a journal bearing development plan was established, and analytical work was started and completed. As identified in the trade study phase, the preferred journal bearing sizing was to use a 1.6-in. dia bearing. Bearing length was left relatively long at 2.4 in. to maximize both load capacity and damping. However, definition of eventual bearing length was left flexible in the event overall length reductions were required. Length reductions to the journal design are more easily accommodated in tools, etc. than length extensions if changes are required at a later date. Table 2-7 gives the key journal bearing performance parameters.

2.3.3.3 Thrust Bearing Design

During the current reporting period, a thrust-bearing development plan was established, and analytical work was started and completed. An existing high-load capacity thrust bearing design exists in the AlliedSignal inventory. This bearing was developed for a space shuttle turbopump application and is an ideal fit to meet both the bearing performance requirements and physical geometry that the 1.6-in. journal diameter establishes. However, this design did require some fine-tuning since the film profiles were originally developed to work under much higher pressures than the current engine core problem statement (1000 psia vs 16 psia).

IG-26853-1

Figure 2-12. Rotor Exit and Loaded Side Thrust Bearing Cooling Flow Circuit

Figure 2-13. Inboard Journal and Unloaded Side Thrust Bearing Cooling Flow Circuit

TABLE 2-6
SUMMARY OF KEY GENERATOR ELECTRICAL DESIGN
PARAMETERS AND PERFORMANCE PREDICTIONS

● Generator Output Power, kw	50
● Rotor Speed, rpm	78,300
● Rotor Tip Speed, ft/sec	718
● Line to Neutral Output Voltage, Vrms	108
● Rated Output Current Per Phase, Arms	197
● Generator Output Frequency, Hz	1,305
● Electromagnetic Efficiency, %	93.6
● Total Generator Efficiency, %	92.4

IW-32181

TOTAL ELECTROMAGNETIC LOSS = 3.427 kW, $\eta_{em} = 93.6 \%$

TOTAL GENERATOR LOSS (INCL. WINDAGE AND FRICTION) = 4.111 kW, $\eta_{tot} = 92.4 \%$

IG-26866

Figure 2-14. Generator Power Loss Breakdown

TABLE 2-7	
JOURNAL BEARING PERFORMANCE SUMMARY	
START TORQUE = 7.3 LB-IN. (PER BEARING)	
POWER LOSS (AT 80,000 RPM MAXIMUM SPEED) = 250 WATTS (PER BEARING)	
LOAD CAPACITY (AT 80,000 RPM MAXIMUM SPEED) = 80-100 PSIA	
IG-27536	

The analytical work was completed on the bearing design modifications during this reporting period. Similar design challenges are imposed on the thrust-bearing design of the journal bearing design. The critical design issues are: (1) achieving the high-load capacity goals at near ambient operating pressures, and, (2) keep bearing preload down to keep bearing startup wear minimized. A summary of the thrust bearing performance parameters are presented in Table 2-8.

TABLE 2-8	
THRUST BEARING PERFORMANCE SUMMARY	
START TORQUE = 3.0 LB-IN. (PER BEARING)	
POWER LOSS (AT 80,000 RPM MAXIMUM SPEED) = 475 WATTS AVERAGE (PER BEARING)	
LOAD CAPACITY AT 80,000 RPM MAXIMUM SPEED) = 70 PSIA	
IG-27537	

Design activities were also initiated during this reporting period, including detail drawings of the bearings and thrust bearing fabrication tooling. Because an existing design was used, this task was minimized since only a new spring stamping tool is required. All other fabrication tooling exists. In addition, no new thrust-bearing test rigs are required.

2.3.3.4 Outboard Cooling Air Impeller

Following the detailed cooling flow and thermal modeling tasks, a small AlliedSignal turbocharger compressor impeller (turbocharger model no. GT15B) has been selected to perform the main cooling flow supply task. The aerodynamic problem statement for this component is presented in Table 2-9, and a performance map of the selected impeller with its operating point noted, is presented in Figure 2-15. As shown on the performance map, the impeller operates well off peak performance point (as expected) and appears close to its surge line. Off-design performance is not a significant issue since impeller power consumption is very low, but surge is a concern. Early testing is planned to verify both the cooling flow needs and the impeller performance. Impeller trimming, to achieve adequate surge margin, will be developed at that time.

TABLE 2-9

MAIN ROTOR/BEARING COOLING AIR IMPELLER PROBLEM STATEMENT

OPERATING SPEED = 80,000 RPM (CORRECTED SPEED = 75,000 RPM)
 COOLING FLOW REQUIRED = 2.60 LB/M (CORRECTED SPEED = 2.67 LB/M)
 INLET PRESSURE = 14.30 PSIA
 INLET TEMPERATURE = 120°F
 OUTLET PRESSURE = 18.70 PSIA
 PRESSURE RATIO = 1.30

IG-27538

2.3.3.5 Inboard Cooling Air Impeller

Development of the inboard cooling impeller (sometimes identified as impeller No. 2) was initiated during this reporting period, following the detailed cooling flow and thermal modeling tasks. Preliminary analysis has indicated that this small impeller design is not as straightforward a task as presumed during the trade study phase. The aerodynamic and physical design problem statement is presented in the list below:

- Inboard inlet: $N = 80,000$ rpm, $N/\sqrt{\theta} = 75,350$ rpm, $P = 14.40$ psia, $T = 125^\circ\text{F}$, $W = 0.40$ lb/min
- Inboard outlet: pressure/temperature = TBD, actual thrust bearing ΔP unknown
- Minimum inducer hub diameter = 1.0 in.
- Maximize pressure rise given physical bearing/rotor constraints
- Compressor efficiency not critical.

Preliminary analysis work identified the design requirement to best fit into the centrifugal compressor impeller category. Other options considered for this problem statement were an axial impeller (fan) and a mixed flow compressor, but both were discarded because of aerodynamic parameters substantially outside the normal design practice range. However, even in the centrifugal compressor design category, this is a tough problem statement because of the physical restriction of a very large inducer hub diameter (1.0 in.). The preliminary design results of the compressor impeller sizing and geometry are presented in the list below and in a graphical form as Figure 2-16.

- Inboard outlet pressure = 15.70 psia
- Inboard outlet temperature = 172°F

IG-26863

Figure 2-15. GT15B Turbocharger Impeller with Cooling Air Operation Noted

IG-26865

Figure 2-16. Inboard Cooling Air Impeller Preliminary Design

- Some design margin exists because thrust bearing ΔP is expected to be small
- Manufacturing impact:
No backward curvature in blade design
Need shrouded impeller: edge leakage critical

In addition, a preliminary clearance study (to determine the effect of edge leakage) was conducted on the design, and it was determined that the impeller must incorporate a rotating shroud to achieve the required pressure rise. This analysis result is presented in the graph of pressure rise versus impeller tip to shroud clearance shown in Figure 2-17.

