EIGHTEENTH QUARTERLY PROGRESS REPORT
BASELINE GAS TURBINE DEVELOPMENT PROGRAM

COMPILED AND EDITED BY:

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Fred W. Schmidt

F. W. Schmidt
Associate Research Engineer

Charles E. Wagner Jr.

C. E. Wagner
Senior Research Staff Engineer
(Program Manager)

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ABSTRACT

Progress is reported for a program whose goals are to demonstrate an experimental Upgraded gas turbine powered automobile which meets the 1978 Federal Emissions Standards, has significantly improved fuel economy, and is competitive in performance, reliability, and potential manufacturing cost with the conventional piston engine powered, compact-size American automobile.

Initial running of the Upgraded Engine took place on July 13, 1976. The engine has proved to be mechanically sound, but was also 43% deficient in power. A continuing corrective development effort has to date reduced the power deficiency to 32%. Compressor efficiency was increased 2 points by changing to a 28-channel diffuser and tandem deswirl vanes; improved processing of seals has reduced regenerator leakage from about 5 to 2.5% of engine flow; a new compressor turbine nozzle has increased compressor turbine stage efficiency by about 1 point; and adjustments to burner mixing ports has reduced pressure drop from 2.8 to 2.1% of engine pressure. Key compressor turbine component improvements are scheduled for test during the next quarterly period.

During this past quarter, progress has also been made on development of the Upgraded Vehicle control system; and instrumentation of the fourth program engine has been completed by NASA. This engine will be used for development efforts at NASA LeRC.

TABLE OF CONTENTS

	<u>Page</u>
Introduction	1
Progress Summary	3
Detailed Progress Statement by Task	5
Task 3.1 Support to ERDA	5
Car 667	5
Car 671	5
Shows and Exhibits	5
Special Show Vehicle	5
Task 3.2 Support to NASA	5
Engine Test	5
Regenerator Test Rig	6
Task 5.7 Hydromechanical Transmission	6
Task 7.2-7.3 Upgraded Engine Procurement and Assembly	7
Procurement	7
Assembly Status	7
Task 7.4 Upgraded Engine Test	7
VIGV Engine Performance Map	7
Summary Test Log	8
Engine 1	8
Engine 3	10
Task 7.5 Upgraded Engine Corrective Development	12
Compressor	12
Compressor Turbine	14
Power Turbine	19
Component Test Rigs	19
Combustor	20
Regenerator Flow Distribution	21
Regenerator Development	21
Regenerator Wall Matrix Leakage	21
Regenerator Crossarm Seal Development	22
Regenerator Cover Temperatures	22
Engine Housing Metal Temperatures	23
Compressor Abradable Shroud	24
Gas Bearing Development	25
Integrated Control System	27

TABLE OF CONTENTS (Con't)

	<u>Page</u>
Task 8.2 Build Upgraded Vehicles	28
Car 130	28
Car 385	28
Task 8.3 Upgraded Vehicle Evaluation	28
Car 386	28
Task 10 Improved Automotive Gas Turbine (IAGT) Design Study	30
Superalloy Base Materials	30
Properties of Refractory Alloys	30
Cycle Studies	31
Future Activities	33

INTRODUCTION

This Baseline Gas Turbine Development Program is being sponsored by the Heat Engine Systems Branch, Division of Transportation Energy Conservation (TEC) of the Energy Research and Development Administration (ERDA). ERDA was established in January 1975 in accordance with the Energy Reorganization Act of 1974. Prior to this reorganization, TEC had been the Advanced Automotive Power Systems Division (AAPSD) of the Environmental Protection Agency (EPA).

The overall mission of AAPSD, which resulted in the generation of this particular program, had been to evaluate, develop and demonstrate automotive powerplant concepts potentially superior, in the area of emissions control and fuel economy, to the standard spark ignition reciprocating engine. A principal area of activity has been the automotive gas turbine. Programs were first initiated in this area early in 1971 when several contracts were let for the development of low emissions combustion, and for studies related to cycle optimization, engine arrangement, and cost to manufacture. Additionally, EPA enlisted NASA for technical support to these programs through an inter-agency agreement.

This Baseline Engine program was implemented on November 22, 1972 with the signing of a contract between EPA and Chrysler Corporation. Specific goals of this program were to demonstrate in 1976 an experimental gas turbine powered automobile which would meet the 1978 Federal Emissions Standards, have significantly improved fuel economy and be competitive in performance, reliability and potential manufacturing cost with the conventional piston engine powered, standard size American automobile.

Structurally, this program was made up of three parts: (1) documentation of the existing automotive gas turbine state-of-the-art; (2) conduction of an extensive component improvement program; and (3) utilization of the improvements in the design, and building of an Upgraded Engine capable of demonstrating program goals.

Because this program was relevant to the mission of ERDA's TEC Division, the program remained intact following the change in administration from EPA to ERDA.

Chrysler Corporation's Sixth Generation Gas Turbine Engine, having been selected as representative of the state-of-the-art at the beginning of this program, was used as the Baseline Engine. This

was a 4:1 pressure ratio regenerative engine with variable power turbine nozzles which meets the .41 HC, 3.4 CO, 3.0 NOx grams/mile original 1975 emissions standards. The Baseline vehicles were intermediate-size, 4-door sedans modified to accept the turbine engines.

As is shown in the detailed Program Plan (Appendix C), Baseline documentation (Tasks 1 and 2) was accomplished early in the program. Also as shown, the Baseline Engine improvement work (Tasks 5 and 6) has been completed except for a recently added Task 5.7 effort for verifying an MTI hydromechanical CV transmission concept. Building and testing of the Upgraded Engine and Vehicle Tasks 7 and 8 are the major remaining efforts.

Initial running of the Upgraded Engine took place on July 13. This engine though mechanically sound started out about 43% deficient in power. The principal problem has been low efficiencies of the aerodynamic subsystems resulting from compromises made to satisfy packaging/interface and other design constraints. Changes in the compressor along with some reductions in leakage and pressure drop have currently reduced the power deficiency to about 32%. A number of improved turbine and compressor section components are scheduled to be tested in the next quarterly period. ERDA is continuing to support corrective development of this engine to provide a sound base for their advanced engine programs.

Chrysler Corporation efforts on this program are being managed and coordinated by the Research section of the Engineering Office. Those who have contributed directly in writing this report include C. Belleau, A. Billington, B. Chapman, J. Corwin, F. Dosenberger, C. Elliott, J. Engel, T. Golec, J. Gross, J. Gumaer, F. Hagen, E. Kohl, H. LeFevre, J. Lewakowski, C. Mader, D. Musgrave, T. Nogle, R. Pampreen, L. Pritchard, N. Sparks, R. Swiatek, and E. Trieskey.

PROGRESS SUMMARY

- . An expanded Upgraded Engine and related displays were exhibited at three technology conferences.
- . The thoroughly instrumented NASA engine is being assembled.
- . Chrysler has started verification tests on the MTI pump/motor module of the hydromechanical transmission.
- . Procurement for the remaining three vehicle engines has begun.
- . Upgraded Engine performance with variable inlet guide vanes has been mapped.
- . Upgraded Engines 1 and 3 are undergoing various corrective development tests on the dynamometer, with the overall power decrement reduced from 43% to 32% of goal:
 - Compressor efficiency was increased 2 points by changing to a 28-channel diffuser and tandem deswirl vanes.
 - Improved processing of seals has reduced regenerator leakage from about 5 to 2.5% of engine flow.
 - A new compressor turbine nozzle has increased compressor turbine stage efficiency by about 1 point.
 - Adjustments to burner mixing ports has reduced pressure drop from 2.8 to 2.1% of engine pressure.
- . Operation of the power turbine near design conditions by using inflow bleed indicates that the power turbine is 5 to 10 efficiency points below goal.
- . Three component test rigs to aid compressor development and turbine performance evaluation have been proposed and design efforts have started.
- . Upgraded Car 386 has undergone control/engine response tests and some emissions testing at steady state and transient conditions.

- . The Upgraded Engine specifications have been updated and are included in their entirety in Appendix B.
- . The design study for an improved automotive gas turbine engine, Task 10, was initiated in this quarter.

DETAILED PROGRESS STATEMENT BY TASK

TASK 3.1 SUPPORT TO ERDA

Car 667 - Baseline

This vehicle is in use by ERDA personnel in Washington, D.C. Two service trips were required during this period. The first was to replace a leaking fuel control and the second was to replace a malfunctioning power turbine nozzle actuator. The servo valve bushing of the replaced actuator had shifted. It was repaired by pinning it in its proper location and is a useable spare.

Car 671 - Baseline

This vehicle is runnable but deactivated in that it is not suitable for demonstration use without extensive service to the steering system.

Shows and Exhibits

An "expanded" Upgraded Engine, Figure 1, was built up and displayed along with mock-up vehicle 385 at the ASME Gas Turbine Conference in Philadelphia, March 28 through April 1. The engine was next displayed along with two running vehicles, Upgraded Car 386 and Baseline Car 667, at the Washington, D.C., ERDA sponsored Fourth International Symposium on Automotive Propulsion Systems April 18 through 22. Here a formal presentation on the Upgraded Engine low emissions burner system was given by T. D. Nogle. The week of April 25 through 29, the engine was displayed at a Chicago Electric Car Symposium.

Special Show Vehicle

Work has begun on a special show version of one of the six Upgraded vehicles. Concept approval is scheduled for the week of May 23. Rework will include new front end, instrument panel and special interior.

TASK 3.2 SUPPORT TO NASA

Engine Test

The engine parts sent to NASA as reported in the previous Quarterly were returned to Chrysler on April 14 thoroughly instrumented. Over 100 temperature and 60 pressure measurement sensors were installed (see Figure 2).

Engine assembly has begun by NASA personnel assisted by Chrysler and will be completed the week of May 16. Because this engine offers potential for adding significantly to the understanding of component performance, shakedown testing at Chrysler will include an initial recording of all instrumentation. It is scheduled for return to NASA Lewis mid-June in a package including build-up logs and airfoil inspection traces.

Regenerator Test Rig

The housing for this fixture was inspected, pressure checked and insulated. Following installation of regenerator, seals, cover and gas generator support with burner vortex, it will be shipped to NASA Lewis.

TASK 5.7 HYDROMECHANICAL TRANSMISSION

The pump-motor module of the MTI hydromechanical transmission has been delivered to Chrysler for completion of its scheduled test program. Consequently, the design and construction of test fixtures and equipment plus preparation of a dynamometer cell constituted the major activities of this development period. A gas turbine was selected as the source of input power for tests of the pump-motor module with an electric dynamometer providing the desired module loading. Preliminary observation of the performance of this combination of test cell components indicates that turbine noise, being of relatively high frequency, causes insignificant interference with audible noise emitted by the transmission module; and hence, it appears that module noise frequency measurements will be more meaningful than originally anticipated.

A procurement delay has resulted in testing to date with a torque meter of greater capacity than desirable. Therefore, its readings at low torques require careful calibration adjustments by procedures still being resolved. At higher torques, however, the data appear reliable. Checks at specific operating conditions covering a range of speed (600-2400 rpm) and load (10-60 BHP) conditions indicate good agreement between MTI and Chrysler data with the latter usually showing slightly higher (1-4%) efficiencies. Chrysler has acquired data with the module operating in four ratio ranges from -1:1 to +1.8:1 with best efficiencies being noted in direct drive (+1:1) ratio. Noise frequency tests are being conducted with the pump-motor module operating and with the module replaced by a solid propeller shaft which permits measurement of background test cell noises.

TASK 7.2 - 7.3 UPGRADED ENGINE PROCUREMENT AND ASSEMBLY

Procurement

Ordering of long lead time items for the three additional vehicle engines has begun.

Assembly Status

Figure 3 summarizes the status of powerplant assemblies. To date, the program is limited to the seven engine housings listed because of the casting vendor's continued difficulty in producing parts of acceptable quality. A total of 14 housings are required to meet engine, vehicle and rig requirements. Of the five completed engines, two are in test cells, one is in Car 386, another is the NASA engine discussed in Task 3.2, and one is being used for display purposes.

For better alignment and concentricity of the power turbine wheel shaft and pinion, electron beam welding has been abandoned in favor of inertia welding.

To support the corrective development program, all key aerodynamic flow surfaces are inspected before assembly. Impeller, turbine wheel blade and nozzle vane airfoil contours are inspected using the optical comparator shown in Figure 4 resulting in a typical tracing as shown in the Figure. Compressor cover, air inlet bowl snout and compressor turbine nozzle inlet and deswirl vane contours are inspected utilizing the fixture shown in Figure 5 which provides a printout of the actual locations of blueprint coordinates. These data are fed into the computer which plots the actual versus required contour and calculates the error. Figure 6 shows a typical plot.

TASK 7.4 UPGRADED ENGINE TEST

During this past quarter, two dynamometer cell engines have been under continuous test. Most of this testing has been directed towards the correction of low engine power and is discussed in the Task 7.5 Corrective Development section of this report. However, one item of general engine documentation was also accomplished; this being an initial test of performance at various variable inlet guide vane (VIGV) settings.

VIGV Engine Performance Map

The operating map with negative VIGV angle settings is shown on Figure 7 and indicates negligible augmentation potential, as would be expected from an engine with deficient compressor

and compressor turbine. Once these elements are corrected and pressure ratio over the compressor turbine stage is reduced to design levels, the 10% augmentation goal should be demonstrable.

The operating map with positive angle settings is shown on Figure 8. These data indicated that use of the compressor inlet guide vanes to control engine output power at low output power levels appears superior to gas generator speed control for cyclic vehicle operation. At gas generator speeds from 50 to 60% and inlet guide vane angles up to 40° in the positive swirl direction, the engine operates along the minimum BSFC line shown on the Figure. The implication is that the guide vanes can be utilized to minimize the gas generator speed transients required for urban cyclic operation, thus providing better vehicle driveability with improved emissions. In addition, cyclic fuel economy should be improved for duty cycles requiring many gas generator speed changes. Current design specifications use positive angle inlet guide vane modulation only at 50% speed.

Summary Test Log

For the purpose of reference documentation, following is a log of the various tests run during this reporting period:

Engine 1 (Power Plant 3-1)

Fifteen engine builds were tested and 159 full sets of engine data recorded. Total engine time is 141.8 hours, of which 86.2 were logged this quarter. Specific tests were:

Build T, Runs 188-194

Engine housing and regenerator cover temperatures were recorded over the 50-100% Ngg range. The sprung-bearing power turbine wheel was run up to 46,000 rpm with no bearing noise or vibration noted.

Build U, Runs 195-250

Ran VIGV performance map, and used a new heavier regenerator seal baseplate for reduced distortion.

Build V, Runs 251-255

Vaneless diffuser inlet tapered to eliminate the chance of a step in the flowpath at the impeller discharge. Static taps added to compressor turbine nozzle throat to compare with trailing edge plane taps under influence of wheel blades (no wheel proximity effect).

Build W, Runs 256-260

Tested compressor diffuser with channels tapered 4° per side to reduce wake size.

Build X, Runs 261-274

Tested tandem compressor diffuser deswirl vanes.

Build Y, Runs 275-278

Tested 28 channel, $+2^{\circ}$ compressor diffuser.

Build Z, Runs 279-283

Burner modifications made to eliminate a flash-back problem above 80%; unsuccessful.

Build AA, Runs 284-287

Tested with VIGV inlet replaced by a vaneless, three-strut inlet.

Build AB, Runs 288-296

Tested with $+1^{\circ}$ 28-channel compressor diffuser.

Build AC, Runs 297-301

Tested with $+0^{\circ}$ (nominal) 28-channel compressor diffuser. Surge encountered at 85% speed.

Build AD, Runs 302-308

Burner premixer further modified to control flashback by increasing the flow area of the inner swirl; significant improvement.

Build AE, Runs 309-316

Tested burner vortex modified for increased vorticity. Tested with $+1^{\circ}$ 28-channel diffuser modified with a 4° exit taper to reduce exit wake.

Build AF, Runs 317-321

Measured static pressures at the impeller discharge, channel discharge, and deswirl vane inlet and exit planes for the original release compressor/diffuser/deswirl arrangement; used increased vorticity vortex.

Build AG, Runs 322-336

Tested Mod. 1 compressor turbine nozzle together with 28-channel +1° diffuser with 4° exit taper and tandem deswirl vanes.

Build AH, Runs 337-344

Retest of enlarged compressor collector (see Figure 13 of Seventeenth Quarterly) with impeller discharge static taps; used standard vortex.

Next scheduled tests will evaluate an axial compressor inlet where engine starting will be accomplished by impingement of shop air on the impeller vanes.

Engine 3 (Power Plant 2-3)

This engine has P_{6T} and T_{6T} compressor turbine wheel discharge instrumentation and as such priority has been given to its usage for testing compressor turbine nozzle and wheel modifications. A total of 13 engine builds were run, and 114 full sets of data recorded. This engine now has 99.3 hours, of which 61.9 were run this quarter. Specific tests were:

Build F,G,H, Runs 42-69

Tested with and without deflectors mounted at the diffuser discharge (see Figure 12 of Seventeenth Quarterly).

Build I, Runs 70-75

Initial test of increased plenum volume compressor collector (see Figure 13 of Seventeenth Quarterly). On Run 74, the inlet air filter box assembly was removed, reducing the 80% speed inlet total pressure loss from 5.8 to 0.8 inch of water, and increasing power from 26.7 to 28.2 hp.

Build J, Runs 76-99

Tested cutback trailing edge compressor turbine nozzle. Ran inflow bleed of shop air to compressor collector to evaluate power turbine at design temperature pressure and flow.