2.3.4 Component Design/Analysis Activities—Power Head

Design activities on the aerodynamic components in the power head were accelerated during this reporting period now that a firm design problem statement was established. In this current reporting period, a substantial amount of analysis was performed with many of the analytical activities currently nearing completion. In the next quarter's report, all the analytical work will be finalized and detail drawings of the all the power head components are expected to be completed.

2.3.4.1 Compressor Aerodynamic Analysis

2.3.4.1.1 Preliminary Aerodynamic Design

Prior to performing the actual analytical design of the compressor stage, various parametric trade studies were performed to determine how best to optimize the compressor stage performance. These trade studies were grouped into the five areas noted below;

- Inlet geometry (flow area, hub radius)
- Impeller blade geometry (backward curvature, blade count, DA/DR, splattered?, exit radius)
- Rotational speed
- Aerodynamic issues (shroud relative velocity ratio, slip factor, loss multipliers)
- Vaneless space radius ratio.

To summarize the results of these trades, Table 2-10 is presented. In this table, an optimum compressor stage solution is compared to the selected preliminary design. There are a number of differences between the optimum and selected design, but the biggest contributor to the efficiency reduction (of approximately three points) is the

IG-26876

Figure 2-17. Inboard Cooling Air Impeller Clearance Study Indicating Need for “Shrouding” of Impeller

TABLE 2-10	
PRELIMINARY TRADE STUDY RESULTS AT 100 PERCENT POWER CONDITION	
Optimal Configuration	Selected Configuration
<ul style="list-style-type: none"> ● Impeller <ul style="list-style-type: none"> ● Inducer hub radius = 0.5 in. ● r, tip, in. = 1.33 in. ● Beta, ex = 55 deg ● Blade count = 15/15 ● DZ/DR = 1.5 ● r, tip, ex = 2.2 in. ● Nc = 70,000 rpm ● W2/WS1 = 0.5 	<ul style="list-style-type: none"> ● Impeller <ul style="list-style-type: none"> ● Inducer hub radius = 0.9 in. ● r, tip, in. = 1.54 in. ● Beta, ex = 55 deg ● Blade count = 15 ● DZ/DR = 1.5 ● r, tip, ex = 2.30 in. ● Nc = 76,939 rpm ● W2/WS1 = 0.5
<ul style="list-style-type: none"> ● Vaneless Space <ul style="list-style-type: none"> ● R/R = 1.07 	<ul style="list-style-type: none"> ● Vaneless Space <ul style="list-style-type: none"> ● R/R = 1.15
<ul style="list-style-type: none"> ● Diffusion System <ul style="list-style-type: none"> ● Omega Bar = 0.217 	<ul style="list-style-type: none"> ● Diffusion System <ul style="list-style-type: none"> ● Omega Bar = 0.212
= > Eff, ad = 0.816	= > Eff, ad = 0.787

large inducer hub radius. This is a parameter that cannot be changed due to the geometric constraints of the super critical rotor design operating on air bearings.

Following the development of a solution that was optimized around the peak power design point, an off-design study was undertaken. This activity (and eventual results) were identified early in the design process to be of great significance because of the need to have the overall engine core performance optimized around the 50-percent power condition. Two physical compressor characteristics were adjusted during this analysis activity to improve the off-design performance of the compressor. The characteristics changed and adjustment technique used are summarized below;

- Closed down the impeller inlet blade angle until the 100%Nc cycle flow matched near top of choke, or until stage was matched at peak efficiency for part power/speed (50-percent power)
- Added work by dropping exit beta angle until the goal cycle pressure ratio was obtained at 100%NC.

At the conclusion of this trade study effort, a preliminary design solution was obtained. The resulting performance predictions for the compressor stage are presented as Figures 2-18 and 2-19 in a conventional aerodynamic map form. The key standard day engine cycle operating conditions are plotted onto the maps. In the following section, detail design information on this preliminary compressor stage will be presented.

2.3.4.1.2 Preliminary Stage Geometry

At the conclusion of this preliminary aerodynamic analysis activity noted above, the preliminary compressor stage was defined. A summary of the key design parameters is shown in Figure 2-20. Additional definition of the three major components that comprise the compressor stage are discussed below.

2.3.4.1.2.1 Impeller Geometry

The impeller geometry and key aerodynamic characteristics were discussed in detail in the previous analysis section. Table 2-11 summarizes (and provides additional detail) of this preliminary impeller design solution.

2.3.4.1.2.2 Vaned Diffuser Geometry

In the early trade study of engine core work, it was decided to have the compressor stage incorporate a vaned diffuser channel prior to the scroll entrance. Although this is not the convention in a turbocharger, it was determined that it was required to meet the overall turbogenerator, and the compressor specifically, performance goals. The purpose of this channel is to take the high velocity air as it leaves the impeller blade tip and to minimize the total pressure (static pressure + dynamic pressure) loss. By use of a diffuser channel, the high dynamic pressure component is effectively changed into an increased static pressure prior to this now reduced velocity air being "dumped" into the exit scroll.

Two critical geometric factors established the boundaries for the vaned diffuser design. The first was the need to design around a relatively large vaneless space. A large vaneless space was chosen because of its influence on impeller blade vibration. Since an aluminum compressor impeller was selected, a large vaneless space will reduce the blade vibration forcing function and a more robust design is achieved at the expense of minor reduction to the compressor performance. The second geometric factor was determining how large a diffuser channel could be installed and not effect the overall packaging of the engine core. Typically, the bigger the diffuser the better but there is usually a point where the increasing benefits are not significant. Because of the internal packaging of the engine core details, a large diffuser was able to be incorporated without effecting the engine core package. Therefore, the compressor diffusion system is near optimum. Details of the vaneless space design (Figure 2-21), vane geometry (Figure 2-22), and vane channel aerodynamic parameters (Figure 2-23) are presented below.

2.3.4.1.2.3 Outlet Scroll

The outlet scroll is the hardware component that transitions the compressor flow from the exit of the diffuser channel into the duct, which is connected to the high-pressure side of the recuperator. This is also considered an aerodynamic component since the airflow continues to be effectively diffused to recover as much of the compressor stage dynamic pressure as possible. The scroll problem statement is defined in Table 2-12, and a preliminary design has been performed. Additional detail to the scroll

Figure 2-20. Preliminary Compressor Cross-Section

TABLE 2-11

IMPELLER GEOMETRIC AND KEY AERODYNAMIC CHARACTERISTICS

» Impeller Geometry:

- $r_{h,in} = .9''$
- $r_{tip,in} = 1.54''$
- $r_{tip,ex} = 2.30''$
- $\text{Beta}_{ex} = 50^\circ$
- Blade Count = 15
- $DZ/DR = 1.5$
- $N_c = 76,939 \text{ RPM}$
- $D_{sn} \text{ Clnrc} = .015''$

» Impeller Aerodynamics:

- $W_{c,in} = .993$
- $P/P = 3.66$
- $T/T = 1.522$
- $\text{Eff}_{ad} = .853$
- $V_{ms}/V_{mm} = 1.20$
- $W_2/WS_1 = .5$
- Slip Factor = .917 (inside)
- Slip Factor = .899 (outside)
- Loss Mult. = 1.00
- $W_2/WS_1 = .5$
- $\text{Alpha}_{ex} = 73.8^\circ$
- $M_{rel,shd,in} = 1.05$
- $M_{ex} = .86$

design will be presented in the next quarter's report as packaging of this hardware component is refined.

2.3.4.2 Compressor Impeller Mechanical Design Analysis

The compressor impeller mechanical design analysis is undertaken to ensure that the aerodynamic designs are mechanically sound. Iterations are conducted between the mechanical and aerodynamic solutions to achieve a design that maximizes performance, while providing sufficient structural life margins.

An aluminum compressor impeller was selected during the trade study phase as the only realistic low-cost solution. In these early trades, soak back was identified as a key structural concern and the more detailed analysis work that occurred this last quarter verified these preliminary results. However, system transient analysis (being performed outside of this program) had not been completed by the end of this reporting period, so soak back thermal/stress data will be presented in next quarter's report.

The starting point of the mechanical impeller analysis was to establish a baseline geometry, determine the thermal profiles in the blade during steady-state conditions,

100%Nc Peak Efficiency

Figure 2-21. Diffuser Vaneless Space Design

ITER_6 -- AUTO TURBOGENERATOR DIFFUSER \DEFAULT\ -- AUGUST 10, 1995
LEGEND:
--COMPUTED LOG SPIRAL

Figure 2-22. Diffuser Channel Geometry

Figure 2-23. Diffuser Channel Aerodynamic Parameters at 100-Percent Power Condition

and determine what cast aluminum materials offered the best technical/cost solution. A206 was selected as the material of choice and the minus three sigma properties (AlliedSignal design practice is to use minus three sigma) are presented in Table 2-13. The baseline geometry is presented in Table 2-14. The thermal maps for the 50-percent and 100-percent power points are presented as Figure 2-24.

Following these definitions, the steady-state mechanical impeller analysis of this quarter achieved the design goals of acceptable stress levels in the disk/hub and blade, as well as developing a blade design that is tolerant to vibratory inputs. In Figures 2-25 and 2-26, the disk and hub stresses are presented. In Figure 2-27, the blade stresses are presented. For pure overstress, the 100-percent power point is critical, but because the engine core will operate most of the time at 50-percent power and less, the stress levels at all the operating conditions must be considered in determining whether the life goals will be met. To summarize the design acceptability to pure over-stress (or burst), Table 2-15 is presented. In Figure 2-28, a Campbell diagram is presented to show that the blade interferences have been minimized, and that the four per revolution (inlet structure exciter) is below the critical first mode.

TABLE 2-12	
SCROLL DESIGN	
•	Scroll Parameters — $r_{\text{mean,in}} = 4.56$ — $r_{\text{mean,ex}} = 4.56$ — $r_{\text{ex}} = 1.32$ in. (if it were a circle) — $\Omega_{\text{Bar}} = 0.746$ — $M_{\text{in}} = 0.144$ — $M_{\text{er}} = 0.11$ — $\alpha_{\text{in}} = 54.5$ deg — $\alpha_{\text{ex}} = 0$ deg

TABLE 2-13				
CAST ALUMINUM 206.T71 MATERIAL PROPERTIES				
Temperature	75	150	250	400
Sy (.2% offset) ksi,	51.3	49.6	46.4	37.8
–3 σ	42.7	41.0	37.8	29.2
Su, ksi,	62.0	59.7	55.4	44.0
3 σ	53.6	51.3	47.0	35.6
Density, (lb/cu in)	0.101			

TABLE 2-14	
COMPRESSOR BASELINE GEOMETRY	
• Tip radius	2.282 in.
• Bore radius	0.171 in.
• RPM (100 percent power)	78414
• Tip speed	1575 ft/s
• Vaneless space	15 percent
• Number of blades	15/0
• Aluminum 206 material	
• Weight	0.991 lbf
• Ip	0.0027 in-lbf-sec ²
• Id	0.0019 in-lbf-sec ²
• CG axial location	-0.7155 in.

TABLE 2-15						
PRELIMINARY BURST ANALYSIS RESULTS						
Elastic Stresses (ksi)						
Operating Condition	σ_x	σ_y	σ_z	σ_1	σ_e	δ_{tip}
50% power RPM=62259 t=0	18.2	4.2	19.6	19.6	19.9	.005
100% power RPM=78414 t=0	29.0	5.7	31.4	31.4	31.0	.008
Burst Ratio						
@ 100% power, 198.6 F, Burst Factor = .85 Ratio = $\sqrt{\frac{(0.85)(49.4)}{(11.0)}} = 1.95$						

The life analysis is still very preliminary since better duty cycle and start-stop transient definitions are required. This will be the major focus of the upcoming quarters-compressor impeller analytical work.

2.3.4.2.1 Prototype Compressor Impeller Manufacturing Trial

A manufacturing trial on the compressor impeller was undertaken during this last quarter with only minor success. The intent was to try to cast an aluminum compressor impeller using rapid prototyping technology. A computer solid model of the initial impeller design (airfoils and disk) was generated and used to create a stereo lithography

Figure 2-24. Compressor Impeller Thermal Maps

RPM = 78414 Steady State Temperature

Iteration 10
Low hub flowpath
15 main blades
0.055 in. backface thickness

ANSYS 5.1 34
SEP 25 1995
15:36:37
PLOT NO. 8
NODAL SOLUTION
STEP=1
SUB =1
TIME=1
S1 (AVG)
DMX =0.122E-03
SMN =-1.79
SMX =31.443
A =0.056359
B =3.749
C =7.441
D =11.134
E =14.826
F =18.519
G =22.212
H =25.904
I =29.597

ANSYS 5.1 34
SEP 25 1995
15:36:38
PLOT NO. 9
NODAL SOLUTION
STEP=1
SUB =1
TIME=1
SEQV (AVG)
DMX =0.122E-03
SMN =2.455
SMX =31.006
A =4.041
B =7.214
C =10.386
D =13.558
E =16.731
F =19.903
G =23.075
H =26.248
I =29.42

Figure 2-25. Hub/Disk Stress at 100-Percent Power Condition

RPM = 62259

Steady State Temperature

Iteration 10

Low hub flowpath

15 main blades

0.055 in. backface thickness

ANSYS 5.1 34
SEP 26 1995
08:59:54
PLOT NO. 9
NODAL SOLUTION
STEP=1
SUB =1
TIME=1
S1 (AVG)
DMX =0.791E-04
SMN =-2.722
SMX =19.64
A =-1.48
B =-1.005
C =3.49
D =5.974
E =8.459
F =10.944
G =13.429
H =15.913
I =18.398