Build K, Runs 100-105

Tested regenerator seal with modified communicating groove; no change in performance.

Build L, Runs 106-113

Tested a compressor turbine wheel with cutback leading edge and a compressor impeller with reduced tip diameter.

Build M, Runs 114-120

Initial test of Mod. 1 (Chrysler design) machined nozzle (1.08 x design flow size). Tested with 14-channel compressor diffuser with +4° taper to inlet throat.

Build N, Runs 121-124

Initial test of abradable shroud compressor cover.

Build O, Runs 125-132

Retest of Mod. 1 nozzle with blade trailing edges closed down to nominal flow size (1.002 x design). Burner flashback problem at high speeds.

Build P, Run 133

Tested premixer which showed improved flashback resistance on Engine 1, Room 12S. Burner improved, but engine surged repeatedly at 90% speed (14-channel +4° taper enlarged throat diffuser).

Build Q, Runs 134-153

Tested Mod. 1 nozzle with 28-channel +1° diffuser with 4° taper, plus tandem deswirl assembly. Also, modifications made to reduce burner pressure drop.

Build R, Runs 154-156

Tested with tandem deswirl assembly removed. Power dropped 5%.

The next test will be to evaluate a machined version of the .8 scale Baseline compressor turbine wheel design.

TASK 7.5 UPGRADED ENGINE CORRECTIVE DEVELOPMENT

Compressor

Initial engine test results showed that the miss in engine power goals could be partially due to the miss in compressor efficiency goals. The magnitude of the deficit in efficiency goals was not expected on the basis of test rig results. However, the rig compressor was not tested with the engine intake, the engine configuration of the exit collector or the split inducer/impeller configuration. The split impeller was the result of configuring a separate steel inducer to accommodate water injection. The engine exit collector consists of three lobes of differing size in an asymmetrical arrangement; the rig collector was proportionately larger and of cylindrical shape.

The Seventeenth Quarterly Report presented the results of tests which initiated the compressor corrective development activity. Two of these tests addressed two of the differences between rig and engine configurations - the inducer/impeller gap and the exit collector. The test results showed a one-point gain in efficiency at 80% speed by sealing the gap. However, the results also showed total-total efficiency at 80% speed of 0.754 and a static-total efficiency of 0.731. This was in contrast to NASA test rig results, which showed a static value of 0.795, with approximately 1/2-point difference between total-total and static-total efficiencies.

In the last Quarterly Report, it was shown that the compressor exit Mach number at 80% speed was 0.19, in contrast to the rig value of 0.10. It was speculated that the small engine collector was adversely affecting the diffuser recovery through a back-pressure effect. To counter this effect, measures were proposed for minimizing the effect through improved diffuser exit conditions and through modifications to the diffuser itself.

It was suggested that a second cascade be installed behind the existing one to remove the remaining swirl and lower the exit Mach number to values obtained on the test rig. The feasibility of this approach was explored with a simple flat-plate cascade. As reported in the last Quarterly Report, the Mach number was lowered, but was not quite as low as rig values. In addition, about 2 points were lost in total-total efficiency and 1/2 point in static total efficiency.

During this quarter, a proper cascade configuration was designed to achieve the Mach number reduction with a net gain in static-total efficiency. In addition, reductions were made in the channel diffuser vane trailing edge thickness to minimize the width of the wakes discharging into the deswirl vanes. It was suggested that

thinner wakes would contribute toward minimizing the back-pressure effect. Furthermore, a 28-channel diffuser with much thinner vanes was designed by NASA and tested. Vane setting angles of $+1^\circ$ and $+2^\circ$ from nominal were tested along with the nominal setting. The alternate angles were selected on the basis of providing adequate surge margin in case the nominal setting adversely affected the surge line.

In addition to diffuser modifications to help reduce the back-pressure effect, changes were made to the collector itself. These consisted of the deflector ring and the enlarged version of the collector, as described in the last report. The purpose of these changes were to minimize the back-pressure effect by means of better diversion of the flow to the lobes and by an increase in plenum volume.

- Test Results

Figure 9 shows a plot of compressor efficiency along the engine operating line. In the figure, comparison is made between program goals, rig results and engine results. For consistency, the rig values of work coefficient were used to compute efficiency from engine pressure measurements. The rig results showed approximately 1/2 point difference in static-total and total-total efficiency. The engine compressor total-total efficiencies are higher than rig static-total values up to 70% speed, but lower beyond 80% speed. On the other hand, the difference between static- and total-based efficiency is much larger for the engine data (2.5-4.0 points) than for the rig data (about 1/2 point, not shown). Neither of the collector modifications made any change in compressor efficiency. Improvements were made through diffuser changes.

The configuration of the tandem deswirl vanes was shown in the last report. Figure 10 shows the modifications made to the original 14-channel diffuser, the shape of the 28-channel diffuser and the modification made to it. The test results are shown in Figure 11. Tandem deswirl vane increased static-total efficiency slightly (0.5-1.0 point). Reduction of trailing edge thickness on the 14-channel diffuser increased efficiency in a range from 0.4-1.4 points, depending on speed. The 28-channel diffuser was tested with tandem deswirl vanes and without VIGV. No appropriate base without tandem deswirl vanes was available for this data. The data show increased efficiency with increased setting angle. This would be expected as the pressure-ratio characteristic shifts to higher flows with decreasing setting angle. The compressor surged at 85% speed at Match T8 with the nominal setting angle.

Figure 12 presents results of the 28-channel diffuser ($+1^\circ$ setting, tapered trailing edge) from engine data with a turbine nozzle with 3% increased flow capacity. With this change in operating line, the compressor flow is closer to match-point goal at 80% speed [0.887 vs. 0.900 lbs/sec (goal)]. These results show the compressor efficiency slightly less than program goals at 50% and 60% speeds. At 80% speed, the compressor is 1.9 points from goal and 1.0 point from rig results. With VIGV loss included, the compressor is 2.6 points from goal and 1.7 points from rig results, at 80% speed. Figure 13 shows the efficiency loss due to the VIGV at zero degrees flap angle. This is much more than expected from preliminary estimates and may be due to poor manufacturing quality. Most of the flaps do not line up with the stators, and there are a number of discrepancies from the blueprint.

Several courses of action are in progress and are summarized on Figure 14. Active effort is in progress to achieve lower running clearance with abradable material on the compressor shroud. Procurement of new VIGV housings is in progress. These new housings are being fabricated out of plastic and are expected to meet print specifications. To investigate differences in rig and engine results, an integral rotor is being procured for test and an axial inlet is being fabricated to replace the vehicle intake system. In addition, intrastage static pressure measurements have been made to track the diffuser recovery. Preliminary results have shown discrepancies between rig and engine data in the diffuser bend. The results are currently being checked out. These data also indicate about 0.5 point decrease in impeller efficiency from rig results. In the area of design change, a new impeller has been designed by NASA. This rotor has 18 full blades and 18 splitters. It is hoped that reduced loading in the splitter area will improve rotor efficiency.

Compressor Turbine

During this quarter, the increased vorticity vortex with standard nozzle was tested, a new nozzle design was tested, and efforts were made to obtain a better estimate of the compressor turbine efficiency and the causes of efficiency deficit. In the Sixteenth Quarterly, an estimate was made of the compressor turbine efficiency from a mean-radius vector diagram analysis with use of shroud static pressure measurements. The efficiency was estimated to be 0.768 versus the program goal of 0.843. In the last Quarterly Report, more detailed data were presented which showed the results of radial surveys of total pressure and angle at the nozzle inlet and exit and at the rotor exit. The nozzle inlet data showed a need to

re-camber the nozzle to obtain design incidence angles. Lampblack traces showed strong secondary flow patterns. Although no wake surveys were taken, it was decided to redesign the nozzle to better match the vortex chamber exit conditions and to reduce secondary flows. This design was procured and tested during this quarter. In addition, the survey results and shroud static pressure measurements were used in radial-equilibrium calculations to give a more accurate assessment of the nozzle and rotor losses.

- Nozzle Test Results and Analysis

Plots of the aerodynamic loading of the Mod. 1 nozzle were presented in the last Quarterly Report. Figure 15 summarizes the geometric and aerodynamic parameters. Figure 16 shows the vector diagrams at hub, mean and tip. Figure 17 shows the mass flow characteristic measured for this nozzle. The data show an increase in maximum flow at high total-static pressure ratios. It is not certain whether this is a true aerodynamic result or a matter of experimental uncertainty. The mass flow characteristic used for engine testing is shown by the curve. Based on the curve, the maximum flow is 8% higher than required. To achieve required maximum flow, the nozzle trailing edge was bent as shown on Figure 18. The calibration results are shown on Figure 19. Engine tests were conducted with the nozzle before and after vane bending. Lampblack traces on the Mod. 1 nozzle, prior to bending, showed less secondary flow accumulation. Figure 20 shows a comparison of Lampblack traces between the original and the Mod. 1 designs.

In the meantime, procurement is in process for a second nozzle which has an increase of 2° in exit vane angle. The miss in required mass flow could be due to lower loss and/or blockage than estimated in the design calculations or due to larger deviation angle than originally estimated. Test results with increased vorticity from the vortex chamber suggest that deviation angle was underestimated (see below). The vane was therefore re-cambered to achieve the intended maximum mass flow. This revision is identified as Mod. 1.1. A comparison with the Mod. 1 design is shown in Figure 21. The design parameters are summarized in Figure 22 and the vector diagrams are shown in Figure 23.

The coordinates for the NASA revised nozzle design (Mod. 2) were received from NASA during this period. Drafting began and was completed for cast and machined versions of this design. The hub and tip profiles are shown in Figure 24A; the original design is shown in Figure 24B for comparison. The design parameters are summarized in Figure 25. The design vector diagrams are shown on Figure 26, and the aerodynamic loadings are shown in Figure 27.

As reported in the last Quarterly, another approach to better nozzle performance might be to reduce the incidence angle. This would be beneficial if, for instance, the vane sections near the endwalls have missed the design exit air angle because of deficiency in turning angle due to partial stall or separation. The change in incidence angle was accomplished by increasing the vorticity of the vortex chamber by 20%. This would reduce incidence angle by about 5 degrees. Two sets of comparative engine tests were run with the increased vorticity and with the standard vortex. Power output with the increased vorticity improved 2% in one case and 5% in the other. This would indicate that no significant triggering of flow improvement through the nozzle and wheel took place. Flow tests of the increased vorticity vortex showed 5-8 degrees change in air angle at mid-span with smaller values in the upper half of the span and larger values in the lower half. Within the data scatter, there was not much change in the inlet total-pressure loss distribution. The exit data showed little change in exit air angle despite the large reduction in incidence angle, even near the endwalls. Consequently, the endwall blade-sections were not originally at incidence angles indicative of maximum turning or stall. This is confirmed in Figure 27 in plots of turning angle versus incidence angle. Plots are presented at every 10% vane height from the hub. The values of turning angle increase virtually linearly with incidence angle. This indicates nearly constant deviation angle and, hence, no indication of stall at the higher values of incidence angle.

On the other hand, there is a lack of turning angle indicated near the endwalls, which suggests larger values of deviation angle than expected. The dashed lines on Figure 28 indicate the turning-angle values expected at the hub and tip based on the change in camber angle relative to mid-span. The lower 30% of the span is well below the expected turning. In addition, turning-angle values in the upper half of the vane fail to achieve values higher than those at mid-span. The section at 90% height is significantly below expected turning.

One interpretation of these results is that the endwalls strongly affect local turning as well as total-pressure loss. This can be shown more clearly by plotting the difference between measured deviation angles and those values predicted from Carter's Rule empirical correlation. Figure 29 shows such a plot for the Upgraded Engine nozzle in comparison with a Chrysler experimental design (A-907). From 30% to 70% span, the Upgraded Engine nozzle shows about 1.5 degrees of overturning compared to Carter's Rule. In the same range, the A-907 nozzle shows about 3 degrees of underturning. Figure 29 shows a dimensionless plot of deviation-angle difference based on the value of local camber. On this dimensionless basis, the two sets of data converge below 30% span. Above 70% span, the two sets of data converge at 90% span.

The dimensionless plot of Figure 29 shows the difference from the empirical estimate of deviation angle expressed as a percentage of local camber. It suggests that, on these small nozzles, corrections should be made for the lack of turning near endwalls. The dashed lines in Figure 29 suggest such a correction, as well as a correction to Carter's Rule for the amount of uncovered trailing edge turning. The mid-span value for the Upgraded Engine nozzle is 9 degrees and for the A-907 nozzle is 15 degrees.

- Engine Test Results and Analysis

Efficiency estimates for the compressor turbine were deduced from intrastage static pressures measured on engine test, in combination with measured values of compressor work and measured values of compressor and turbine mass flows. Tip static pressures were measured at the nozzle and rotor exits. Radial equilibrium calculations were performed at these stations with assumed distributions of loss. The loss distributions were based on estimates of profile, endwall and clearance losses for the nozzle and the rotor. The loss distributions and blockage estimates were adjusted until a match was obtained between calculated and measured values of static pressures at the required work and flow. The required work was computed from compressor temperature rise and mass flow and from parasitic losses taken from cycle estimates. The rotor exit relative air angle was estimated from Carter's Rule. Survey results were used for the nozzle exit air angle distribution, for the original nozzle, and estimated values were used for the Mod. 1 nozzle.

In general, the procedure was to assume values for nozzle blockage and nozzle and rotor loss until the required work was obtained. If the static pressures did not match, then nozzle blockage was continually revised, along with loss levels, until a match was achieved. The final result gave an estimate of the stage efficiency, along with the loss breakdown between nozzle and rotor. These results were also used as input for blade-to-blade loading calculations in the NASA Computer Program TSONIC. By this means, the efficiency was estimated, the breakdown between nozzle and rotor determined, and a possible cause of loss investigated through loading calculations.

Calculations were performed with engine test data for the original nozzle design, the Mod. 1 nozzle and the bent Mod. 1 nozzle. The resultant efficiency distributions together with a single value based on mass averaged work and pressure ratio are shown on Figure 30 for each case. These are shown to be in fair agreement with mean-radius values of efficiency computed from measured total pressure and the turbine work. The original design has an

efficiency of 0.77, which agrees with the estimate from the mean-radius vector diagram analysis. Reducing rotor incidence angle through nozzle trailing edge cutback further lowered efficiency to 0.73. The revised nozzle design increased stage efficiency to 0.80, but this was reduced to 0.76 with the bent nozzle. As a partial check on the validity of the analysis, the figure shows a comparison between radial variation in pressure ratio obtained from survey results and values deduced from the analysis. The agreement is good.

Overall, these calculations indicate that compressor turbine stage efficiency is 7 points below the program goal. With the new nozzle design indicating an efficiency increase of only 3 points, the possibility exists that the rotor may be low in efficiency by at least 4 points. To gain insight into this, design point loadings of the wheel were rechecked with a recent version of TSONIC. The results are shown in Figure 31. In general, the design point loadings show suction-surface diffusion near the trailing edge with values of diffusion ratio and diffusion gradient that exceed established limits (Reference 1).

Additional loading calculations were performed at 80% speed with input obtained from the radial equilibrium calculations discussed above. The rotor loadings with the original nozzle are shown in Figure 32. On the rear half of the suction surface, values of diffusion ratio and diffusion gradient exceed established limits from hub to tip. The rotor loadings with the Mod. 1 nozzle are shown in Figure 33. Again, the suction surface diffusion parameters exceed limits. The analytical results show that the rotor efficiency increased with this nozzle. This may be due to the increase in reaction across the rotor, which may yield more favorable boundary layer parameters prior to suction surface diffusion. The injection of chrome oxide into the engine showed less suction surface deposit on the rotor when tested with the Mod. 1 nozzle. This may be a result of loss reduction through increased reaction.

It is difficult to consider a revised design with increased reaction without encroaching on the ability to augment power with VIGV. However, in order to get a clearer insight into the turbine performance problem, an 0.8 x scale of the Baseline Engine turbine is in procurement. Comparison with the original rotor is shown in Figure 34. The rotor leading edge angles differ by 5 degrees at the hub, 4 degrees at the mean, and 7 degrees at the tip, between the two rotors. This rotor has slightly higher reaction and is being

Reference 1: THE DESIGN AND PERFORMANCE ANALYSIS OF AXIAL-FLOW TURBINES. VOLUME I - THEORY AND PRACTICE OF DESIGN REPORT NO. 1174-1. Northern Engineering and Research Corporation.

reviewed analytically for testing with the original nozzle and Mod. 1 nozzle. In a second course of action, an effort has started on a revised rotor design.

Power Turbine

An analysis in the Sixteenth Quarterly Report indicated that with the variable nozzle power turbine stage at design conditions, design efficiency would be achieved. An attempt was made to experimentally verify this conclusion by conducting a test in which power turbine pressure ratio was increased almost to the program goal value at 80% speed. This was accomplished by injecting high pressure air downstream of the compressor. The results are shown in Figure 35.

Extrapolation to the program goal pressure ratio and referencing to cycle inlet temperature shows the power turbine misses the goal by 20%. Though its performance could be effected by wakes and blockage in the first stage discharge flow, the discrepancy is of sufficient magnitude to warrant concern.

Consequently, to establish the efficiency level of the power turbine and also indicate the importance of compressor turbine exit conditions on power turbine efficiency, priority efforts have been initiated on the design and building of a power turbine rig.

Component Test Rigs

Three component rigs are currently under consideration: (1) a non-regenerative engine rig; (2) a compressor turbine nozzle flow rig; and (3) a power turbine stage rig.