ANSYS 5.1 34
SEP 26 1995
08:59:55
PLOT NO. 10
NODAL SOLUTION
STEP=1
SUB =1
TIME=1
SPQV (AVG)
DMX =0.791E-04
SMN =1.573
SMX =19.921
A =2.593
B =4.631
C =6.67
D =8.709
E =10.747
F =12.786
G =14.825
H =16.863
I =18.902

Figure 2-26. Hub/Disk Stress at 50-Percent Power Condition

100% Power RPM = 78414 Steady State Temperature

Figure 2-27. Blade Stress at 100-Percent Power Condition

Figure 2-28. Impeller Campbell Diagram—Main Blade (Iteration 6)

part. This part was intended to be used as the actual pattern for a trial casting. However, when the stereo lithography part was inspected, it was determined to be dimensionally unacceptable. This task has been postponed for the time being, but other rapid prototyping techniques are being investigated. One good aspect of this trial was it gave the impeller design team an early opportunity to get an actual part to physically review.

2.3.4.3 Turbine Aerodynamic Analysis

The radial turbine aerodynamic design encompasses the development of four components. These four components are the inlet scroll, the inlet vane (or nozzle), the wheel, and the exit diffuser. The design condition selected for the turbine was the 50-percent power condition, and the aerodynamic problem statement is defined in Table 2-16. As an outcome of this quarter's analysis work, which optimized the turbine design at both design and off-design conditions, a preliminary design of all four components has been generated. This design is summarized in Table 2-17. More in-depth aerodynamic analytical work is currently in progress and a much more detailed presentation of the final analysis results will be presented next quarter.

2.3.4.4 Turbine Wheel Mechanical Design Analysis

2.3.4.4.1 Structural Analysis Design Definition

The turbine wheel mechanical design analysis is undertaken to ensure that the aerodynamic designs are mechanically sound. Iterations are conducted between the mechanical and aerodynamic solutions to achieve a design that maximizes performance while providing sufficient structural life margins. Within the engine core, the turbine wheel is expected to be the component with the lowest life margin. This is typical of most small gas turbines.

A Mar-M-247 "equiaxed" turbine wheel was selected during the trade study phase as the only realistic, high-performance, moderate cost solution. Mar-M-247 "fine grain" castings are typically used in high-performance small gas turbines, but was identified as being too costly during the trade study phase, and, therefore, eliminated as an option. To perform a valid turbine wheel structural analysis, several key items must be well defined; these are the definitions of the physical geometry, the life (etc.) design requirements, the duty cycle, and the start-stop transients. In the list below, the turbine wheel design requirements for this program are identified.

- Minimum of 4,000 hr of creep-rupture life
- Minimum of 15,000 cycles of LCF life
 - 5,000 cold start cycles
 - 10,000 hot start cycles
- Minimum of 122-percent burst margin
- Minimum material properties

TABLE 2-16

TURBINE AERODYNAMIC PROBLEM STATEMENT

- **Turbine designed at 50 % power (operated for 60% of time)**
- **PRTT = 2.0**
- **Corrected flow w corr = .63022 lbm/sec**
- **Corrected speed NRT = 29825 RPM**
- **Turbine efficiency η_{TT} = 89.3**
- **Clearances**
 - **axial cl = 0.025**
 - **radial cl = 0.015**
 - **Backface cl = 0.03**
 - **Inlet total temperature T_{Tin} = 2260 R (constant over most of the operating range due to catalytic converter)**
 - **Inlet total pressure P_{Tin} = 30.08 psi**
 - **blade tip speed v = 1630 ft/sec (2055 ft/sec @ 100% power)**

TABLE 2-17

RADIAL TURBINE PRELIMINARY DESIGN OF SCROLL, VANE, WHEEL, AND EXIT DIFFUSER

Scroll		Vane		Wheel		Diffuser	
Inlet area	8.55	B-width	0.57	Rotor tip radius	3.00	Inlet Swirl	15.000
RMS radius	5.30	No. of vanes	19.00	No. of blades	14.00	Area Ratio	1.563
Mach-No.	0.14	Solidity	1.24	Incidence	6.10	Pressure recovery	0.292
		Zweifel	0.44	TTrel/TTabs	0.94	DeltaP/P	0.009
		% Blockage	12.45	Reaction (Bld Spd)	0.43		
		Stator exit metal angle	77.00	Reaction (Rel Vel)	0.14		
				Axial Length	1.65		
				Exit Blade angle	-58.80		

- Non-damaging HCF
- Non-impairing creep

In Figure 2-29, the wheel pertinent dimensions and design parameters are specified. In Table 2-18, a duty cycle was developed based on the turbogenerator cycle analysis and information obtained from the car companies on how they intend to use the turbogenerator. And lastly, a preliminary start-stop transient (presented as Figure 2-30) was created based on our gas turbine experience. This transient was used because the turbogenerator start-stop transient analysis was still being worked and better data were not yet available.

One additional item required prior to conducting the detailed structural analysis of the turbine wheel is accurate thermal maps at the various operating conditions. A thermal map of the turbine wheel at the 50-percent power point is presented as Figure 2-31.

The detailed turbine wheel structural analysis entails optimizing and iterating the design as follows:

- Iterating the aero/mechanical design to optimize for:
 - (a) Maximizing creep-rupture life in inducer region
 - (b) Minimizing airfoil load to rotor hub
 - (c) Proper placement of inducer and exducer first mode vibration crossing in engine operation map
- Rotor hub optimization for LCF life
- Predicting the thermal response and optimizing the operating cycle for maximum life
- Detail life calculation at critical locations

The above list defines the basic structural design task. However, within the turbine wheel life calculations alone there are seven high stress areas that need to be optimized. These seven areas are identified in the bulleted list below.

- Airfoil inducer region (creep-rupture, HCF)
- Airfoil exducer region (HCF)
- Airfoil hub region mid chord (LCF)
- Airfoil hub near trailing edge region (LCF, TMF)
- Rotor saddle region (LCF, TMF)
- Rotor back-plate region (LCF)
- Rotor centerline region (LCF)

- 14 blades, 19 vanes
- Trel/Tabs of 0.94 at design point and .97 at 100% power level
- $\Delta p/p$ of 0.009
- Saddle radius is at 68% of tip radius
- Tip speed of 1630 ft/sec at design point and 2053 at 100% power level
- An^{*2} at exducer at design point is 3.8 and 5.6 at 100% power level

Figure 2-29. Turbine Wheel Pertinent Dimensions and Design Parameters

TABLE 2-18

DEVELOPED DUTY CYCLE FOR TURBINE WHEEL MECHANICAL ANALYSIS

% Power	Inlet Condition		Speed	TIT	% Operation	
100	Standard Day		78414	1850	3.8	
82	Hot Day	MAX. SPEED	79035	1850	0.15	4.0
100	Cold Day		64284	1800	0.05	
75	Standard Day		70284	1800	4.75	
75	Hot Day		77513	1800	0.18	5.0
75	Cold Day		59234	1800	0.07	
50	Standard Day		62259	1800	57.97	
50	Hot Day		67881	1800	2.25	61.0
50	Cold Day		52805	1800	0.78	
25	Standard Day		52044	1775	26.61	
25	Hot Day		55951	1791	1.03	28.0
25	Cold Day		44633	1751	0.36	
10	Standard Day		42320	1694	1.90	
10	Hot Day		45435	1705	0.07	2.0
10	Cold Day	MIN. SPEED	36076	1678	0.03

- Each start assumes one power level loading only
- An idle period of 1-3 minutes is desirable before each loading

Figure 2-30. Preliminary Estimate of Start-Stop Transient

Figure 2-31. Thermal Map of Turbine Wheel at 50-Percent Power Point

2.3.4.4.2 Preliminary Structural Analysis Results

The first design analysis result presented is with respect to creep-rupture life. Turbine wheel creep-rupture life is typically driven by the following parameters:

- Turbine inlet temperature and rotor speed
- Relative temperature to absolute temperature ratio
- Airfoil heat transfer to rotor hub
- Airfoil thickness distribution
- Adverse thickness tolerances
- Engine deterioration (based on speed not included yet)
- Engine ship margin distribution (based on speed not included yet)
- Engine general uncertainties (based on temperature)

The variation parameters noted at the end of the list have yet to be defined for the hybrid EV turbogenerator engine core. However, typical design criteria were used where possible for these parameters, and as these parameters get better defined, the creep-rupture life model and calculations will be updated. The result of the creep rupture life calculation is presented as Table 2-19.

The second analysis result to be presented is the turbine wheel blade vibration analysis. A picture of the first inducer and exducer blade vibration modes is presented in Figure 2-32. In Figure 2-33, a Campbell diagram is shown with the first inducer and first exducer mode interference (with the 19 nozzle vane count) crossing below the minimum operating speed.

In Figure 2-34, the stress results of the rotor hub "backplate" optimizations are presented. By tailoring the physical geometry, the low-cycle fatigue life in this area is expected to exceed 20,000 cycles. In Figure 2-35, the stress results of the rotor hub "centerline" optimizations are presented. As with the backplate optimizations, the turbine wheel physical geometry was adjusted in various areas to obtain a satisfactory solution. As Figure 2-35 shows, there are two relatively high stress areas. These preliminary results indicate an adequate low-cycle fatigue margin. However, this area is affected significantly by the start and stop transients, and some refinement of this optimization may be required if the transient analysis is substantially different than the current rough start-stop transients noted in the previous section.

The last turbine wheel preliminary stress analysis result, which by the end of the quarter was not undergoing iterations, was that of turbine wheel burst. This calculation is used to ensure that there is an adequate margin from pure structural overload. Average tangential stress is used for this calculation and a typical acceptance criteria is a burst margin of 125 percent. As Figure 2-36 shows, this criteria is met.

TABLE 2-19
CREEP-RUPTURE LIFE SUMMARY

- Critical span is at 2.712 in. radius in the inducer region
- Minimum 1% rotor duty cycle life is **10,500** hours
- Stress at critical span is 31 ksi at 100% speed

%Power	%Duty cycle	%Life damage
76-100	4	46.5
51-75	5	7.2
26-50	61	40.5
0-25	30	5.8

ITER-13

Figure 2-32. Inducer and Exducer First Mode of Vibration

ATG50 TURBINE BLADE ITER-13 CAMPBELL DIAGRAM

Figure 2-33. Turbine Wheel Blade Campbell Diagram

LCC LIFE IS EXPECTED TO
BE IN EXCESS OF 20,000 CYCLES

Figure 2-34. Rotor Hub Backplate Stress Results

ANSYS 4.4
SEP 11 1995
13:10:16
PLOT NO. 6
POST1 STRESS
STEP=1
ITER=1
SICE (AVG)
SMN =467.206
SMX =99877
SMXB=150230

ZV =1
DIST=1.65
XF =1.5
YF =-2.396
ANGZ=90
EDGE
A =5990
B =17035
C =28081
D =39126
E =50172
F =61217
G =72263
H =83308
I =94354

*TRANSIENT STRESS ANALYSIS
IS UNDERWAY

*LCF LIFE CALCULATION IS
PENDING THE TRANSIENT
ANALYSIS

1st CRITICAL LOC.
92 KSI

2nd CRITICAL LOC
55 KSI

Figure 2-35. Rotor Hub Centerline Stress Results

AVG. TANGENTIAL STRESS
IS = 58.6 KSI

MINIMUM BURST MARGIN
= 128.6%

@78,414 RMP AN $T_{BULK} = 1000^{\circ}F$

Figure 2-36. Rotor Hub Stress Result Showing Acceptable Burst Margin

Updates to the presented analysis results are planned for next quarter. In addition, the various major analysis tasks, which were still undergoing refinements at the close of the quarter and were not presented in this report, will be in the next quarterly report.

2.3.4.5 Static Structure Mechanical Design Analysis — Power Head

The power head static structure mechanical design analysis is a substantial task. Although these components do not have the rotational effects with which the compressor impeller and turbine wheel must cope, very high temperatures are present. Although pressure loading effects are low, large thermal gradients within the static structure components impose potentially very high stresses. This means that accurate thermal maps are critical to performing this analysis task. Because the thermal modeling task had been postponed until the engine cycle was finalized, not much static structure analysis activity had been performed prior to this quarter.

During this quarter, preliminary thermal models were generated (once the engine operating cycle was finalized) and design changes initiated. This process is very iterative, with the structural engineer working closely with the design and manufacturing engineers to develop realistic solutions. These very detailed designs are then modeled and analyzed. Typically, as one high stress area is resolved, additional high stresses show up someplace else. In the engine core static structure, this iterative process is shown in Figure 2-37. Material selections are also critical and the current materials selected for the early hardware builds are noted in Figure 2-38.

The static structure design results are still too immature to present at this time. Substantial design iterations were still ongoing at the end of the quarter to solve high thermal gradient issues. Resolution of these issues is a concern strictly from the point of not imposing increased cost to the engine core. Detailed results of the static structure analysis work will be presented in the next quarterly report.

- ◆ Newer configurations have
 - reduced part count
 - lower stress and longer life
 - reduced heat transfer

Figure 2-37. Static Structure Iterative Process

Figure 2-38. Materials of the Static Structure

3. TASK 3: NEGLIGIBLE EMISSION CATALYTIC COMBUSTOR DEVELOPMENT

3.1 INTRODUCTION

During this reporting period, fabrication of the catalytic combustor components and test rigs was completed. Testing was conducted on a generic catalytic combustion rig to investigate emissions during a startup transient. Testing continued for the preheater and the premix section of the bus demonstrator catalytic combustor. Detailed test planning continued for the full catalytic combustor rig.