- Non-Regenerative Engine Rig

The major differences between rig and engine compressor configurations are the air inlet system, the exit collector and the split impeller. To better understand the influences of those differences, especially the exit collector, a non-regenerative engine has been proposed for full-scale compressor testing, as shown in Figure 36. This rig would allow evaluation of collector volume on full-scale engine hardware through modifications not possible on the engine.

This rig also allows for the installation of instrumentation in the turbine section for precise evaluation of the compressor turbine efficiency. The engine housing does not permit satisfactory probing to accurately measure turbine efficiency. The

rig provides the opportunity to take turbine inlet and exit surveys to measure ideal work while the compressor and bearing losses provide actual work.

- Compressor Turbine Nozzle Flow Rig

Data taken to date at the turbine nozzle exit has been limited to cobra probe surveys between vane wakes. To get an assessment of nozzle loss, it is necessary to take wake surveys. Test activity priorities on the present flow rig have not permitted wake survey testing. It has been proposed that a second flow rig be built specifically for detailed measurements, while the present rig is used for nozzle calibrations for engine testing and limited survey tests. The proposed flow rig is shown in Figure 37.

- Power Turbine Stage Rig

As discussed in the preceeding section, work on a power turbine rig is receiving first priority. The concept sketch is shown in Figure 38. A bellmouth inlet is used together with swirl vanes to establish uniform inlet conditions. The purpose of initial testing will be to determine whether or not the power turbine can perform with the expected operating line efficiency levels under ideal inlet conditions. The testing will be conducted under actual conditions of power turbine inlet pressures and temperatures.

Combustor

During recent testing of both cell engines and the car engine, the combustor has been plagued by flash-back at operating speeds above 80% Ngg. Earlier running had been accomplished to 100% Ngg without flashback. No explanation is available for this apparent deterioration of premixing ability. The result of flashback operation is not only high NOx levels, but also higher burner tube metal temperatures, which therefore have prevented operation above 90% Ngg on most builds. Modifications to the premixer and injector are being evaluated in an effort to correct this problem.

Quotes have been requested for ceramic burner tubes from four vendors. These should improve the ability to provide full speed range operation at low emissions and should also be able to tolerate the reaction temperature resulting from premixer flashback.

An extension housing and tube have been fabricated to evaluate burner outlet distribution. The housing is being insulated and the tube extension is being instrumented with thermocouples.

The torch assembly has also shown a need for further refinement. Occasional carbon formation in the discharge tube may indicate a need for more airflow (leaner operation) or other flow modification. Soft carbon also forms after shutdown and sometimes shorts the spark plug, preventing restarts. Improved fuel purging is necessary.

Regenerator Flow Distribution

Tests will be conducted at Creare, Inc. to evaluate the flow distribution on the LP side of the regenerator on a scale model test rig. Since the performance of the turbine and the flow distribution into the regenerator is dependent on the performance of the exhaust diffuser/collector, a detailed map of static pressure recovery for the exhaust diffuser will be obtained. The turbulence generated by a turbine rotor will be simulated with use of a free runner which has vane solidity and thickness based on actual rotor geometry. Data will be obtained with six values of exit swirl ranging from 0° to 50° . Work has begun on the design and procurement of test rig hardware.

An engine housing is being adapted for cold flow development of high pressure regenerator uniformity. This work is proceeding somewhat slowly due to deferral of priority to engine power improvement efforts.

Regenerator Development

A recurrence of the previously reported graphite seal wear problem has demonstrated the need for improved matrix rim surface finish. Rim roughness, due to excessive rim filler material, can be safely removed only by careful hand lapping. Excessive pressure causes matrix fracture and worsens the problem. The vendor has agreed to change his process so that the finish grinding operation follows the application of rim filler. Cores #14 and #15 incorporate this change, and cores #1-#13 will be reworked at no charge. They will be returned as work load permits.

Regenerator Wall Matrix Leakage

Corning Glass requested information on our experience with cross-passage matrix leakage. Another turbine manufacturer had reportedly blamed recent starting problems on the increased porosity of the new thinwall matrix. The requested test was run on core #6. The test showed that a one square inch area of matrix, one inch thick, at a ΔP

of 1 psi, would flow .095 pounds per hour at room temperature. Previously reported, cold leakage checks in a sealed engine assembly have shown internal matrix leakage of less than 1% of engine airflow.

Regenerator Crossarm Seal Development

An acceptable nickel oxide seal on an 18-SR substrate has not yet been produced. Problems include excessive bowing (both longitudinal and transverse), and coating loss due to corrosion between the bond coat and the substrate. However, a zirconia-coated 18-SR seal was successfully prepared, and is now awaiting the next available engine test opportunity. Another 18-SR substrate is available for a second zirconia seal, and nickel oxide will be sprayed on two X-750 substrates as a backup. A blank of each material has been given to an outside source, who hope to demonstrate the feasibility of using electro-chemical grinding to produce flat, stress-free substrates.

Regenerator Cover Temperatures

As reported in the Sixteenth Quarterly, initial observations of a gap appearing between the regenerator cover and the center bolt during engine operation indicated significant deflections occurring in this area.

In order to establish the cause of this gap, parallel efforts were directed to examine thermal gradients and deflections of both the regenerator cover and the engine housing bulkhead/center bolt area during engine operation. Thermocouples were applied both on the bulkhead of the engine housing and on the regenerator cover. Measurements of bulkhead deflections were reported in the Sixteenth Quarterly. Temperature measurements made on the cover and adjacent areas of the engine housing are shown in Figure 39. The temperature levels are generally as expected. However, those shown in the section view indicating the gradient through the cover thickness would have a prime influence on cover deflection and these appear questionable, i.e., the relatively low temperature of 386°F on top of the rib between the two higher adjacent readings of 455°F and 470/475°F should be verified prior to any further analysis.

Engine Housing Metal Temperatures

Data from thermocouples placed on the bulkhead and adjacent areas of the housing of Power Plant 3-1T has been examined. The measured temperatures were compared with the estimated temperatures used in the finite element model for stress and deflection analysis. Both the measured temperatures at 100% gas generator speed and selected estimated temperatures are given on Figures 40 and 41 which computer plots of the model, with portions of the housing shown for clarification.

The measured temperatures are, in general, significantly lower than the temperatures estimated from Baseline data; however, the following should be noted:

- . Turbine exhaust temperature (%7.5) was set within 23⁰F of design, but due to the problem of low engine power, turbine inlet temperature (T5) indicated 170⁰F below design.
- . The use of compressor discharge air ducted to the bulkhead for cooling along with a well insulated power turbine nozzle support appears to be satisfactory. Metal temperature levels indicate some margin for adjustment will be available even with engine gas temperatures (T5 and T6) increased to their design values.
- . Cooling air is excessive in most sections of the bulkhead. The cooling air should be reduced (if this is not detrimental to compressor performance) and/or redirected to reduce thermal gradients by lowering temperatures around the center of the crossarm.
- . Radial bulkhead deflections are believed to be a factor in causing a gap between the regenerator cover and its center bolt. It appears that the pertinent gradients measured are similar to those estimated for the stress/deflection model which indicated deflections contributing to the gap in question.

The one measured temperature (523⁰F) indicates a large (220⁰F) thermal gradient existing along the length of the crossarm would tend to cause it to bow forward, changing the stress and deflection picture significantly. The extent of the gradient should be verified and other intermediate temperatures determined on the bulkhead.

Compressor Abradable Shroud

An abradable shroud was fabricated and tested this period without apparent damage to the impeller (2618-T61 aluminum). These initial tests were mechanically oriented, specifically to evaluate starting and low speed running. A fair performance evaluation could not be conducted, as the impeller was not to print specification.

A compressor cover was modified to accept a microsphere filled silicone rubber shroud. This shroud was molded in place using a plaster contour mold to define the flow surface.

Significant contour variations using this technique resulted.

The shroud was reworked for a "just touch" condition over 95% of the face.

Initial cranking tests did not produce any significant contact areas. However, inspection following .5 hour at 50% speed produced a significant uniform abrade condition on the complete radial face area of the shroud. Startup and coastdown conditions appeared normal. Observations of the contact area and debris are as follows:

- . The abradable material has a torn surface texture (see Figure 42).
- . Visually the impeller showed no significant wear; however, the face of the blades were coated with a black gummy residue.
- . The abraded particle sizes ranged from very small (dust) to large and resembled eraser debris in the larger sizes.
- . The particles were randomly distributed throughout the gas generator with some trapped on the leading edges of the diffuser and deswirler vanes.
- . A small deposit of particles was trapped in the air bearing feed hole (air bearing supplied externally for these tests).
- . Deposits were found in the small cooling duct of the housing feeding the bulkhead.

Subsequent running up to 80% produced no significant additional material removal. Shroud contact was very light and randomly distributed. No debris was evident in the gas generator or

housing. (A complete engine teardown was not made.) However, a large deposit of shroud material was later found at the cover center bolt bushing beneath the regenerator L-seals. The impeller exhibited a local polish at the contact areas.

Future development will include:

- . Improve tooling and molding techniques for better contour control.
- . Investigate other materials and the long term effect on potential impeller wear.
- . Investigate other materials which produce debris that will not contaminate or impair component/part life and/or performance.

Gas Bearing Development

Phase 2 work funded for the extended program on the bearing rotor system development has been completed with the exception of fabrication and delivery of five sets of front oil bearing/damper assemblies. This work included shock and vibration tests. Oil inlet temperature for all tests was approximately 93°C (200°F). At the idle condition, 50% speed, a shock of approximately 4 "g's" was applied. At 90% speed, a shock load of 16 "g's" was applied which was the largest shock simulated. The 16 "g" load was used to simulate a gyroscopic slew loading on the bearing. The rotor operated normally after each shock load was applied. A total of 25 drops were recorded varying from 2 "g's" to 16 "g's" through the speed range.

The unbalance level was then changed from .003 to .006 oz. in. in-phase and the rotor operated to full speed. Air bearing temperature was approximately 93°C (200°F). Both rotor orbits were just slightly larger than those recorded previous to the change in the unbalance level. The unbalance level was then increased to .010 oz. in. in-phase and the rotor run to full speed. Rotor orbits were not significantly different than the previous tests and appeared to contain less sub-synchronous frequency components. During this test, the simulator heaters were turned on and maximum temperatures recorded were: 215°C (420°F) at the air bearing; 207°C (405°F) air bearing cooling air; and 218°C (425°F) on the rear tapered housing.

The simulator was disassembled for inspection. All three floating carbon seals showed indications of light rubs, none appeared serious; the front oil bearing/damper assembly was in a "like-new"

condition; and the air bearing showed very little additional dry film wear. The air bearing which was used throughout this test program is the same bearing used in tests conducted in June and July of 1976. This bearing now has 72 hours of running time of which approximately 30 hours were at full speed and temperature and has undergone approximately 150-175 starts.

The tasks in Phase 3 through Phase 5, which are necessary for completion of the program, have been defined and submitted to ERDA for approval. This program includes: additional damper studies to optimize damper clearance and establish tolerances, work with elastomer spring arrangements, testing with various air bearing clearances to determine tolerance range, evaluation of new air bearing lubricants, procurement of unassembled floating carbon seals for possible later studies on the influence of the carbon seals on the dynamic system, and provides for contingency studies which may be required as development of the engine by Chrysler continues.

- Engine Testing

Gas generator rear air bearing no. 5 and bearing sleeve no. 4 were submitted to MTI for inspection and a letter report outlined below was sent to Chrysler. The bearing had been operated in engines for 14.7 hours and was removed when the coating had worn significantly. In addition, it was noted that the shim used to set bearing clearance had worked itself out of position and projected .18 inch beyond the rear edge of the bearing. This bearing was one of the first five assembled, which did not have the shim attached to the bearing cartridge. The shim was spot welded in place on subsequent bearings.

The mechanical condition of the load-bearing components of the bearing was good. No deformation of the bump or smooth foils was evident and the bearing clearance had not changed.

The dry film lubricant over the rear 1/4 inch of the bearing had worn away and the foil also showed up to .0003 inch of wear in this area. The lubricant on the forward part of the bearing was intact although it was thin in places. It is believed this coating wear is occurring during the start cycle since contact at high speed would cause a seizure.

Inspection of the chrome carbide journal surface showed it to be very poor (25 μ -in.). The material was porous with a large number of sharp-edged voids. Transfer of dry film lubricant and metal from the bearing to the journal was evident. The coating was also gouged where heavy rubbing had occurred resulting in a surface finish of 60 μ -in. A .0002 inch taper and cut-of-round condition

was also noted, which is within specifications. A means of obtaining the desired coating surface finish is being investigated.

Integrated Control System

The programmable units for the test cell systems continues to function satisfactorily. However, in one instance, the fuel flow sensor failed to operate at low cranking voltage which inadvertently resulted in excess fueling of the engine during starting. Though the engine was not damaged, the program software was rewritten to safeguard the engine against flow sensor signal loss and the faulty sensor replaced. The start accel schedule is still closed loop but now reverts to a scheduled open loop with loss of signal.

This safeguard has also been incorporated in the Car 386 software program.

Considerable effort was expended this period in documenting the response characteristics of the fuel loop for Car 386. Future tests will examine the benefits of open loop versus closed loop fuel flow along with T5 transient fuel limiting. In optimizing the present design, it appears that the software changes for any future improved turbine stage (improved rotor acceleration-faster system dynamics) will be relatively straight forward. A Chrysler proprietary fuel flow sensor with superior response has been bench tested and will be system evaluated when time permits.

Figures 43A and 43B show the engine control and vehicle indicator modules. Full time readout of corrected speed and temperature is provided along with sequence and function indicating lites (LEDs). A portable diagnostic module (Figure 43C) is available which is wired to the ECU's diagnostic system. If additional system parameters are required on site, a break-out box is available for both dynamic and static checkouts.

Technical systems support by Ultra continues as required to resolve system/component anomalies. A field test set has been loaned to Chrysler for static checkout and trouble-shooting of ECU units on hand.

NASA personnel were familiarized with the ECU system and diagnostic support equipment available. It is anticipated that Ultra will directly support the Upgraded Engine controls activity at NASA Lewis with Chrysler providing backup as required.

- Redesigned Nozzle Actuator

The basic unit with power-to-braking control is working satisfactorily. No fluid leakage through the unit is evident. Components for the modulating control are being machined.

TASK 8.2 BUILD UPGRADED VEHICLES

Car 130

The floor pan and suspension modifications have been completed. Work has been curtailed due to other priorities.

Car 385

This vehicle was used for the ASME show (see Task 3.1). Installation of a functional engine will be delayed until the engine power deficiency problem is resolved.

TASK 8.3 UPGRADED VEHICLE EVALUATION

Car 386

Initial running was devoted to documenting control/engine responses for this Upgraded system. Chassis rolls steady state tests indicated emissions were controlled at high speeds but the burner requires some developmental effort to resolve transient and low speed operation. The steady state emissions are shown below and are included for reference only. The cycle match was 50-60°F cold. Transient tests were terminated at 80% speed due to the limited service compressor turbine wheel.

<u>Mode</u>	<u>HC</u> <u>ppm</u>	<u>CO</u> <u>ppm</u>	<u>NOx</u> <u>ppm</u>
Idle	447	17	4
Idle Drive	399	20	4
20 mph	27	2	1
30 mph	27	.6	.7
40 mph	21	.5	.9
50 mph	21	.45	1.7

The gas generator was updated and included a tandem deswirler at the compressor discharge and a new compressor turbine wheel which

permitted full speed range operation. In addition, T5 vortex thermocouples were installed for control system development. The inlet bowl was insulated to reduce inlet noise. Efforts late in this period were expended in preparing the vehicle for limited demonstration at the ERDA Washington meeting. Considerable control and general vehicle work still remains to be done to achieve an acceptable car for ride and drive demonstrations.

Overall impressions of the vehicle as noted below are good:

- . Inlet and exhaust noise are minimal and less than the Baseline Vehicles both in the passenger compartment and in drive-by tests.
- . Vehicle response is strong once the gas generator is up to speed.
- . Figure 44 shows some preliminary transient rotor response data. The gas generator acceleration time is about 2.3 seconds, with improved efficiencies in the gas generator section. Acceleration temperatures under these conditions are at the maximum allowable level. Top speed is intentionally limited to 95% Ngg.

Problem areas to be resolved are:

- . Identify and minimize power turbine gearing noise.
- . Develop and install an IGV actuator to control engine output power (power turbine speed) and minimize creep torque.
- . Test and develop the power turbine nozzle actuator for complete engine/vehicle braking control.
- . De-sensitize the power brake system.
- . Improve transmission shift quality.
- . Continue to optimize control system parameters.
- . Improve torch plug life through improved fuel purging during shutdown.
- . Improve pedal feel.

- . Resolve burner growth during warmup and decelerations.
- . Document ECU underhood temperatures.

TASK 10 IMPROVED AUTOMOTIVE GAS TURBINE (IAGT) DESIGN STUDY

Efforts on this task were started in this quarter.

Superalloy Base Materials

Data on the best modern conventional superalloys, advanced directionally solidified eutectic alloys (in-situ composites), oxide dispersion strengthened alloys, and tungsten fiber reinforced superalloys are being consolidated, evaluated and plotted.