The bus demonstrator catalytic combustor layout is shown in Figure 3-1. The combustor may be divided into the following modules:

Preheater—A conventional diffusion-flame combustor, which is necessary to preheat the catalyst to operating conditions during a cold start. Fuel flow to this unit is shut off after warmup.

Premix section—After warmup, fuel and air are premixed in this section, so that a nearly uniform mixture is presented to the catalyst inlet.

Catalyst section—Catalytic combustion of the fuel-air mixture occurs in this section.

3.2 CATALYTIC COMBUSTOR DESIGN AND FABRICATION

All hardware for the bus demonstrator catalytic combustor and associated test rigs were received by the end of this reporting period.

3.3 TEST RIGS

3.3.1 Catalytic Pipe Rig

Testing was completed for the catalytic pipe rig. This rig consisted of six catalyst discs installed in a pipe, using the same installation technique that will be used in the bus demonstrator catalytic combustor. Propane was used for the fuel to avoid potential problems with evaporation and autoignition. A total of about 25 hr of hot time was accumulated on the rig, with no hardware problems associated with the catalyst. The testing consisted of two parts:

Steady-State Performance Testing—This testing is intended to show catalyst conversion efficiency and emissions versus temperature, pressure, fuel-air ratio, and flow rate. The results showed that at all conditions, a catalyst outlet temperature above 1800°F was required to obtain 99-percent combustion efficiency. Emissions of NO_x and HC at that condition were less than 1 ppm, but CO was never less than about 100 ppm. Figure 3-2 shows measured CO versus catalyst outlet temperature for a range of flow rates. It is currently postulated that the CO was high because the walls of the catalyst pipe were too cold (about 1100°F), which caused quenching of the gas-phase reactions in the outer catalyst channels. In the bus demonstrator catalytic combustor, the walls are designed to run at about 1350°F.

Figure 3-1. Bus Demonstrator Catalytic Combustor Layout

Steady-State Catalytic Combustor Performance

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Figure 3-2. Measured CO vs Catalyst Outlet Temperature

Transient Startup Emissions Testing — Initial results showed that with 1000°F inlet air flowing when fuel is introduced, the catalyst exit temperature takes 10 to 20 sec to reach steady state. Significant HC and CO spikes were measured on the emissions equipment during that time. However, quantitative results are not yet available.

3.3.2 Preheater Rig Test

Rig testing was completed for the preheater section of the bus demonstrator catalytic combustor. Both steady-state performance and transient ignition testing was conducted. Steady-state testing was hampered by leaks that developed in the water-cooled emissions sampling rake, which made emissions measurements impossible. The leaks also resulted in very cold combustor exit temperature measurements whenever the thermocouple rake traversed through the lower half of the combustor exit plane. Figure 3-3 illustrates the measured combustor exit temperature distribution at the operating conditions that will prevail just prior to switching the fuel flow from the preheater to the premix nozzles. The upper half of the exit plane, which was probably not affected by the leaks, shows a circumferentially uniform temperature distribution without any significant hot streaks. This indicates that the preheater steady-state performance is acceptable, pending further testing after the rake is repaired. The rake is currently being redesigned for better durability and leak resistance. Further results of the steady-state testing indicate that the preheater pressure drop and wall temperatures are acceptable.

Ignition testing was conducted for both cold and hot (to 1000°F) preheater inlet conditions. This testing revealed that the fuel nozzle airblast air has a significant adverse effect on the preheater ignition performance. As shown in Figure 3-4, ignition fuel-air ratios were lower without airblast air than with the airblast air on. Evidently, at ignition conditions, the airblast air pushes the spray downstream past the ignitor, making starting more difficult. This indicates that engine starting will be improved if the airblast solenoid is left closed during start, and not opened until after the combustor is lit. The chances of damaging the catalyst and recuperator from a hot start will also be reduced.

3.3.3 Premix Rig Test

All preparations were in place to run the premix rig test early in the next reporting period. The objectives of the premixer test are to characterize the following:

- (a) Premixer pressure drop over the range of expected inlet corrected flows.
- (b) Pressure distribution over the external surface of the premix duct, in order to estimate the potential effect on internal mixing.
- (c) Evaporation and exit fuel-air distribution at full power and switchover conditions.
- (d) Occurrence of flashback or autoignition.
- (e) If time and resources permit, evaporation and exit fuel-air distribution at off-design points, for use by later programs.

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Figure 3-3. Measured Combustor Exit Temperature Distribution

Ignition Performance for ARPA Preheater

Figure 3-4. Fuel-Air Ratio at Ignition

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All testing will be performed with DF-2 or commercial grade 2-D diesel fuel.

All test hardware has been acquired. Required instrumentation has been identified and fabricated. The test plan has been written, and a data acquisition and reduction scheme has been developed.

3.3.4 Catalytic Combustor Rig Test

The catalytic combustor assembly will be tested in the rotating rake test rig early in the next reporting period. The assembly will consist of the preheater, the premix section, and the catalytic section. The objectives of the catalytic combustor test are to characterize:

- (a) Catalytic combustor pressure drop over the range of expected inlet corrected flows.
- (b) Emissions and exit temperature distribution.
- (c) Catalytic section wall temperatures.
- (d) Occurrence of flashback or autoignition.
- (e) If time and resources permit, the effect of fuel-bound nitrogen (FBN) on NO_x emissions.

In addition, a demonstration will be made of the complete combustor lightoff procedure, including (1) ignition of the preheater; and (2) switchover of the fuel system from preheater operation to premixer operation.

Most of the testing will be performed with DF-2 or commercial grade 2-D diesel fuel. If time and resources permit, a special low-nitrogen diesel fuel will be used to investigate the effects of fuel-bound nitrogen. For this purpose, a diesel fuel has been identified that has an organic nitrogen level below 40 ppm, and does not contain nitrate cetane improver additives.

3.4 PROGRAM SCHEDULE

Testing of the catalytic combustor hardware is scheduled for completion by the end of November 1995.

4. TASK 4: TURBOGENERATOR DEMONSTRATION ON AN ELECTRIC BUS

4.1 INTRODUCTION

Turbogenerator development work continued during the last quarter of FY 95. Although the conventional combustor turbogenerator system test schedule has been delayed by one month, the final vehicle installation date remains unchanged (para. 4.6 discusses this in greater detail.) Work during this quarter continued to support the three basic program objectives:

- (a) Integrate a turbogenerator into an electric vehicle.
- (b) Integrate a more efficient recuperator into the turbogenerator system.
- (c) Use the electric bus as a test bed for a low-emission catalytic combustor.