For turbine blades, "temperature capability" is frequently arbitrarily defined as the temperature to produce rupture at 138 MPa (20 ksi) in 1000 hours. On this basis, conventional superalloys cannot be expected to attain a temperature capability, even with continued development, of over 1010°C (1850°F). For lower stressed nozzle vane applications, using the temperature to produce rupture at 69 MPa (10 ksi) in 1000 hours as the criterion, the temperature capability of conventional superalloys may be extended to about 1090°C (2000°F). Further increases are being obtained by applying the directionality concept to eutectic systems and tungsten fiber superalloy composites. Directionally solidified eutectic superalloys offer the potential for an increase of 40°-90°C (100°-200°F) in turbine blade/vane capability. Tungsten fiber reinforced superalloys afford potentially the highest use temperature capability for turbine blades, i.e. 1200°C+ (2200°F) but their application is most remote. Oxide dispersion strengthened (ODS) oxidation resistance ferritic alloys appear to have a potential use temperature up to 1230°C (2250°F) as stator vanes (10 ksi).

All of the advanced alloy systems mentioned will probably require coatings at temperatures above 1040°C (1900°F), and in almost every case present serious cost and fabrication problems.

Properties of Refractory Alloys

A computerized literature search on refractory alloys has been ordered from A.S.M. to assist in establishing state-of-the-art properties for application to the design of the high temperature engine.

Some alloys of molybdenum and columbium, in particular, have mechanical properties at temperatures in excess of 1200°C, that make them attractive candidates for high temperature hardware in the projected engine.

Cycle Studies

The analysis of thermodynamic cycles with the potential for a minimum 20% gain in vehicle drive cycle fuel economy has started. Candidate engine arrangements are being compared at 74.6 kW (100 hp) output.

The initial studies are concentrating on regenerative, low pressure ratio engine arrangements at increased cycle temperatures, typically 1370°C (2500°F).

The potential fuel flow reduction of a scaled Upgraded Engine at the increased cycle temperature ranges from 15 to 20% for fixed regenerator inlet temperature at part load operation, as shown in Figure 45. An additional reduction of up to 10% could ideally be obtained for fixed turbine inlet temperature operation at part load. "Real" compressor operating range characteristics as well as the turbine stage excess temperature capability required during engine acceleration transients would limit this additional reduction.

A high temperature single-shaft engine arrangement utilizing a single radial turbine stage has fuel flows from 25 to 30% lower than those of the Upgraded Engine at constant regenerator inlet temperature operation, as shown in Figure 46. The additional reduction for increased part load temperature operation is also applicable; however, the acceleration temperature consideration of the single shaft engine, with potentially longer acceleration times, may impose a lower temperature limit schedule than would be applicable for the free turbine arrangement of the Upgraded Engine.

The improved efficiency of the single shaft engine arrangement is due to the increased efficiency of the turbine stage with lower tip clearance effects, the reduction and/or elimination of some of the heat and flow leak paths, and the reduction of the parasitic power requirement of the engine accessories and bearings. However, realizing this improved engine efficiency in a vehicle installation requires a transmission that can effectively expand the limited operating range of the engine as well as a control system that maximizes engine/vehicle response.

The output power characteristic of the single shaft arrangement is compared to that of the free turbine Upgraded Engine in Figure 47. The operating speed range of the free turbine arrangement is about

8:1, with at least 80% of maximum output power available over 75% of the speed range. The operating speed range of the single shaft arrangement is limited to about 2:1, with 80% maximum engine output power available over only 12% of the speed range. Additionally, at a given output power, the loss in engine efficiency for non-optimum speed operation is much more severe for the single shaft engine than for the free turbine arrangement.

Preliminary single shaft engine/vehicle response calculations indicate the importance of the apportionment of engine acceleration power between that used for engine acceleration and that for vehicle acceleration. These calculations assumed that the rotor would consist of an aluminum radial impeller and a ceramic radial turbine. The time required to accelerate the engine from idle up to maximum speed (i.e., power) ranges from about 1.3 seconds with all of the available power used to accelerate the engine, to 5 seconds with 25% of the available power used for engine acceleration and 75% for vehicle acceleration. In the former case, the vehicle would not move for 1.3 seconds, but then maximum power would be available for vehicle acceleration, within tire traction limitations. In the latter case, both the engine and vehicle accelerate at the same time, but slowly, since peak engine power would not be available for 5 seconds. Calculations of the power split for optimum overall vehicle acceleration requires characterization of a suitable transmission. As a first step in this effort, a previously developed computer program for analyzing various traction drive CVT (Continuously Variable Transmission) systems has been modified to accept single shaft gas turbine engine characteristics. The CVT used in this program is a regenerative (split) torque, single cavity, cone roller toroidal drive.

Typical steady state performance curves for a 1700 kg vehicle with a 1370°C (2500°F) single shaft gas turbine with constant regenerator inlet temperature are shown in Figure 48. Engine and transmission operating points are shown for road load and maximum power versus vehicle speed. The BSFC values from Figure 46 were used to calculate the road load fuel economy. CVT Hertz stresses were acceptable throughout the operating range.

FUTURE ACTIVITIES

In the next quarter, it is anticipated that:

- . The NASA Upgraded Engine will be instrumented and dynamometer checkout underway.
- . Performance tests on the MTI transmission pump/motor module will be complete and the final analysis report will be written.
- . Corrective development of the Upgraded Engine will be continuing.
- . The regenerator test fixture will be in operation.
- . Development of Upgraded Vehicle 386 will be continuing.
- . The Upgraded Engine compressor section will be operating near design goals.
- . An improvement will have been realized in the Upgraded Engine turbine stage efficiency.

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APPENDIX A

FIGURES

FIGURE

- 1 Expanded Upgraded Engine Display
- 2 NASA Instrumented Upgraded Engine
- 3 Upgraded Engine Status
- 4 Upgraded Engine Power Turbine Airfoil Inspection in Optical Comparitor
- 5 Digital Electronic Inspection of a Reduction Gear Housing
- 6 Best Fit Error Values
- 7 Upgraded Engine Performance Map with Variable Inlet Guide Vane Variation
- 8 Upgraded Engine BSFC vs. Output Horsepower
- 9 Upgraded Engine Compressor
- 10 14-Channel Diffuser-Trailing Edge Modification
- 11 Upgraded Engine Compressor, Efficiency Comparisons with Different Diffuser Configurations
- 12 Upgraded Engine Compressor, Summary of Efficiency Status
- 13 Upgraded Engine Compressor, Efficiency Loss Due to Variable Inlet Guide Vane Assembly
- 14 Upgraded Engine Summary of Compressor Efficiency Status at 80% Speed
- 15 Original Chrysler Nozzle Redesign (Mod 1.0)
- 16 Upgraded Engine Compressor Turbine Nozzle, Mod 1.0 Vector Diagrams
- 17 Upgraded Engine Compressor Turbine Nozzle Flow Test
- 18 Upgraded Engine Compressor Turbine, Mod 1.0 Nozzle Bent at Tip
- 19 Upgraded Engine Compressor Turbine Nozzle Flow Test
- 20A Upgraded Engine Compressor Turbine Nozzle, Lampblack Traces on Vanes
- 20B Upgraded Engine Compressor Turbine Nozzle, Mod 1 Lampblack Traces on Vanes
- 21 Upgraded Engine Compressor Turbine Nozzle, Chrysler Revised Design
- 22 Recambered Chrysler Nozzle Redesign (Mod 1.1)
- 23 Upgraded Engine Compressor Turbine Nozzle, Mod 1.1 Vector Diagrams
- 24A Upgraded Engine Compressor Turbine Nozzle, NASA Revised Design-Mod 2
- 24B Upgraded Engine Compressor Turbine Nozzle, Original Design
- 25 NASA Nozzle Redesign (Mod 2.0)
- 26 Upgraded Engine Compressor Turbine Nozzle, Mod 2.0 Vector Diagrams

LIST OF FIGURES - Continued

FIGURE

- 27A Mod 2 Stator Loading Diagrams, Hub
- 27B Mod 2 Stator Loading Diagrams, Mean
- 27C Mod 2 Stator Loading Diagrams, Tip
- 28 Upgraded Engine Compressor Turbine Nozzle, Turning Angle Characteristics
- 29 Compressor Turbine Nozzle Deviation Angle Difference from Modified Carter's Rule
- 30 Upgraded Engine Compressor Turbine Efficiency Distributions, Calculated versus Measured
- 31A Upgraded Engine Compressor Turbine Rotor Blade Loading Diagram at Design Point, Hub
- 31B Upgraded Engine Compressor Turbine Rotor Blade Loading Diagram at Design Point, Mean
- 31C Upgraded Engine Compressor Turbine Rotor Blade Loading Diagram at Design Point, Tip
- 32A Upgraded Engine Compressor Turbine Rotor Blade Loading Diagram, 80% Speed, Run 41, PP 2-3, Original Nozzle and Rotor, Hub
- 32B Upgraded Engine Compressor Turbine Rotor Blade Loading Diagram, 80% Speed, Run 41, PP 2-3, Original Nozzle and Rotor, Mean
- 32C Upgraded Engine Compressor Turbine Rotor Blade Loading Diagram, 80% Speed, Run 41, PP 2-3, Original Nozzle and Rotor, Tip
- 33A Upgraded Engine Compressor Turbine Rotor Blade Loading Distribution, 80% Speed, Run 128, PP 2-3, Mod 1 Nozzle, Bent; Original Rotor, Hub
- 33B Upgraded Engine Compressor Turbine Rotor Blade Loading Distribution, 80% Speed, Run 128, PP 2-3, Mod 1 Nozzle, Bent; Original Rotor, Mean
- 33C Upgraded Engine Compressor Turbine Rotor Blade Loading Distribution, 80% Speed, Run 128, PP 2-3, Mod 1 Nozzle, Bent; Original Rotor, Tip
- 34 Blade Profile & Pitch Comparison-Baseline vs. Upgraded Engine
- 35 Upgraded Engine Power Turbine Performance, 80% Gas Generator Speed
- 36 Non-Regenerative Engine Rig
- 37 Compressor Turbine Nozzle Flow Rig
- 38 Power Turbine Stage Rig
- 39 Regenerator Cover Temperatures

LIST OF FIGURES - Continued

FIGURE

- 40 Estimated and Measured Temperatures, Upgraded Engine Housing
- 41 Estimated and Measured Temperatures, Upgraded Engine Housing
- 42 Abradable Shroud
- 43 Engine Control Panel, Vehicle Indicator Module, Diagnostic Test Module
- 44 Preliminary Testing - Car 386
- 45 BSFC vs. Output Power, Upgraded Engine Cycle Comparisons
- 46 BSFC vs. Output Power
- 47 Engine Output Characteristic, No Power Augmentation
- 48 Gas Turbine Performance

FIGURE 1
EXPANDED UPGRADED ENGINE DISPLAY

FIGURE 2
NASA INSTRUMENTED UPGRADED ENGINE

FIGURE 3
UPGRADED ENGINE STATUS

4/27/77

ENGINE	1	2E	3	4C	5G	6N	7M		
P.P. No.:	3-1	4-2	2-3	5-4	6-5	7	8		
Completed	7/12/76	8/15	9/28	1/7/77	3/10/77	A	A		
Eng. Hsg. No.:	3	4	2	5	6	7	8		
P.T. Nozzle	1	2	3	4	5	6A			
Gas Gen. No.:	2	1	4	5	7G			3K	6I
CTW	108	111	110	117	101			115	112
CTN	108	101	117F	103	110			100	112
Impeller	2	5	4	1	10H			7D	6
Pwr. Turbine No.:	2	1	3	4	6				
PTW	202	210	200	212	201H				
Air Inlet	6	2	1	4	5				
VIGV	7L	8L	1L	4	5				
Oil/Air/Clutch	1	2	3	4	5				
Regen. Assembly:									
Core	12	9	5	7	6				
Cover	2	6	4B	3	9				
Combustor:									
Cover	5	3	2	1					
Tube	8	7	6	5					
NOTES:									
A - In process									
B - "L" seal									
C - NASA									
D - Reduced Diameter									
E - Car 386									
F - U-700, 3 Piece P/N 3814973									
G - Display									
H - Scrap									
I - Abradable Shroud Compressor Cover									
J - Waiting new CTW									
K - Alternate with GG2 And GG4									
L - Three Strut									
M - Chrysler Regenerator Flow Fixture									
N - NASA Regenerator Flow Fixture									

FIGURE 4

UPGRADED ENGINE POWER TURBINE AIRFOIL INSPECTION IN
OPTICAL COMPARATOR

FIGURE 5
DIGITAL ELECTRONIC INSPECTION OF A
REDUCTION GEAR HOUSING

BEST FIT ERROR VALUES

SURFACE NUMBER 1

INPUT POSITION TOLERANCE= 0.2000 INPUT ANGLE TOL(DEGS.)= 0.0000
XSHIFT= -0.1040 YSHIFT= -0.0391 ROTATION(DEGS.)= 0.0000 (CCW)

PN 3813810 SHROUD
MANY MORE COORDINATES ADDED TO IMPROVE CURVE
DSM MAY 6, 1977
CTN 3813812-103 SHROUD CONTOUR

XMEAS.	YMEAS.	ERROR	DANG
20.7181	74.0665	0.7538	0.0000
20.8187	72.7369	0.0748	0.0000
20.4541	71.1519	-0.1829	0.4127
26.4080	69.7009	-0.1818	0.2043
27.0003	67.3101	-0.1606	0.4609
24.8153	64.7701	0.1876	-0.4505
20.5798	60.0711	-0.2320	0.1529
16.3003	57.3025	-0.0806	0.0135
11.7348	55.5483	-0.0532	0.0011
3.4329	55.0165	0.0489	0.0000
X1= 8.2800	DX= 0.6900	XLEN= 31.0000	
Y1= 54.5100	DY= 0.6900	YLEN= 26.0000

FIGURE 6

FIGURE 7

UPGRADED ENGINE PERFORMANCE MAP
WITH VARIABLE INLET GUIDE VANE
VARIATION

A-926 PP 3-1U

VIGV Angle 0° Δ 30° □
10° + 40° ◇
20° ○ 50° ▽

BSFC vs. OUTPUT HORSEPOWER

4-23-76 CHARACTERIZATION

UPGRADED ENGINE COMPRESSOR

EFFICIENCY COMPARISON WITH PROGRAM GOAL RIG DATA AND ENGINE DATA

ORIGINAL DIFFUSER DESIGN

- ENGINE STATIC-TOTAL EFF.
- ENGINE TOTAL-TOTAL EFF.
- PROGRAM GOALS STATIC-TOTAL EFF.
- RIG DATA STATIC-TOTAL EFF.

14-CHANNEL DIFFUSER-TRAILING EDGE MODIFICATION

MODIFICATION #1

MODIFICATION #2

28-CHANNEL DIFFUSER-TRAILING EDGE MODIFICATION

UPGRADED ENGINE COMPRESSOR

EFFICIENCY COMPARISONS WITH DIFFERENT DIFFUSER CONFIGURATIONS

FIGURE 11

UPGRADED ENGINE COMPRESSOR
SUMMARY OF EFFICIENCY STATUS

- — PROGRAM GOALS
- △ — △ RIG, 14 CHANNEL DIFFUSER, NO VIGV
- ▽ — ▽ ENGINE DATA, 28 CHANNEL DIFFUSER, NO VIGV
- PP 3-1 RUNS 322-336

UPGRADED ENGINE COMPRESSOR

EFFICIENCY LOSS DUE TO VARIABLE INLET GUIDE VANE ASSEMBLY

VIGV ANGLE = 0°

UPGRADED ENGINE

SUMMARY OF COMPRESSOR EFFICIENCY STATUS AT 80% SPEED

ORIGINAL CHRYSLER NOZZLE REDESIGN (MOD 1.0)

Radius	4.445	5.031	5.563
Axial Chord	1.727	1.727	1.727
Chord	2.921	2.921	2.921
Pitch	1.862	2.107	2.330
Axial Solidity	.927	.820	.741
Solidity	1.573	1.390	1.257
Diagram Loading	.502	.439	.627
Total Diffusion	.127	.198	.132
Reaction	.843	.803	.800
Aspect Ratio	.383	.383	.383
Blade Number	15	15	15
L.E. Radius	.0725	.0725	.0725
Inlet Blade Angle	38.0°	38.0°	38.0°
Incidence	-2°	-10°	-2°
T.E. Radius	.0241	.0241	.0241
Exit Blade Angle	66.5°	66.5°	66.5°
Unguided Turning	13.0°	14.9°	16.7°
Throat	.835	.972	1.103
Camber Angle	28.5°	28.5°	28.5°
Exit Flow Angle	64.0°	63.84°	63.69°
Stagger Angle	53.35°	53.35°	53.35°
Deviation	2.5°	2.66°	2.81°
Choke Margin	8.3%	10.5%	17.3%

Dimensions in cm.

COMPRESSOR TURBINE NOZZLE

MOD 1.0 VECTOR DIAGRAMS

TIP
r = 5.563 cm

REL.M = .371
 $\beta_1 = 27.1^\circ$

ABS.M = .318
 $\alpha_1 = 40.0^\circ$

ABS.M = .748
 $\alpha_2 = 63.8^\circ$

U = 340.8 m/sec.

MEAN
r = 5.030 cm

REL.M = .520
 $\beta_2 = 40.0^\circ$

ABS.M = .366
 $\alpha_1 = 49.0^\circ$

ABS.M = .890
 $\alpha_2 = 63.4^\circ$

U = 308.1 m/sec

HUB
r = 4.445 cm

REL.M = .608
 $\beta_2 = 47.8^\circ$

ABS.M = .368
 $\alpha_1 = 40.0^\circ$

ABS.M = .953
 $\alpha_1 = 64.6^\circ$

U = 272.3 m/sec.