Most work during this reporting period concentrated on assembling and integrating all subcomponents into the turbogenerator system in preparation for initial system testing. The following sections discuss each portion of the development work in detail.

4.2 TURBOGENERATOR/VEHICLE INTEGRATION

Most vehicle integration work performed during this quarter focused on ensuring that all system subcomponents fit properly into the test rig. Because the rig represents the bus chassis, these subcomponents should also fit properly into the bus.

4.3 SYSTEM STATUS

There are two separate turbogenerator systems that will run during this demonstration program. These systems differ only in their respective combustor configurations.

The first system built will operate with a conventional diffusion-flame combustor. This combustor is the same combustor used on the M1 tank under armor APU. Predicted operation of the newer system components (recuperator and generator) will be checked with this initial turbogenerator configuration. This system will only be run in the laboratory.

The second turbogenerator system will run both in the laboratory and on the GM Hughes electric bus. This system uses a catalytic combustor. Although this combustor is much larger than the conventional combustor, its combustion is more thorough and the combustion process is performed at a lower temperature than the conventional combustor. As a result, the concentrations of carbon monoxide, hydrocarbon, and nitrous oxide gasses in the engine exhaust will be much lower than the engine with the conventional combustor. Task 3 of this report details progress of the catalytic combustor during the last quarter.

The first system is built and now being installed into the laboratory for system tests (para. 4.5 discusses this activity in greater detail).

4.4 SUBCOMPONENT STATUS

4.4.1 Turbogenerator Engine Core/Generator

The turbogenerator engine core and generator were built last quarter. Most component tests were completed during the last reporting period and at the beginning of this reporting period. The engine core and generator are now integrated into the turbogenerator system and are being prepared for the conventional combustor system tests.

4.4.2 Recuperator

AlliedSignal performed extensive tests on the recuperator used in the turbogenerator system during this reporting period. These tests included the following:

- (a) Pressure test the air (high-pressure) side of the recuperator to 109 psig.
- (b) Conduct a sustained pressure test on the air (high-pressure) side of the recuperator at 50 psig. Check the recuperator core for leaks.
- (c) Perform performance tests on the recuperator using a laboratory setup to simulate the turbogenerator system.

In the performance test set, the recuperator was run through several isothermal pressure drop and heat transfer performance points. The tests ran air to air (instead of air to gas) at somewhat lower flow rates and temperatures than the design point.

Heat transfer and pressure drop performance predictions used the actual recuperator inlet conditions and fluids used in the test. The pressure drops were higher than predicted on the air side, while the heat transfer performance was lower. The gas side pressure drops were very close to predicted values. Table 4-1 shows a summary of heat transfer test results versus predicted performance.

The extra pressure drop is caused by a small, tightly turned inlet duct on the recuperator air side. Further tests are planned in the upcoming quarter on a spare recuperator core with ideal inlet and outlet manifolds.

4.4.3 Combustor

Section 3 of this report details all combustor activity for the quarter, including all combustor work related to the bus demonstration.

4.4.4 Power Electronics Unit

During this reporting period AlliedSignal continued the buildup of the power electronics unit (PEU). Earlier in the quarter, the logic unit was tested and assembled. The logic unit consists of the digital motor controller, the fuel pump driver, and the DC-DC power supply. The power unit was also tested and debugged.

The PEU was assembled and a small motor was tested to ensure that the motor controller worked properly. It was then linked to the turbogenerator engine core and motoring tests performed. Thirty stop and start cycles were performed successfully.

TABLE 4-1

RECUPERATOR TEST RESULTS VS PREDICTED PERFORMANCE

Test Point No. 201			Test Point No. 202		
	Gas Side	Air Side		Gas Side	Air Side
Flow, lb/min	50.38	50.34	Flow, lb/min	54.23	50.98
T in, F	993	77	T in, F	972	77
T out, F	224.4 (231.25)	858.8 (841)	T out, F	247.3 (255.25)	860.5 (841)
P in, psia	160.09	23.5	P in, psia	16.38	23.65
P out, psia	15.91 (15.90)	22.11 (21.8)	P out, psia	16.18 (16.17)	22.23 (21.89)
Efficiency		0.854 (0.834)	Efficiency		0.875 (0.854)
Test Point No. 203			Test Point No. 204		
	Gas Side	Air Side		Gas Side	Air Side
Flow, lb/min	42.21	41.31	Flow, lb/min	42.28	40.18
T in, F	1010.5	81	T in, F	1007.5	82
T out, F	230.6 (242.25)	890 (881.75)	T out, F	242.5 (249.5)	899.5 (887.25)
P in, psia	15.69	21.07	P in, psia	15.72	20.83
P out, psia	15.54 (15.56)	19.92 (19.67)	P out, psia	15.57 (15.55)	19.70 (19.47)
Efficiency		0.871 (0.861)	Efficiency		0.883 (0.870)
NOTE: Numbers in parentheses () indicate test data					

The system required modification in both hardware and software to perform upchopping. Upchopping boosts the rectified voltage level out of the generator to meet the varying electrical bus voltage. The field-programmable gate array was first modified to perform upchopping. The digital signal processor firmware was upgraded to accomplish this task. The upchopping control loop was simulated to ensure the stability of the loop and to verify the control procedure.

A Variac was used to generate 3-phase input to the PEU. Output power of the PEU went into a resistive load bank; up to 2 kW of power was produced during these initial upchopping tests.

During these tests, electrical noise interfered with the central processing unit and logic devices in the PEU. This caused the upchopper to shut off in a short amount of time. The PEU wires are currently being rerouted, shortened, and shielded to eliminate this noise.

An RS232 communication channel is now being added to the PEU. This enables the PEU to communicate with a remote computer to monitor key digital data.

4.4.5 Fuel Control Unit

AlliedSignal has completed the software and hardware modifications (and check-out) of the fuel control unit that will control the conventional combustor in the first turbo-generator system. This unit will be integrated and run in the conventional combustor turbogenerator during the next reporting period.

Software for the second fuel control unit, which will control the fuel system in the catalytic combustor turbogenerator, is now under development.

4.5 LABORATORY TESTING

4.5.1 Engine Core

All engine core tests were completed at the beginning of this reporting period. Test results showed that the core engine power output is 20.36 kW at its operating point of 65,000 rev/min under steady-state conditions. This increases to 21.26 kW when the unit runs at 68,000 rev/min. Also, the unit output increases slightly (approximately 0.2 kW) if the core engine is cold, indicating that increased generator cooling may increase the power output of the engine core. Conventional combustor system tests will examine these trends in greater detail.

4.5.2 Conventional Combustor Turbogenerator System Test

System test setup activities with the conventional combustor continued during the last quarter. Figures 4-1 and 4-2 show this unit installed in the test rig. Note that the turbogenerator and test rig are physically installed in the test cell upside down. This is done to make the unit more accessible during the initial set of tests. These photographs show all system components integrated, except for the power electronics unit, the fuel control unit, and the fuel tank. The first two of these will be installed prior to the first system test. The fuel tank will be installed in the test rig when the unit is turned right side up.