FIGURE 16

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UPGRADED ENGINE

COMPRESSOR TURBINE NOZZLE FLOW TEST

MOD 1 NOZZLE S/N 117

VORTEX S/N 102

UPGRADED ENGINE

COMPRESSOR TURBINE NOZZLE FLOW TEST

MOD 1 NOZZLE S/N 117

VORTEX S/N 102

FIGURE 20A

UPGRADED ENGINE COMPRESSOR TURBINE NOZZLE
LAMPBLACK TRACES ON VANES

FIGURE 20B

UPGRADED ENGINE COMPRESSOR TURBINE NOZZLE, MOD. 1
LAMPBLACK TRACES ON VANES

UPGRADED ENGINE
COMPRESSOR TURBINE NOZZLE
CHRYSLER REVISED DESIGN

RECAMBERED CHRYSLER NOZZLE REDESIGN (MOD 1.1)

Radius	4.445	5.031	5.563
Axial Chord	1.682	1.682	1.682
Chord	2.921	2.921	2.921
Pitch	1.862	2.107	2.330
Axial Solidity	.903	.798	.722
Solidity	1.569	1.386	1.254
Diagram Loading	.515	.451	.644
Reaction	.815	.801	.784
Aspect Ratio	.383	.383	.383
Blade Number	15	15	15
L.E. Radius	.0725	.0725	.0725
Inlet Blade Angle	38.0°	38.0°	38.0°
Incidence	-2°	-10°	-2°
T.E. Radius	.0241	.0241	.0241
Exit Blade Angle	68.5°	68.5°	68.5°
Unguided Turning	13.2°	15.4°	17.7°
Throat	.791	.926	1.055
Camber Angle	30.5°	30.5°	30.5°
Exit Flow Angle	50.38°	62.09°	57.66°
Stagger Angle	54.42°	54.42°	54.42°
Deviation	18.12°	6.41°	10.84°
Choke Margin	2.8%	3.7%	12.2%

Dimensions in cm.

FIGURE 22

LJP
4/26/77

UPGRADED ENGINE

COMPRESSOR TURBINE NOZZLE

MOD 1.1 VECTOR DIAGRAMS

FIGURE 23
LJP

UPGRADED ENGINE
COMPRESSOR TURBINE NOZZLE
NASA REVISED DESIGN - MOD 2

HUB

TIP

3.4 X SIZE

FIGURE 24A

UPGRADED ENGINE
COMPRESSOR TURBINE NOZZLE

ORIGINAL DESIGN

NASA NOZZLE REDESIGN (MOD 2.0)

Radius	4.445	5.004	5.563
Axial Chord	1.400	1.5275	1.655
Chord	2.5194	2.7475	2.9754
Pitch	1.862	2.096	2.330
Axial Solidity	.753	.729	.710
Solidity	1.345	1.303	1.270
Diagram Loading	.628	.570	.733
Total Diffusion	.206	.234	.351
Reaction	.862	.832	.847
Aspect Ratio		.407	
Blade Number		15	
L.E. Radius	.0500	.0500	.0500
Inlet Blade Angle	42.0°	39.0°	36.0°
Incidence	+3.1°	-8.5°	+3.1°
T.E. Radius	.0191	.0191	.0191
Exit Blade Angle	65.0°	65.1°	65.0°
Unguided Turning	7.0°	7.0°	7.0°
Throat	.757	.844	.943
Flow Turning	27.2°	18.6°	33.2°
Exit Angle	66.1°	66.1°	66.1°

Dimensions in cm.

UPGRADED ENGINE

COMPRESSOR TURBINE NOZZLE

MOD 2.0 VECTOR DIAGRAMS

MOD 2 STATOR LOADING DIAGRAMS

MOD 2 STATOR LOADING DIAGRAMS

MOD 2 STATOR LOADING DIAGRAMS

UPGRADED ENGINE

COMPRESSOR TURBINE NOZZLE

TURNING ANGLE CHARACTERISTICS

MEAN CAMBER = 18.5°

% HEIGHT FROM HUB

○ 10	△ 60
□ 20	◻ 70
◇ 30	▽ 80
△ 40	◀ 90
▽ 50	

COMPRESSOR TURBINE NOZZLE
DEVIATION ANGLE DIFFERENCE FROM MODIFIED CARTER'S RULE

FIGURE 29

UPGRADED ENGINE

COMPRESSOR TURBINE EFFICIENCY DISTRIBUTIONS

CALCULATED VERSUS MEASURED

80% GAS GENERATOR SPEED

UPGRADED ENGINE
COMPRESSOR TURBINE ROTOR
BLADE LOADING DIAGRAM
AT DESIGN POINT

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4/28/77

UPGRADED ENGINE
COMPRESSOR TURBINE ROTOR
BLADE LOADING DIAGRAM
AT DESIGN POINT

MEAN

UPGRADED ENGINE

COMPRESSOR TURBINE ROTOR

BLADE LOADING DIAGRAM

AT DESIGN POINT

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4/28/77

UPGRADED ENGINE

COMPRESSOR TURBINE ROTOR

BLADE LOADING DIAGRAM

80% SPEED, RUN 41, PP 2-3

ORIGINAL NOZZLE AND ROTOR

UPGRADED ENGINE

COMPRESSOR TURBINE ROTOR

BLADE LOADING DIAGRAM

80% SPEED, RUN 41, PP 2-3

ORIGINAL NOZZLE AND ROTOR

UPGRADED ENGINE

COMPRESSOR TURBINE ROTOR

BLADE LOADING DIAGRAM

80% SPEED, RUN 41, PP 2-3

ORIGINAL NOZZLE AND ROTOR

UPGRADED ENGINE
COMPRESSOR TURBINE ROTOR
BLADE LOADING DISTRIBUTION
80% SPEED, RUN 128, PP 2-3
MOD 1 NOZZLE, BENT; ORIGINAL ROTOR

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5/10/77

BLADE PROFILE & PITCH COMPARISON — BASELINE vs. UPGRADED ENGINE

A128-1 = - - - -
 UPGRADED ENGINE = ————

80

.8x SCALE OF BASELINE
 ENGINE TURBINE

FIGURE 34

UPGRADED ENGINE

UPGRADED ENGINE

COMPRESSOR TURBINE ROTOR

BLADE LOADING DISTRIBUTION

80% SPEED, RUN 128, PP 2-3

MOD 1 NOZZLE, BENT; ORIGINAL ROTOR

LJP
5/10/77

FIGURE 33C

UPGRADED ENGINE
POWER TURBINE PERFORMANCE
80% GAS GENERATOR SPEED

PP 3-1

△ RUNS 93-98 avg. P_{t6}

○ RUNS 135, 140, 151-153 avg. P_{t6}

A hand-drawn technical sketch of a mechanical assembly, likely a pump or engine component. The drawing shows a cross-section of the assembly, revealing internal parts and the external housing. Key features and labels include:

- Top Section:** A rectangular component labeled "Pleum" is shown at the top, with a section line "A-A" indicated above it.
- Insulation:** The area between the top and bottom components is labeled "Insulation".
- Internal Components:** The central part of the assembly shows a complex internal structure, possibly a pump or engine, with various mechanical parts and a central shaft.
- Bottom Section:** A rectangular component labeled "Pleum" is shown at the bottom, with a section line "A-A" indicated below it.
- Labels and Dimensions:**
 - "Simms A 216" is written on the left side of the central shaft.
 - "Simms A 216 Pump Teste Alloy" is written on the right side of the central shaft.
 - "3013750" is written below the right-side label.
 - "Simms Pump Teste to make" is written on the left side of the central shaft.
 - "New" is written near the bottom left corner.

COMPRESSOR TURBINE NOZZLE FLOW RIG

NEW PARTS: NOZZLE, PROBE, GEAR DRIVE,
SUPPORT, LINKAGE / ACTUATOR
(A 926 REGENERATOR RING GEAR
MAY BE USED)

USE EXISTING TEST RIG 9310-0015

PROPOSAL -
COMPRESSOR TURBINE NOZZLE
COLD FLOW TEST RIG
ROOM 8-11
APRIL 26, 77 PLF

ESTIMATED AND MEASURED TEMPERATURES

UPGRADED ENGINE HOUSING

PARTIAL FINITE ELEMENT MODEL OF HOUSING - REAR BULKHEAD SECTION VIEWED FROM FRONT

(---)=MEASURED TEMPERATURES, PP 3-1T, 100% GAS GENERATOR SPEED

----- ESTIMATED TEMPERATURES USED FOR STRESS AND DEFLECTION ANALYSIS

PP 3-IT 1-28-77 DATA
100% GAS GENERATOR SPEED

X = INNER SURFACE

△ ENGINE HOUSING FLANGE

ESTIMATED AND MEASURED TEMPERATURES
UPGRADED ENGINE HOUSING
PARTIAL FINITE ELEMENT MODEL OF HOUSING
FRONT BULKHEAD (BURNER WALL) VIEWED FROM FRONT - GAS GENERATOR SPEED 100%

(---) ← MEASURED TEMPERATURES PP 3-1T
--- ← ESTIMATED TEMPERATURES USED FOR STRESS
AND DEFLECTION ANALYSIS
(9-29-76 RUN)

100% NGG		
TEMP.	PP3-1T (1)	SPEC.
T ₁	71	85
T ₂	427	433
T ₃	1755	1925
T ₆	1450 (2)	1595
T _{7.5}	1355	1378
T ₉	611	548

(1) 1-28-77 Data
(2) Estimated

FIGURE 42
ABRADABLE SHROUD

ENGINE CONTROL PANEL

VEHICLE INDICATOR MODULE

DIAGNOSTIC TEST MODULE

PRELIMINARY TESTING - CAR 386

TYPICAL ACCELERATION

ROOM 10 S 04-12-77

RUN "Y" OPEN LOOP FUEL

WITH ACCEL ELIP

BSFC VS. OUTPUT POWER

UPGRADED ENGINE CYCLE COMPARISONS

Max. Turbine Inlet

Control Temperature

△ 1050C (1925F)

Regenerator Inlet

▽ 1050C (1925F)

Turbine Inlet

□ 1370C (2500F)

Regenerator Inlet

92

FIGURE 45

BSFC VS. OUTPUT POWER

Engines

- Upgraded Single Shaft
- △ 1050C (1925F)
- ▽ 1370C (2500F)

Control Temperature

- Regenerator Inlet
- - - Turbine Inlet

BSFC, g/kW-h

FIGURE 46

ENGINE OUTPUT CHARACTERISTIC

NO POWER AUGMENTATION

FIGURE 48

Handwritten:
 +W.D.
 5/6/77

APPENDIX B

SPECIFICATIONS

PREFACE

The goal of the Upgraded Engine Program is to demonstrate an experimental gas turbine engine powered vehicle which meets the 1978 Federal Emission Standards, has significantly improved fuel economy and is competitive in performance, reliability and potential manufacturing cost with the conventional piston engine powered, compact size American automobile.

The purpose of these specifications is to provide a concise summary of the pertinent design features of the Upgraded Engine-Vehicle Program. They are intended to be a functional tool, providing the central source of information necessary in the design process. They are based on the best information available at the time and are expected to change as the design process continues. The specification sheets are identified by section number, page number and date of revision.

The engine and component performance characterizations represent the program goals, attainable with a moderate development program. They do not necessarily represent the performance of the initial assemblies. The developmental parameters affecting performance include operating clearances, optimizing aerodynamic matching and minimizing parasitic losses.

The International System of Units, SI, is being used in this program. The U.S. customary units of measurements are also included in these specifications for the convenience of the user.

April 29, 1977

CONTENTS

Vehicle

Engine System

Integrated Control System

Engine Sub-Systems

Aerothermo Components

The detailed contents of a section are listed on the Title Page of that section.

UPGRADED ENGINE SPECIFICATIONS

SECTION 1

VEHICLE

<u>Contents</u>	<u>Page</u>	<u>Revision Date</u>
Vehicle - Description	1.1	4/13/76
- Modifications	1.1.1,2	4/29/77
Road Load Fuel Economy Prediction, SI Units	1.2	9/11/74
US Units	1.2.1	9/11/74
Vehicle Power Required	1.3	3/6/75
Air Intake System	1.4	12/17/74
Vehicle Accessory Arrangement	1.5	3/30/76
Vehicle Heater - A/C System	1.6	8/27/76
Vehicle Heater Package Schematic	1.6.1	8/28/76
Heater - A/C - Pneumatic Control System	1.6.2	10/15/76
Powerplant Assy - Left Side View	1.7	12/1/75
- Top View	1.7.1	12/1/75
- Front View	1.7.2	12/1/75
- Rear View	1.7.3	12/1/75
- Regenerator Cover	1.7.4	12/1/75
Upgraded Engine - Vehicle Installation	1.8	4/29/77
- Vehicle Installation Position	1.8.1	4/29/77

UPGRADED ENGINE SPECIFICATIONS

Page 1.1.1

Title Vehicle - Modifications

Date 4/29/77

The vehicle modifications required for the installation of the Upgraded Gas Turbine Engine in a 1976 F-body vehicle are summarized below. The Uniform Parts Grouping numbering system is used to identify the parts affected.

11A01 Underbody - front, center and rear floor pans modified in the tunnel area for exhaust duct clearance.

11A11 Hood - inner panel reinforcement changes.

11A13 Radiator Yoke - structural sheet metal changes for the engine air inlet, air conditioning condensor, fans, fender area engine and transmission oil coolers, and hood latch support.¹

13C Instruments - essentially the same as for piston engines; i.e. oil pressure indicator, engine temperature (hot gas rather than water temperature).

13K,M Heater, Air Conditioning controls - modified for pressure operation rather than vacuum. Add pressure regulator.

15F01 Floor Covering and Padding - carpeting and pad for modified tunnel (11A01)

17A12,13 Radiator and Front Structural Metal Rubber Parts - modified for air conditioning condensor only usage (no radiator).

30 Power Plant

Transmission - lock-up torque converter, air-to-oil cooler in front of the right front tire, remove noise "dampers", no engine-to-transmission linkage.¹

Air Filter - two large elements, insulated (noise), inlet through radiator yoke openings.

Power Steering Pump - standard unit modified for engine control and hydraulic brake booster usage (additional hoses)

Engine Oil Cooler - air-to-oil cooler in front of the left front tire.¹

Controls - electronic fuel metering and engine control system; wiring from control unit to engine sensors: engine speed (2), engine temperature, ambient temperature and pressure, accelerator pedal position.

Water Injection System (engine power augmentation). includes water spray nozzle on the air filter box, water reservoir, solenoid, hoses.

Title Vehicle - Modifications

- 31A Propellor Shaft - longer (transmission position altered)
- 32 Frame, Front Structure - engine mount location on K-member altered; transmission drop out crossmember modified for revised transmission position; transmission support crossmember modified at tunnel for exhaust duct clearance.
- 35D Brake Pedal and Bracket - modified for pedal ratio change (35E)
- 35E Power Brake Booster- hydraulic brake booster rather than vacuum (used with power steering pump)
- 36E Exhaust System - single, large duct with sound deadner, heat insulation (no muffler)
- 36F Fuel system - fuel return tube; vapor savor system to be determined.
- 36G Fender shield - battery tray and hold down location changed.
- 36K01 Battery - 95 amp hour (580 amp)
- 36K11 Electric Fuel Pump - tank mounted.
- 79 Wiring - engine compartment and instrument panel wiring changes as required for the engine control system.
- 80 Heating, Air Conditioning - standard heater core and evaporator assembly.
- Heater - add: hot gas-to-water heat exchanger, electric motor driven water pump, water reservoir, hot gas valve, hot gas tubing, water temperature sensor and control.
- Air Conditioning - Add: Electric motor driven fan for the condensor.
- 85W Window Washer - reservoir relocated.

¹Original build uses a single pass combination oil cooler located under the vehicle in the left rear seat area.

ROAD LOAD FUEL ECONOMY

ROAD LOAD FUEL ECONOMY

VEHICLE POWER REQUIRED

Date 3/6/75

FRONT VIEW

PLAN VIEW

VEHICLE ACCESSORY ARRANGEMENT

Date 3/30/76

92 K.W. (123 HP) Gas Turbine with
Single Ceramic Regenerator Tilted 20°
Compact Vehicle Installation

		SPEED RATIO, ACCY./ENGINE OUTPUT
Alternator	65 Amp	3.04
Air Conditioning Compressor	10 C.I.D. 4 cyl. Radial	1.48
Power Steering Pump	0.94 C.I.D. Vane Pump	1.11

VEHICLE HEATER - A/C SYSTEM

H ₂ O Temp	Air Pressure at Valve
195°F	5 psig (fully open)
205°F	1 psig (fully closed)

MODE	SW. PORTS						
	1	2	3	4	5	6	7
OFF	⊗	⊗	○	○	⊗	⊗	○
MAX A/C	⊗	⊗	○	○	⊗	⊗	○
A/C	○	⊗	⊗	○	⊗	⊗	○
VENT	○	⊗	⊗	○	⊗	⊗	○
HEAT	○	⊗	⊗	○	○	⊗	⊗
DEF.	○	○	⊗	⊗	○	⊗	⊗

⊗ PRESSURE

○ VENTED

Powerplant Assy. - Front View

Date 12/1/75

FRONT VIEW

Powerplant Assy. - Rear View

Date 12/1/75

REAR VIEW

UPGRADED ENGINE-VEHICLE INSTALLATION POSITION

Page 1.8.1
Date 4/29/77

Engine Position:

- engine centerline 35 mm left of vehicle centerline
- engine output, transmission on vehicle centerline
- front of engine inclined $20^{\circ}45'$ upward from horizontal

SECTION 2
ENGINE SYSTEM

<u>Contents</u>	<u>Page</u>	<u>Revision Date</u>
General Description	2.1	7/31/75
Engine Schematic	2.1.1	10/25/76
Engine Characterization		
- Schematic	2.2	3/17/75
- Flow Leak Description	2.3	10/31/75
- Flow, Heat Leaks	2.3.1	4/23/76
- SI Units	2.4	4/23/76
- US Units	2.4.1	4/23/76
Specific Fuel Consumption, BSFC-kW	2.5	4/23/76
BSFC-HP	2.5.1	4/23/76
Power Output Characteristic, US Units	2.6.1	6/10/75
Engine Response	2.7	9/11/74
Influence Coefficient - Power	2.8	1/12/77
- BSFC	2.8.1	1/12/77

GENERAL DESCRIPTIONDate 7/31/75GENERAL

Type: Regenerative gas turbine
*Rated Output: Power - 92kW (123 hp) @ 3280 rpm output shaft speed
Torque - 480 N.m (355 lb-ft) @ zero output shaft speed
Augmentation: Variable inlet guide vanes and water injection
Weight: 180 kg (400 lb)
Engine Configuration: Free rotor, designed for compact vehicle installation
Maximum gas generator acceleration, idle-to-full speed - 1.1 sec.
Fuels: Unleaded gasoline, diesel fuel, kerosene, JP-4, etc.

COMPONENTS

Compressor - Backward Curved Centrifugal with variable inlet guide vanes
First Stage Turbine - Axial
Second Stage Turbine - Axial, variable geometry stator
Regenerator - One rotating ceramic disc
Burner - Fixed geometry

DESIGN POINT CHARACTERISTICS

Maximum gas generator shaft speed - 58500 rpm
Maximum Second Stage Turbine Speed - 70000 rpm
Maximum output speed (after reduction gears) - 4700 rpm
Maximum Regenerator Speed - 31 rpm
Compressor Pressure Ratio - 4.2:1; 4.9:1 augmented
Compressor Air Flow - 0.61 kg/s (1.34 lb/s) design
0.69 kg/s (1.52 lb/s) augmented
First Stage Turbine Inlet Temperature - 1052°C (1925°F) design
Second Stage Turbine Outlet Temperature - 748°C (1378°F)

*Ambient Conditions: Temperature 29°C (85°F)
Pressure 1.013 bar (14.696 psia)

ENGINE SCHEMATIC

Date 10/25/76

ENGINE CHARACTERIZATION SCHEMATIC
STATION NOTATION
UPGRADED ENGINE

Engine Characterization, Flow Leak DescriptionDate 10/31/75

From Station () to ()

- 20 Shaft Seal Air
CTW Gas Bearing Flow
- 25 Fuel Nozzle Atomizing Air
- 34 Ring Seal - Access Panel to Power Turbine Nozzle Support
Regenerator Seal
- 36 Power Turbine Nozzle Stem Leakage
- 38 Regenerator Seal
- 39 Regenerator Seal + Carry-over
- 46 Ring Seal - Compressor Turbine Nozzle to Power Turbine
Nozzle Support
- 48 Regenerator Crossarm seal + Carry-over
- 68 Ring Seal - Power Turbine Outer Shroud to Power
Turbine Nozzle Support

Title Engine Characterization-Flow, Heat Leaks Date 4/23/76UPGRADED ENGINE CHARACTERIZATION (4/23/76)
FLOW LEAKS, HEAT LEAKS

Gas Generator Speed Fraction	0.5	0.6	0.7	0.8	0.9	0.95	1.0
<u>Flow Leaks, kg/kg; lb/lb Compressor Air Flow</u>							
From Station () to ()							
20	.00242	.00251	.00257	.00266	.00280	.00285	.00286
25	.00431	.00360	.00308	.00270	.00242	.00230	.00218
34	.00251	.00220	.00199	.00170	.00152	.00143	.00135
36	.00910	.00942	.00971	.01011	.01068	.01093	.01104
38	.00345	.00360	.00371	.00384	.00399	.00404	.00402
39	.00478	.00546	.00616	.00696	.00785	.00826	.00852
46	.00539	.00577	.00616	.00667	.00735	.00768	.00792
48	.00133	.00186	.00245	.00312	.00386	.00422	.00450
68	.00236	.00245	.00247	.00245	.00239	.00232	.00218

Heat Leaks, BTU/LB. Compressor Air Flow

30	1.303	1.345	1.392	1.429	1.533	1.587	1.632
40	2.147	1.739	1.444	1.214	1.055	.991	.947
43	2.656	2.001	1.506	1.128	.864	.757	.680
50	1.061	.885	.747	.629	.543	.505	.471
60	.213	.173	.141	.115	.097	.089	.082
63	.339	.266	.210	.164	.131	.117	.107
80	2.465	1.870	1.855	1.628	1.468	1.406	1.351
83	.898	.685	.524	.398	.310	.273	.247

Heat Leaks, J/kg Compressor Air Flow

30	3031.	3128.	3238.	3324.	3566.	3691.	3796.
40	4994.	4045.	3359.	2824.	2454.	2305.	2203.
43	6178.	4654.	3503.	2624.	2010.	1761.	1582.
50	2468.	2058.	1737.	1463.	1263.	1175.	1096.
60	495.	402.	328.	267.	226.	207.	191.
63	788.	619.	488.	381.	305.	272.	249.
80	5734.	4350.	4315.	3787.	3415.	3270.	3142.
83	2089.	1593.	1219.	926.	721.	635.	574.

Engine Characterization, S.I. Units

Date 4/23/76

UPGRADED ENGINE CHARACTERIZATION (4/23/76)29.4°C, 1.013 bar
(SI Units)

Gas Generator Speed, RPM	29 250	35 100	40 950	46 800	52 650	55 575	58 500
FRACTION	0.5	0.6	0.7	0.8	0.9	0.95 ₁₈	1.0
Power Turbine Speed, RPM	26 000	34 000	40 600	44 500	50 000	52 000	50 000
Compressor Pressure Ratio	1.498	1.788	2.177	2.700	3.383	3.770	4.185
Component Efficiency							
Compressor, 1-2*	0.769	0.784	0.798	0.804	0.804	0.797	0.785
Compressor Turbine, 5-6	0.832	0.836	0.841	0.843	0.846	0.847	0.846
Power Turbine, 6-8	0.719	0.719	0.741	0.755	0.764	0.765	0.762
Burner	0.998	0.998	0.999	0.999	0.999	0.999	0.999
Regenerator	0.957	0.948	0.939	0.928	0.916	0.911	0.905
Parasitic Loss, kW							
Gas Generator	0.63	0.90	1.23	1.61	2.03	2.26	2.51
Power Turbine	1.46	2.15	3.12	3.90	5.15	5.73	5.57
Output Power, kW Net	4.87	10.69	20.71	36.24	56.70	67.48	77.81
Fuel Flow, kg/h Diesel	2.49	4.10	6.55	10.20	15.16	18.02	21.01
Specific Fuel Consumption, g/w-h	0.512	0.384	0.316	0.281	0.267	0.267	0.270
Gas Flow, kg/s							
Station 1	0.196	0.256	0.323	0.408	0.503	0.555	0.606
3	0.191	0.249	0.315	0.397	0.489	0.538	0.587
4	0.191	0.249	0.314	0.396	0.488	0.537	0.586
5	0.192	0.250	0.315	0.397	0.489	0.539	0.588
6	0.194	0.253	0.320	0.403	0.497	0.548	0.598
8	0.196	0.255	0.322	0.407	0.502	0.554	0.604
9	0.197	0.256	0.324	0.410	0.506	0.558	0.610
Pressure, kPa							
Station 1	101.17	101.05	100.89	100.64	100.28	100.05	99.80
2=3	151.55	180.68	219.64	271.71	339.23	377.19	417.69
4	150.40	179.38	218.23	270.20	337.66	375.60	416.08
5	147.95	175.90	213.67	264.35	330.56	367.86	407.76
6	120.41	132.10	147.31	165.41	186.19	195.61	203.37
8	103.30	104.13	105.20	106.76	108.84	110.10	111.48
9	101.59	101.79	102.12	102.68	103.54	104.11	104.77
Temperature, °C							
Station 1	29.4	29.4	29.4	29.4	29.4	29.4	29.4
2	77.7	99.2	123.8	152.6	185.4	203.4	222.7
3	83.8	103.0	125.7	153.3	184.9	202.4	221.3
4	700.7	700.1	698.1	695.6	692.8	692.2	691.2
5	829.4	864.7	907.3	954.7	1005.6	1029.6	1051.7
6	776.1	792.1	812.6	834.1	855.5	863.4	868.5
7.5	747.8	747.8	747.8	747.8	747.8	747.8	747.8
8	738.8	740.7	740.8	741.5	742.1	742.3	742.5
9	128.7	152.8	180.0	212.3	248.3	266.8	286.8

* Numerics Refer to Station Notation

Engine Characterization, English UnitsDate 4/23/76UPGRADED ENGINE CHARACTERIZATION (4/23/76)85°F, 14.696 PSIA
(English Units)

Gas Generator Speed, RPM FRACTION	29,250 0.5	35,100 0.6	40,950 0.7	46,800 0.8	52,650 0.9	55,575 0.95	58,500 1.0
Power Turbine Speed, RPM	26,000	34,000	40,600	44,500	50,000	52,000	50,000
Compressor Pressure Ratio	1.498	1.788	2.177	2.700	3.383	3.770	4.185
Component Efficiency							
Compressor, 1-2*	.769	.784	.798	.804	.804	.797	.785
Compressor Turbine, 5-6	.832	.836	.841	.843	.846	.847	.846
Power Turbine, 6-8	.719	.719	.741	.755	.764	.765	.762
Burner	.998	.998	.999	.999	.999	.999	.999
Regenerator	.957	.948	.939	.928	.916	.911	.905
Parasitic Loss, HP							
Gas Generator	.84	1.21	1.65	2.15	2.72	3.04	3.37
Power Turbine	1.95	2.89	4.19	5.22	6.90	7.68	7.46
Output Power, HP Net	6.53	14.33	27.77	48.60	76.04	90.50	104.34
Fuel Flow, Lb/Hr Diesel	5.49	9.05	14.44	22.49	33.43	39.72	46.33
Specific Fuel Consumption Lb/Hp-Hr	.842	.631	.520	.463	.440	.439	.444
Gas Flow, Lb/Sec							
Station 1	.433	.564	.713	.900	1.110	1.223	1.335
3	.421	.549	.694	.875	1.078	1.186	1.295
4	.421	.548	.693	.874	1.076	1.185	1.293
5	.423	.550	.695	.876	1.079	1.187	1.296
6	.428	.558	.704	.889	1.096	1.207	1.318
8	.431	.562	.711	.898	1.107	1.220	1.332
9	.433	.565	.715	.904	1.116	1.230	1.344
Pressure, PSIA							
Station 1	14.673	14.657	14.633	14.596	14.544	14.511	14.476
2=3	21.980	26.206	31.856	39.409	49.201	54.706	60.580
4	21.814	26.017	31.651	39.190	48.974	54.476	60.348
5	21.458	25.513	30.990	38.341	47.944	53.353	59.141
6	17.463	19.159	21.365	23.990	27.004	28.370	29.495
8	14.983	15.102	15.258	15.484	15.785	15.968	16.169
9	14.734	14.764	14.811	14.893	15.018	15.100	15.196
Temperature, °F							
Station 1	85.0	85.0	85.0	85.0	85.0	85.0	85.0
2	171.9	210.6	254.8	306.8	365.8	398.1	432.8
3	182.8	217.3	258.3	307.8	364.9	396.3	430.4
4	1293.3	1292.1	1288.5	1284.1	1279.0	1277.9	1276.2
5	1525.0	1588.4	1665.2	1750.4	1842.0	1885.3	1925.0
6	1429.0	1457.5	1494.6	1533.4	1571.9	1586.2	1595.2
7.5	1378.0	1378.0	1378.0	1378.0	1378.0	1378.0	1378.0
8	1361.8	1364.7	1365.4	1366.8	1367.7	1368.1	1368.6
9	263.7	307.1	355.9	414.1	479.0	512.3	548.3

* Numerics Refer to Station Notation

Minimum
BSFC, g/kWh

POWER OUTPUT CHARACTERISTIC, US UNITS

Date 6/10/75

POWER TURBINE GROSS OUTPUT
100% N_{GG} , -30° VIGV *

ENGINE ACCELERATIONT/C OUTPUT TORQUE@ STALL

INFLUENCE COEFFICIENT - POWERDate 1/12/77

Upgraded Engine Characterization (4/23/76)

$$I.C. = \frac{\Delta HP/HP}{\Delta x/x}$$

$$* P_{loss} = \frac{r_{CT} r_{PT}}{r_{CP}}$$

INFLUENCE COEFFICIENT - BSFC

Date 1/12/77

Upgraded Engine Characterization (4/23/76)

$$I.C. = \frac{\Delta BSFC / BSFC}{\Delta X / X}$$

I.C., Influence Coefficient - BSFC

$$*P_{loss} = \frac{r_{CT} \quad r_{PT}}{r_{CP}}$$

TABLE OF CONTENTS

<u>Subject</u>	<u>Page</u>
<u>FUNCTIONAL SPECIFICATION:</u>	
Control Mode	3.2
Fuel Control - Single Stage Burner	3.2
Fuel Splitter Adaptation - Dual Stage Burner	3.8
Power Turbine Nozzle Control	3.9
Variable Inlet Guide Vane Control	3.11
Water Injection Control	3.11
Start Sequencing and Engine Protection	3.14
<u>DESIGN SPECIFICATION:</u>	
Sensors	3.17
Fuel System	3.23
Power Turbine Nozzle Actuator	3.23
Inlet Guide Vane Actuator	3.26
Water Injection System	3.26
Electronic Control Unit	3.28
Pedal	3.29
Control Wiring Harness	3.29
Instrumentation	3.29
Diagnostics	3.30

PRELIMINARY DRAFTCONTROL MODE

The integrated control system for the Upgraded Engine will have full authority over engine operation. It will be a three-loop system (fuel control, power turbine nozzle control, and inlet guide vane control) along with controls for water injection and start sequencing.

The control system will safeguard the engine from overspeeding and overtemperatures. It will also provide diagnostic interfaces readily available from the control system for ease of maintenance and periodic evaluations.

Included in the following details are environmental, functional, and interfacing specifications for the various sub-systems. Engine and control system parameters are shown in Figure 1.

FUEL CONTROL - SINGLE STAGE BURNER

The integrated control must function with a single stage burner having an associated torch igniter. A fuel splitter adaptation, if required, will be utilized for the dual stage burner. Figure 2 shows a functional block diagram for the fuel control sub-system.

Fuel Metering

Metering of the fuel for the single-stage burner and torch igniter is shown on the fuel schedule, Figure 3.

FIG. 3 - FUEL SCHEDULE-SINGLE STAGE BURNER

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATIONDate 4/30/76ENGINE AND CONTROL SYSTEM PARAMETERS

T_8	Low Pressure Regenerator Inlet Temperature (Exhaust Diffuser Outlet Temperature, $T_{7.5}$, Sensed)
$T_8/\sqrt{\theta}$	Corrected L.P. Regenerator Inlet Temperature
T_a	Ambient Temperature
N_{gg}	Gas Generator Shaft Speed
$\frac{N_{gg}}{\sqrt{\theta}}$	Corrected Gas Generator Speed
δ	Ambient Pressure Correction = $\frac{P_1 \text{ PSIA}}{14.7 \text{ PSIA}}$
θ	Ambient Temperature Correction = $\frac{T_1 \text{ } ^\circ\text{R}}{545 \text{ } ^\circ\text{R}}$
N_{os}	Output Speed, Vehicle Acc'y Drive Shaft
α	Throttle Pedal Position
β	Power Turbine Nozzle Angle Position
$\Delta\beta$	Power Turbine Nozzle Trim Angle
γ	Inlet Guide Vane Angle Position
P_2	Compressor Discharge Pressure
$W_f/\delta\sqrt{\theta}$	Corrected Fuel Flow in Pounds Per Hour
P_a	Ambient Pressure

FIGURE 1

FIGURE 2 - FUEL CONTROL SYSTEM BLOCK DIAGRAM

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATION

Date 4/30/76

Torch Igniter

- Fuel Pressure Supply 20 psi (ΔP above P_2)
- Fuel Flow @ 50% N_{gg} 0.5 to 1.5 lbs. per hour
- @ 100% N_{gg} 1.0 to 5.0 lbs. per hour

Start/Acceleration Fuel Schedule

The start/acceleration fuel schedule is based on a scheduled turbine inlet temperature (T_5) for material life considerations, compressor stability, fuel economy, and emissions. It represents a constant corrected turbine inlet temperature (T_5/θ) as a function of $N_{gg}/\sqrt{\theta}$.

A compressor map (see Pg. 5.1.1) shows various T_5 operating lines from match to surge conditions.

Deceleration

The main burner fuel nozzle will have a positive shut-off. However, flow will be maintained to the torch igniter at all times in any operating mode.

Governing

Gas generator speed governed at steady state by an isochronous governor in accordance to a pedal schedule as shown in Figure 4.

FIGURE 4 - PEDAL SCHEDULE OF $N_{gg}/\sqrt{\theta}$

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATIONDate 4/30/76

For low emissions combustor control to avoid extended operation in the stoichiometric zone, a switched governor gain configuration with speed error logic is required.