AlliedSignal also developed a system test plan for the conventional combustor system during this reporting period, which is described in the remainder of para. 4.5.2.

4.5.2.1 Preliminary Motoring/Lightoff/Startup Tests

4.5.2.1.1 Fuel Pump/Nozzle Calibration

Plot the fuel mass flow rate vs fuel pump speed for DF-2 fuel using the ARPA nozzle. Measure the fuel mass flow rate by weighing the amount of fuel that flows out of the pump in a fixed amount of time at a set speed. Plot fuel mass flow rates from 0 lbm/min to 60 lbm/min.

Figure 4-1. Side View of Turbogenerator System Prior to Laboratory Testing

Figure 4-2. Frontal View of Turbogenerator System Prior to Laboratory Testing

4.5.2.1.2 PEU Input/Output Check

Turn PEU fuel solenoid on/off using the PEU external "run" computer. Run the fuel pump at different speeds with the nozzle detached from the turbogenerator.

4.5.2.1.3 Motor Unit With PEU Only

Motor the turbogenerator with the PEU at 320 vdc. Motor the unit to its maximum motoring speed. Do not run the unit for longer than 2 min. Ensure that a Doric meter is attached to the unit and monitoring the thrust bearing temperature during all motoring tests. Let the turbogenerator cool if this temperature reaches 300°F. Conduct 20 starts with a 1-min break between starts.

4.5.2.1.4 Communication Check Between DFCU/PEU

Establish proper communication between the PEU and DFCU. Ensure that both units work properly.

4.5.2.1.5 DFCU input/output check

Using the external "run" computer, turn the following DFCU-controlled components on/off: DFCU fuel solenoid; purge air solenoid; flow divider solenoid; and ignitor.

4.5.2.1.6 Initial Lightoff Tests Modes (0 to 5)

Run the unit as if it were going to be started. Once the unit enters Mode 1, immediately shut down. Look over the O-graph/Macsym/Labview data to ensure that the unit entered and operated in this mode properly. Run the unit again, except this time run through Mode 1, enter Mode 2, and shut down. Check all data again to ensure proper operation through Mode 2. Continue to repeat this gradual addition of modes until proper operation of Modes 1 through 5 is confirmed. Remember to shut the unit down immediately once the operation of the last mode of each test is confirmed. The modes are noted below.

MODE SUMMARY LIST

- MODE 0 - DFCU Initiation
- MODE 1 - Motoring Mode
- MODE 2 - Lightoff Initiation
- MODE 3 - Transition From Lightoff to Speed Ramp
- MODE 4 - Speed Ramp
- MODE 5 - Constant Speed Mode

4.5.2.1.7 Preliminary Lightoff Development

Adjust the prelightoff fuel flow schedule and the lightoff speed to minimize the time between the beginning of sparking and the increase of turbine inlet temperature (T4). Measure this delta T from the O-graph. Ensure that the unit starts properly with both cold and hot recuperators.

4.5.2.2 Steady-State Runs at Reduced Speed (50k rpm)

Run the unit steadily at 50 k rev/min. Ensure that the unit warms up completely. Check the compressor wheel operating point to ensure that is operating sufficiently far away from the surge line. Open the air bleed valve at the compressor outlet to move the compressor operating point away from surge, if necessary. Check the unit for air leaks along the entire airflow path. Check all Macsym data to ensure that all instrumentation works properly. Check all data along all airflow paths to ensure that the instrumentation values agree reasonably well with the predicted data. Check the temperatures on both sides of the thrust disc to ensure that the rotating assembly does not have excessive thrust in either direction. (The temperatures should be within 50°F of one another.) Check to ensure that the temperature difference between the sides of the thrust bearing agrees with the Labview thrust calculation. If the unit has too much thrust in the direction of the turbine, install orifice plates on the compressor wheel back-face vent until the thrust is balanced. Shut the unit down after completion of this test and repair any instrumentation that does not work properly.

4.5.2.3 Steady-State Runs at Design Speed (65 k rpm)

Run the unit steadily at 65 k rev/min. Ensure that the unit warms up completely and performs the same checks as in para. 4.5.2.2. Additionally, check both stator cooling mass flow rates. Ensure that a total of 5 lbm/min flows from both stator cooling outlet ports. (Enlarge the orifices in both fittings if the mass flow rate is low.) Shut the unit down after completion of this test.

4.5.2.4 Power Generation without Upchopper Operation

Warm up the unit at 65 k rev/min. Generate 5, 10, 15, 20 kW of power without the upchopper in operation. Run at least 15 min for each power level. Record all current, voltage, and power output data from the Yokogawa meter. Monitor the stator outlet air temperature carefully while the unit runs steadily at 20 kW. If this temperature ever exceeds 300°F, record the air mass flow rate through the stator cooling outlet ports and stop power generation. Enlarge both stator cooling flow outlet orifices and re-run the unit at 20 kW. At the conclusion of all runs, let the unit run at zero load at 65 k rev/min and shut it down once it reaches a steady-state idle temperature.

4.5.2.5 Power Generation with Upchopper in Operation; Re-check Generator Model

Conduct all tests as in para. 4.5.2.4, except with the upchopper in operation. Ensure that the upchopper operation does not electrically interfere with the laboratory instrumentation.

4.5.2.6 Endurance Testing

Run the unit for 6 hr continuously generating 20 kW with the upchopper in operation. Print a Macsym and Labview printout every 30 min.

4.5.2.7 Modify Unit to Bus Configuration

Turn the unit upside down (as it will be installed in the bus). Replace the laboratory recuperator outlet piece with the abrupt transition piece used in the bus. --Install the inertial particle separator onto the air inlet and disconnect the air inlet measuring section.

4.5.2.8 Preliminary Runs with Bus Configuration

Repeat the test described in para. 4.5.2.4.

4.5.2.9 Endurance Runs with Bus Configuration

Repeat the test described in para. 4.5.2.6.

4.5.2.10 Acoustic Testing

Run the unit at 65 k rev/min with no power generation. Take acoustical emissions (dBA) measurements 5 ft away from the exhaust outlet over a 180-deg arc. Perform an identical series of acoustical tests while the unit generates 20 kW. Initiate muffler design if the value exceeds 85 dBA.

4.5.3 Catalytic Combustor Turbogenerator System Test

These tests will include emission tests and the 2.1-sq in. nozzle installation. The final test plan will be finalized when preliminary conventional combustor test results are known.

4.6 SCHEDULE

Figure 4-3 shows the program schedule for the electric bus demonstration. The turbogenerator system testing has been delayed by four weeks because of the power electronics unit noise problems. Despite these delays, all activities beyond system testing will occur at the originally planned times.

Figure 4-3. Program Schedule for the Electric Bus Demonstration