Idle

Ngg speed will be corrected by T_a to prevent excessive creep torque at lower ambient temperatures. Idle speed adjustable from 45% to 50% Ngg.

If the Ngg speed decays to starter cut-off speed ($\cong 35\%$ Ngg), the engine shall be shut down to prevent an excessive over-temperature condition. (Failure to re-light burner.)

Maximum Speed - 58500 rpm (Ngg)

- Over-speed protection ($100\% \pm 2\%$)
- Rated power turbine maximum speed referenced to $N_{os} = 5200$ rpm.

Power Limiting

Ngg will be speed limited (range 68-77% Ngg) in Start/Park and neutral transmission positions.

The following operating mode will be used for power limiting during power augmentation as well as low output power control at idle speed to obtain satisfactory vehicle operation and not exceed material property stress limitations under ambient conditions.

As shown in Figure 5, low output power control at a corrected gas generator idle speed ($N_{gg}/\sqrt{\theta} = 50\%$) is maintained by holding a corrected regenerator inlet temperature ($T_8/\sqrt{\theta}$) constant at 1380°F over the ambient temperature range from -20°F to $+85^\circ\text{F}$. Above $+85^\circ\text{F}$ to $+130^\circ\text{F}$, a constant actual T_8 is maintained at 1380°F . During this idle mode of operation the IGV's are modulated from 0° to $+60^\circ$ to maintain accessory drive idle speed.

As shown in Figure 6 for power limiting during augmentation (IGV's @ -30° and $N_{gg}/\sqrt{\theta} > 96\%$) both $T_8/\sqrt{\theta}$ and $N_{gg}/\sqrt{\theta}$ are held constant at 1350°F and 100% respectively over an ambient temperature (T_a) range from -20°F to $+85^\circ\text{F}$. Above $+85^\circ\text{F}$ to $+130^\circ\text{F}$ both actual T_8 and Ngg are held constant at 1350°F and 100% respectively. With water injection augmentation utilized above a fixed ambient temperature (T_a) of $+60^\circ\text{F}$ ($\pm 5^\circ$), $T_8/\sqrt{\theta}$ and $N_{gg}/\sqrt{\theta}$ are held constant at 1320°F and 100% in the ambient temperature range of $+60^\circ\text{F}$ to $+85^\circ\text{F}$. Above 85°F to $+130^\circ\text{F}$ actual T_8 and Ngg are held constant at 1320°F and 100%.

Output Shaft Overspeed

Safety will control Ngg to limit power turbine shaft speed to a maximum safe limit. ($100\% N_{os} = 5200$ rpm).

FIG. 5 - SCHEDULE-IDLE SPEED AMBIENT CONTROL

FIG. 6 - SCHEDULE-AMBIENT POWER LIMITING

FUEL SPLITTER ADAPTATION - DUAL STAGE BURNERFuel Splitter Adaptation (If Required)

A fuel splitter adaptation is required with a dual stage burner. The total fuel flow requirement is the same for both the dual and single stage burner. However, shown in Figure 7 is the split of the total fuel flow between the two stages of the burner. The requirements for the torch igniter remain the same. A description of this splitter schedule affects different engine modes follows:

Start/Acceleration - The first stage fuel flow will commence at 5-10% Ngg, and the second stage at 35% Ngg (starter cut-off speed). The first stage will flow all of the start cycle flow from 5-10% to 35% Ngg and only 1/3 ($\pm 10\%$) of the total start and acceleration fuel flow above 35% Ngg.

Deceleration - Both first and second stage fuel flow will have a positive shut-off during deceleration. However, flow will be maintained to the torch igniter at all times in any operating mode.

FIG. 7 - FUEL SCHEDULE-DUAL STAGE BURNER

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATIONDate 10/31/75

Governing - The same as the single stage except the first stage will have a minimum volumetric floor equivalent to 4.5 lbs. per hour ($\pm 20\%$).

Idle - Same as single stage.

Maximum Speed - Same.

Power Limiting - Same.

Power Turbine Overspeed - Same.

POWER TURBINE NOZZLE CONTROL

A functional description of the power turbine nozzle control in the various control modes follows. A functional block diagram for the power turbine nozzle control system is shown in Figure 8.

Start - Fixed nozzle angle position (range $+2^\circ \pm 2^\circ$). The 0° angle represents the engine match condition as determined by the engine on the test stand at 100% design Ngg speed.

Acceleration - Fixed nozzle angle ($+10^\circ \pm 2^\circ$).

Steady State - The Nozzles will be modulated to maintain a scheduled $T_8/\sqrt{\theta}$ over the full Ngg/ $\sqrt{\theta}$ corrected speed range. See Figure 9 for the $T_8/\sqrt{\theta}$ schedule which includes water injection and power augmentation at the top end of the speed range. $T_8/\sqrt{\theta}$ reference will be 1380°F actual in the normal mode, 1350°F augmented with IGV's, and 1320°F augmented with IGV's and water injection.

FIG. 9 - SCHEDULE-POWER TURBINE NOZZLE ACTUATOR

FIGURE 8 - POWER TURBINE NOZZLE CONTROL BLOCK DIAGRAM

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATIONDate 4/30/76

Engine Braking - Will be modulated and the power turbine nozzles controlled to a maximum T8. Engine braking will be affected by pedal position (α) 0-3% range, Nos above 750 to 800 rpm.

Safety - If loss of electrical power occurs, the power turbine nozzles will revert to a fixed nozzle position ($10^\circ \pm 2^\circ$). If loss of hydraulic pressure or temperature signal (T8) occurs, the fixed nozzle position will be ($2^\circ \pm 2^\circ$).

VARIABLE INLET GUIDE VANE CONTROL

Inlet guide vanes are installed on the engine just ahead of the centrifugal compressor to augment power at or near maximum gas generator speed over the whole ambient temperature (T_a) range. In addition, the vanes control engine air flow at idle speed while maintaining cycle temperature. A block diagram for the inlet guide vane control is shown in Figure 10. Figure 11 shows a schedule of the vane angle (γ) as a function of the modulated pedal position (α); additionally modified for Nos idle governing. A description of the inlet guide vane control functions in the various control modes follows:

Start/Acceleration/Deceleration - The inlet guide vanes will be in the ($0^\circ + 5^\circ/0^\circ$).

Governed Power Turbine Idle Speed - Power turbine idle speed at curb Ngg idle is governed for varying accessory loads by modulating the inlet guide vanes from $+60^\circ$ to 0° .

Low Power Modulation Control - Low power modulation at idle gas generator speed for 0-6% of the pedal travel is controlled by modulating the inlet guide vanes from $+60^\circ$ to 0° .

Power Augmentation - At 96% ($+2\%/0\%$) Ngg, the inlet guide vanes are slewed from 0° to -30° for power augmentation as shown in Figure 11.

Safety - An electrical or hydraulic power loss to the inlet guide vane actuator will position the inlet guide vanes to ($0^\circ + 5^\circ/-0^\circ$).

WATER INJECTION CONTROL

For power augmentation, water will be sprayed through a nozzle or nozzles into the compressor intake downstream of the air filters. A block diagram of the water injection control is shown on Figure 12.

Operating Mode - Water injection will be utilized above a fixed inlet temperature (T_a) of 60°F ($\pm 5^\circ$). A fixed flow (range 100-120% of fuel flow at 100% Ngg) will be metered at maximum demand (100% pedal position and 96% ($+2\%/0\%$) Ngg).

FIGURE 10 - IGV CONTROL BLOCK DIAGRAM

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATION

FIGURE 11 - IGV POSITION (γ) SCHEDULE

FIGURE 12 - WATER INJECTION BLOCK DIAGRAM

INTEGRATED CONTROL SYSTEM FUNCTIONAL SPECIFICATIONDate 4/30/76

Actuation - A switch and an indicator light will be utilized at maximum pedal travel for the demand signal.

START SEQUENCING AND ENGINE PROTECTION

The typical start control system uses automatic sequencing to arrive at a start/run condition after initiation by a signal from the key switch. Additional logic circuits are included to prevent or abort a start once initiated by a signal from the key switch. The start sequencing and engine protection control block diagram is shown in Figure 13.

Start/Run Logic - The following chronological events occur for a successful start:

- Transmission selector level in "Start-Park" position.
- Key switch initiation.
- Starter, Accessory Clutch, Fuel Pump, Ignition, and Start Sequence Logic Circuits energized.
- Speed Sensor checked satisfactory.
- Fuel enable energized @ 10% (+2%/-0%) Ngg.
- Burner light-off occurs.
- Gas Generator accelerates.
- Starter and Accessory Clutch de-energized @ 35% Ngg.
- Gas generator self-sustaining.
- Fuel Control Governor effective at Ngg idle speed.

"No-Start" Logic - The following conditions prevent a start condition from being initiated.

- Transmission Selector Lever not in a "Start-Park" position.
- Gas Generator still in "run" logic (prevents starting motor from being energized with resultant damage to clutch mechanism).
- Abort or shut-down not reset.

"Start Abort" - The following logic will abort a start during the start cycle mode. An automatic reset period of 30 seconds is required before another key start is attempted. (Allows excess fuel within the gas generator to be "scavenged".)

- Loss of Ngg Speed Sensor or feedback signal.
- No light-off in 10 seconds (light-off represents a rate of rise of 100°F/second based on T8.)

FIGURE 13 - START SEQUENCING AND
ENGINE PROTECTION BLOCK DIAGRAM

INTEGRATED CONTROL SYSTEM FUNCTION SPECIFICATIONDate 10/31/75

- Failure to reach cut-off speed (35% Ngg) in 20-25 seconds.
- Over-temperature for 2-second duration.

Engine Protection (Running Mode) - The engine is protected during a running mode (gas generator above starter cut-out speed of 35% Ngg) by the following logic:

- Shut Down - If Ngg decays to 30-35% speed, the engine will be shut down to prevent extreme over-temperature damage to the engine.
- Indicator Lights - Activated by the following:
 - Over-temperature
 - Low oil pressure
 - No start

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

PRELIMINARY DRAFT

SENSORS

Gas Generator Speed (N_{gg}) - Speed Sensing Monopole

- Diameter 3/8 inch. (maximum)
- Shoulder assembly
- Pigtail outlet with connector
- Temperature environment
 - 40°F to 200°F Running
 - 40°F to 300°F Soak
- Speed range 10% to 100% N_{gg} (5850 to 58500 rpm)

Unit to be mounted to compressor inlet housing and will function by means of the steel inducer blade tips. See Figure 14.

Output Speed, Vehicle Accessory Drive Shaft (N_{os})

Sensor identical or equivalent to the N_{gg} speed sensing monopole.

Unit to be mounted on a bracket and will function by means of a 60-tooth wheel attached to or a part of the vehicle acc'y. drive pulley (speed range 600-5200 rpm). See Figure 15.

Low Pressure Regenerator Inlet Temperature (T₈)

- Resistance type sensor probe preferred (Fluidic Probe Optional).
- Two-hole mounting flange. See Figure 16.
- Length (approximately 4 in.).
- Push-on connector at end of probe.
- Temperature range 1600°F (maximum).
- Tolerance @ 1400°F ($\pm 10^\circ\text{F}$).

FIGURE 14 - Ngg Sensor Installation

FIGURE 15 - Output Speed Sensor Installation, N_{OS}

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

FIGURE 16 - T8 SENSOR INSTALLATION

Ambient Temperature (Ta)

- Thermistor or resistance type probe
- 1/4 to 3/8 inch diameter
- Flange mounted. See Figure 17.
- Temperature range: -40°F to +200°F.

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

FIGURE 17 - Ta SENSOR INSTALLATION

Ambient Pressure (Pa) Transducer

- Range: Sea level to 14,000 feet
- Mounted in Electronic Control Unit
- Temperature environment:
 - 40°F to 150°F Running
 - 40°F to 200°F Soak

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONSDate 10/31/75Pedal Position Potentiometer (α)

- Cermet or plastic type potentiometer rotary motion with 100% switch.
- Mounted on central control linkage mounting. See Figure 18.
- Operated by pedal through the central control linkage.
- Temperature environment:
 - 40°F to 200°F Running
 - 40°F to 250°F Soak
- Push-on connectors

TO BE DETERMINED

FIGURE 18 - PEDAL POTENTIOMETER INSTALLATION

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONSDate 10/31/12FUEL SYSTEM

The design specifications for the fuel metering and supply system are as follows:

- Design life 3500 hours (minimum) with gasoline.
- Ambient pressure operating range - sea level to 14,000 feet.
- Fuel system components in engine compartment environment temperature range
 - 40°F to 250°F Running
 - 40°F to 300°F Soak
- Multi-fuel capability (unleaded gasoline to Diesel #2).
- Fuel system must tolerate dirt and water. A 40 micron filter (suction line) is provided.
- Fuel system must be immune from vapor lock conditions.
- Fuel system operating pressure - $\Delta P = 20$ psi above P_2 for torch igniter.
- Provisions for running a 75 psig fuel line from the gas tank pump to the engine with a safety shut-off feature in case of line rupture.

POWER TURBINE NOZZLE ACTUATOR

The design specifications for the power turbine nozzle actuator and control are as follows:

- Load capacity
 - 50 pounds (static)
 - 20 pounds (dynamic)
- Travel
 - 1.80 in. total stroke for 122° total nozzle blade angle change. See Figure 19 for P.T. nozzle angle control program.

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONSDate 10/31/75

FIGURE 19
P.T. NOZZLE ANGLE CONTROL PROGRAM

- Response
 - 0.1 second for 1.38 in. stroke (not to exceed 0.2 second).
- Power source (supplied by Chrysler)
 - Fluid pressure differential - 70 psid @ 2 GPM
 - Fluid temperature - 100°F to 275°F
 - Fluid type - Texaco 4630
 - Chrysler MS5931 (see Specification, page 4.18.1)
- Mounting
 - See Figure 20.
- Temperature environment
 - 40°F to 250°F Running
 - 40°F to 300°F Soak

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

FIGURE 20
P.T. NOZZLE ACTUATOR INSTALLATION

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONSDate 10/31/75

- Safety

 Ideal failure position - idle (+ 2°) (Spring actuated in case of hydraulic power failure).

INLET GUIDE VANE ACTUATOR

The design specifications for the inlet guide vanes actuator and control are as follows:

- Load

 15 pounds (static)

 10 pounds (dynamic)

- Travel: 1.6 in. (linear)

- Response

 0.1 second for 1.07 in. stroke (not to exceed 0.2 seconds)

- Power Source (supplied by Chrysler)

 Fluid pressure differential - 70 psid @ 2 GPM

 Fluid temperature - 100°F to 275°F

 Fluid type - Texaco 4630

 - Chrysler MS5931 (specification attached), see pg. 4.18.1

- Mounting

 See Figure 21.

- Safety

 Ideal failure position - 0° (+5°/-0°)

WATER INJECTION SYSTEM

- Water specification

 Commercial distilled

Filtration

 40 micron in tank filter (same as gasoline tank)

<u>Vane Angle, °</u>	<u>Ring Gear Rotation Degrees From Ref.</u>	<u>Actuator Piston Motion, mm From Ref.</u>	<u>Vane Torque Per Piston Force, mm (1 Vane)</u>
-30	-7.3956	-12.0035	18.313
-24	-6.1335	- 9.9747	20.408
-18	-4.7478	- 7.7358	22.317
-12	-3.2525	- 5.3087	23.995
- 6	-1.6638	- 2.7198	25.400
0	0	0	26.489
6	1.7189	2.8160	27.232
12	3.4714	5.6905	27.604
18	5.2353	8.5842	27.597
24	6.9878	11.4571	27.211
30	8.7067	14.2705	26.462
36	10.3705	16.9875	25.376
42	11.9592	19.5745	23.985
48	13.4545	22.0017	22.329
54	14.8402	24.2433	20.448
60	16.1023	26.2778	18.381

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

Date 3/24/76

FIGURE 21
INLET GUIDE VANE ACTUATOR INSTALLATION

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONSDate 10/31/75

- Freezing protection

Water tank to be of a freeze design type.

- Nozzle

TO BE DETERMINED

- Power source

Water tank to be pressurized by compressor discharge pressure.

Water injection controls including nozzle and solenoid valve will be the full responsibility of AiResearch.

Chrysler will provide the water tank, water volume sensor, and tank gauge.

ELECTRONIC CONTROL UNIT

The Electronic Control Unit contains all of the circuitry required to condition sensor input signals, provides the necessary schedules and control logic, and controls power outputs to the various control system interfaces for full authority control over engine operation.

- Unit must be compact to be installed in the instrument panel area if required space is available.

- Temperature environment

40°F to 150°F Running

-40° to 200°F Soak

- Voltages in the electrical system are as follows:

Nominal - 12v DC (negative ground)

Continuous - 15v DC

Starting - 5 to 15v DC

Transient voltages typical during automotive operating conditions

- Power requirements - Limited to 120 watts steady state operation except for power augmentation.

- Unit shall be capable of full operation and not be adversely affected by any electromagnetic interference normally encountered in automotive operation.

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONSDate 10/31/75

- The unit and components within must be properly mounted to withstand any shock and vibration encountered with automotive operations.
- One or more of push-on multiple automotive type connectors will be used on the Electronic Control Unit.

PEDAL

The pedal will actuate by means of proper linkage the following:

- Pedal position potentiometer
- Transmission control lever
- Water injection switch

CONTROL WIRING HARNESS

The Subcontractor shall provide specifications for all engine control wiring and harnesses. Chrysler will fabricate wiring and harnesses in accordance with the Subcontractor's specifications.

- The harness connectors must be low cost, automotive type and provide protection from water splash, engine compartment and underbody environments.
- Temperature environment: -40°F to $+300^{\circ}\text{F}$.
- All connections and wire splices will be secured by standard automotive type welds or soldered joints.
- The wiring harness must be protected from external effects that could cause engine overspeed, overtemperature, loss of control, etc.
- The harness connectors must be color coded and protected from accidental interchange.

INSTRUMENTATION

Chrysler will provide the following instrumentation with the Subcontractor providing the output signals to drive them.

- Regenerator inlet temperature (T8) meter: 800°F to 1500°F
Signal: 0 to 1 MA.

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

Date 1/23/76

- Gas generator speed tachometer ($N_{gg}/\sqrt{\theta}$) 0 to 100% speed.
Signal: 0 to 1 MA.
- Miles per gallon (MPG) meter (optional): 0 to 40 MPG
Signal: 0 to 1 MA.

All output signals must have limits for preventing meter or indicator damage due to transients or external effects.

DIAGNOSTICS

The integrated control system will have built-in diagnostic capabilities. Output signals will be connected to a diagnostic plug for ease of maintenance and for routine engine and vehicle evaluations.

Only sensor outputs and command signals available from normal control requirements will be utilized for diagnostic purposes.

Control input parameters (sensed):

- Compressor inlet temperature - T_a
- Regenerator inlet temperature - T_8
- Ambient pressure - P_a
- Volume fuel flow - W_f
- Gas Generator speed - N_{gg}
- Vehicle Accessory drive shaft speed - N_{os}
- Throttle pedal position - α
- Start/Park position (on-off)

Control system output signals:

- Fuel flow - (W_f)
- Power turbine nozzle trim angle - $\Delta \beta$
- Inlet guide vane angle - γ
- Corrected gas generator speed - $N_{gg}/\sqrt{\theta}$

INTEGRATED CONTROL SYSTEM ELECTRO/HYDRO/MECHANICAL
DESIGN SPECIFICATIONS

- Regenerator inlet temperature - T8
- Vehicle Accessory Drive Shaft Speed - Nos
- Water injection (on-off)
- Starter switch (on-off)
- Start cycle (on-off)
- Overtemperature (on-off)
- 10% speed error switch - (on-off)
- PTN Actuator braking mode switch - (on-off)
- $\propto$ Braking switch - (on-off)
- N_{OS} braking switch - (on-off)

UPGRADED ENGINE SPECIFICATIONS

SECTION 4

ENGINE SUBSYSTEMS

<u>Contents</u>	<u>Page</u>	<u>Revision Date</u>
General Configuration		
- Description	4.1	1/23/76
- Engine Cross Sections		
- Longitudinal	4.1.1	10/25/76
- Air Inlet	4.1.2	10/25/76
- Burner Chamber	4.1.3	10/25/76
- Bulkhead	4.1.4	10/25/76
Engine Housing		
- Front View	4.2	4/2/76
- Burner Side View	4.2.1	4/2/76
- Regenerator Side View	4.2.2	4/2/76
- Rear View	4.2.3	4/2/76
- Structure at Bulkhead	4.2.4	12/17/74
Gas Generator Assembly	4.3	4/20/76
- Front View	4.3.1	4/20/76
Cover - Gas Generator	4.4	10/3/75
Diffuser - Compressor	4.5	12/17/74
Support - Gas Generator	4.6	12/17/74
Compressor Oil Bearing Carrier	4.7	12/17/74
Turbine Wheel Air Bearing Carrier	4.8	4/29/77
Gas Bearing Design (7 pages)	4.9	11/23/76
Interstage - Component Data	4.10	12/17/74
- Assembly	4.10.1	4/30/76
Power Turbine Assembly	4.11	4/20/76
Gear and Sprocket Data	4.12	1/23/76
Drive Schematic	4.13	1/12/76

2

165

GENERAL CONFIGURATION-DESCRIPTION

Engine Housing:

Material: Nodular Cast Iron - High Silicon

Wall Thickness: 6 mm except at flanges, high stress areas, etc.

Casting Weight: 77 kg

Single Regenerator Cavity: 76 mm offset from engine rotor

Single Burner Opening: 84 mm offset from engine rotor

Three Compressor Discharge Openings:

Two direct to regenerator cavity

One for cooling bulkhead, then into regenerator cavity.

Second Stage Nozzle Bulkhead:

Double walled, cooled by compressor discharge

Reinforced with integral ribs

Power Section:

Second Stage Turbine Bearing Support:

Material: Aluminum

Thickness: Contoured to control bearing soakback temperatures.

Reduction Gear Housing with Integral Oil Sump:

Material: Cast Iron

Regenerator Cover:

Material: Aluminum

ENGINE - AIR INLET CROSS-SECTION

ENGINE - BURNER CHAMBER CROSS-SECTION

Date 10/25/76

Engine Housing - Front View

Date: 4/2/76

ENGINE HOUSING SHOWING BULKHEAD AND BURNER WALL
STRUCTURAL ARRANGEMENT

Page 4.3.1

Title Gas Generator Assembly, Front View

Date 4/20/76

Aluminum casting

Intermittent pilot with Support

Collects compressor outlet air and routes it into three integral lobes which discharge into engine housing.

Provides mount for VIGVs and air intake housing.

COMPRESSOR COVER STRUCTURAL ANALYSIS

FINITE ELEMENT MODEL

STRESS AND DEFLECTION SUMMARY

Section	Temperature (Estimate)	Axial Deflection (in.)		Equivalent Stress (PSI)	
		Original	Final	Original	Final
1	400° F	.0170	.0036	16,800	2,100
2		.0160	.0047	11,300	3,100
3		.0150	.0055	10,800	5,400
4		.0160	.0060	11,000	5,200
5		.0170	.0058	7,700	4,600
6		.0155	.0049	10,300	4,400
7		.0215	.0038	21,500	2,800
8		.022	.0031	6,000	2,500
9		.0206	.0030	13,300	1,200

Material: Aluminum

Integral machined vanes

Pilots into cover

Vane Inserts: Machined steel segments

Material: Aluminum casting

Pilot Ring of steel pinned to Support and machined concentric. Circular pilot with engine housing.

Internal passages for oil, air and oil drain.

Mounts separate turbine wheel air bearing carrier and compressor oil bearing carrier.

Material: Aluminum

Pilot: Into Support

Sealing by "O" rings

Sub-Components: Journal and thrust bearing
Graphite seal assembly

TURBINE WHEEL AIR BEARING CARRIER

Material: Cast Iron

Pilot: Long "finger" to cope with thermal gradients

Alternate: Provision for oil bearing

Sub-Components: Bearing assembly

Graphite seal assemblies

The following guideline outlines the requirements of the compressor turbine journal gas bearing used in the upgraded engine. Specifications for the engine and gas generator have been estimated and included where they are critical to bearing operation.

- A. Service requirements: 3500 hours of engine operation with 25,000 start/stop cycles; this corresponds to approximately 100,000 miles of vehicle operation.
- B. Miscellaneous loads on rotor:
 - 1. Rotor unbalance at bearing planes, 2000 rpm: 0.071N-mm (0.010 oz. - in.).
 - 2. Radial "g" load: 3.5 maximum.
 - 3. Gyroscopic forces during a vehicle skid at 2 rad/sec at 90% gas generator speed.
- C. Temperatures:
 - 1. Ambient range: -29°C to 46°C (-20°F to 115°F).
 - 2. Turbine inlet gas: 1052°C (1925°F) maximum.
 - 3. Maximum bearing soakback (calculated): 343°C (650°F).
 - 4. Compressor discharge air: 246°C (475°F).
- D. Initial starting torque must not be excessive so that starting time can be held to a reasonable value. Bearing friction (prior to shaft lift off) must not cause the gas bearing loss to exceed roughly 0.6KW (0.8 HP) at 11% of maximum speed (6500 rpm). If the predicted loss approaches this value further study is warranted.
- E. General gas generator specifications:
 - 1. Pressure ratio: 1.5 to 4.9.
 - 2. Speed range: 29,300 to 58,500 rpm.
 - 3. Bearing support material: cast iron.
 - 4. Running clearance control - turbine wheel blade to shroud: the maximum operating eccentricity of the geometric center of the wheel relative to the geometric center of the bearing support should be limited to 0.13 mm (.005 in.).

GAS BEARING DESIGNDate 11-23-76

5. Component specifications

	I_p <u>kgm² (in-lb-sec²)</u>	<u>Weight</u> <u>Kg. (lb)</u>
Impeller with steel front section, hub	0.00083 (0.0074)	0.77 (1.7)
Turbine Wheel with integral shaft	0.00047 (.0042)	0.74 (1.63)
Sleeve	0.00011 (.001)	0.53 (1.17)
Total	0.00142 (.0126)	2.04 (4.5)

Impeller c.g. is 19.56 mm (0.77 in.) forward of the impeller rear face mating with hub.

SUMMARY OF AIR BEARING CONCEPTBasic CharacteristicsAir Bearing Dimensions

(MTI313E31; Chrysler 3813871)

Diameter	38.1mm	(1.5 in.)
Length	38.1mm	(1.5 in.)
L/D	1.0	
Projected area	1451mm ²	(2.25 in ² .)
No. of pads	1	
Smooth foil thickness	0.10mm	(0.004 in.)
Bump foil thickness	0.08mm	(0.003 in.)
Bump characteristics		
Height	0.521/0.533mm	(0.0205/0.0210 in.)
Length	3.56mm	(0.140 in.)
Pitch	4.57mm	(0.180 in.)
Form	partial arc	

Title GAS BEARING DESIGNDate 11-23-76Environmental Estimates

Operating ambient	290°C	(550°F)
Soak back	340°C	(650°F)
Bearing air supply	1.0 to 4.96 bar	(14.7 to 72 psia)
Operating diametral clearance	0.03mm	(0.001 in.)
Rotor speed, maximum	58,500 rpm	
Rotor weight-total	2.04kg	(4.5 lb.)
-at air bearing	0.86kg	(1.89 lb.)
Static bearing loading	5.78kPa	(0.838 psi)

Dynamic Loading (at 90% Speed)

Rotor weight	8.41N	(1.89 lb.)
Rotor unbalance (0.071 N-mm)	31.14N	(7.0 lb.)
Total steady state loading	39.55N	(8.89 lb.)
Radial "g" load (3.5g)	29.45N	(6.62 lb.)
Gyroscopic (2 rad/sec. skid)	167.61N	(37.68 lb.)
Total short time loading	236.61N	(53.19 lb.)
	163.0kPa	(23.6 psi)

Power Loss Due To Friction

Maximum speed, steady state	0.056kW	(0.075 hp)
Air Flow Required to Remove Heat	4.1kg/hr	(9 lb/hr)

Shaft Radial Growth, Calculated

Centrifugal	0.0096mm	(0.00038 in.)
Thermal; 316°C (600°F)	0.056mm	(0.0022 in.)

Materials

Shaft	17-4 PH
Coating	Chrome Carbide
Foils	Inconel X750
Dry Film	H-1284

VIEW IN CIRCLE

M.T.I. HYDRESIL COMPLAINT FOIL BEARING

GAS BEARING DESIGNDate 11/23/76

Bearing Load vs Shaft Motion, Minimum Film Thickness,
 And Frictional Power Loss
 (No End Leakage Correction Factors Applied)

Unbalance Response - Rigid Oil Bearing with Squeeze
 Film Damper and Hydresil Air Bearing System
 Unbalance In Phase

Unbalance Response - Rigid Oil Bearing with Squeeze
 Film Damper and Hydresil Air Bearing System
 Unbalance Out of Phase

CRITICAL SPEED vs FRONT BEARING STIFFNESS

20,000 lb/in Air Bearing Stiffness

20,000 lb/in Support Stiffness At Oil Bearing

INTERSTAGE COMPONENT DATADate 12/17/74

Materials:

Interstage Support: Cast 430 Stainless Steel

Piloting Struts (3): Inconel 713C

Shrouds: 310 Stainless Steel

Vanes: Chrysler Research Material 18D

Drive Parameters:

Nozzle Vane Total Displacement: 1.92 rad (110°)

Nozzle Vane Aerodynamic Torque: .057 N-m (~0.5 in-lbf)

Drive Shaft Design Torque: 7.35 N-m (65 in-lbf)

Actuator Stroke: 64 mm (2.5 in)

Actuator Axial Thrust: 222N (50 lbf)

INTERSTAGE ASSEMBLY

Date 4/30/

* No.	Description	Max. RPM	Material	Face Width In.	No. Of Teeth	Pitch Dia. In.	Press. Angle Deg.	Location
A	Power Turbine Pinion	70,000	SAE 6150 Steel	1.1	28	1.0	20	P.T. Reduction
B	Intermediate Gear	12,405	SAE 6150 Steel	1.0	158	5.65	20	P.T. Reduction
C	Intermediate Pinion	12,405	SAE 6150 Steel	1.2	72	2.59	20	P.T. Reduction
D	Transmission Input Gear	4,701	SAE 6150 Steel	1.1	190	6.79	20	P.T. Reduction
E	Accessory Drive Pinion	4,701	Nodular Iron	0.8	66	4.12	30	Accessory Drive
F	Accessory Drive Gear	3,978	SAE 1010 Steel	0.7	78	4.88	30	Accessory Drive
G	Worm	3,978	SAE 8620 Steel	1.75	6	.84	20	Accessory Drive
H	Worm Gear	265	Manganese Bronze	.62	90	3.99	20	Accessory Drive
I	Regenerator Pinion	265	SAE 1046 Steel	1.0	38	2.71	20	Accessory Drive
J	Regenerator Gear	32	Cast Iron	.62	315	22.50	20	Regenerator
K	V.P.T.N. Sector Gear	--	Stainless Steel	.3	20	1.1	20	Interstage
L	V.P.T.N. Ring Gear	--	Cast Iron	.6	150	8.2	20	Interstage
M	Driver Sprocket	12,405	SAE 1141 Steel	.5	21	2.4	--	P.T. Reduction
N	Driven Sprocket	5,210	SAE 1045 Steel	.5	50	6.0	--	Ancillary, N _{OS}
O	Starter Pinion	6,825	SAE 8640 Steel	.98	16	.44	20	Accessory Drive
P	Starter Idler Gear	4,748	Extruded Bronze	.70	23	.64	20	Accessory Drive
Q	Starter Gear	1,285	SAE 8640 Steel	.83	85	2.36	20	Accessory Drive

Title

GEAR AND SPROCKET DATA

Date 1/23/76

Page 4.12

* See Figures 4.13 - 4.15.2

REDUCTION GEAR SYSTEM

ANCILLARY SHAFT, VEHICLE ACCESSORY SHAFT

Date 2/27/76

REDUCTION GEAR HOUSING OIL PAN FACE

STARTING SYSTEM DRIVES - PRELIMINARY

Date 4/14/75

92 K.W. (123 HP) Gas Turbine with
Single Ceramic Regenerator Tilted 20°
Compact Vehicle Installation

STARTING SYSTEM DRIVES - SIDE VIEW

Page 4.16.1
Date 10/31/75

LUBRICATION SYSTEM - GENERAL

Page 4.17

Date 6/4/75

Supply: Power Stage Driven Gear Pump
Theoretical Displacement, 1.88 Cu. In./Rev.
Rotor Width, 1.18 In.
Regulated Pressure, 40 PSI
Automatic Transmission Fluid, MS4228

Demand: Gas Generator Bearings, 1 GPM
Power Section Bearings,
Accessory Drive 2 GPM
TOTAL 3 GPM

LUBRICATION SYSTEM - FLUID

Date 4/15/75

ENGINEERING OFFICE

MS - 4228

CHRYSLER CORPORATION MATERIAL STANDARDCHANGE - D
ASL Req'd.FLUID - AUTOMATIC TRANSMISSION - HIGH VI
Part No. 2658402A - GENERAL

This standard presents the requirements of a fluid material to be used as a hydraulic medium and lubricant for automatic transmissions.

B - REQUIREMENTS

- 1 - This fluid shall consist of a well refined mineral oil and shall contain such additives as may be necessary to provide a suitable automatic transmission fluid meeting the requirements of this standard. The fluid shall be dyed a distinctive red.
- 2 - This fluid shall contain no additives or impurities detrimental to the materials or operation of an automatic transmission.
- 3 - This fluid shall be free from water, suspended matter and objectionable odors. It shall not form offensive odors during storage or service and shall be non-toxic.
- 4 - Acceptance of this fluid is not contingent upon qualification by any other agency.
- 5 - It shall conform to the following specifications:

Laboratory Procedures

Oxidation Stability ----- Min. 360 hours Chrysler LP 461C-115
before break
point

Viscosity at 210°F,
cSt ----- Min. 7.25 ASTM D-445

Viscosity at -20°F,
cp ----- Max. 2300 CRC L-45

*Viscosity Stability
Viscosity at 210°F Chrysler LP 461C-112
after 8 hours shear
test, cSt ----- Min. 6.00

(Continued)

* Denotes area of change from previous issue.

DATE	EFF. CODE	P C R NO.	CHANGE	RELEASES - CHANGES AND CANCELLATIONS
7-1-70.	2/B	00608-006	D	Par. B-5 revised
9-23-68	2/B	80916-005	C	Standard updated
1-22-64				Date Issued (Dept. 5120)

VOLUME U

PAGE 291(1)

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LUBRICATION SYSTEM - FLUID

Date 4/15/75

ENGINEERING OFFICE

MS -4228

CHRYSLER CORPORATION MATERIAL STANDARD**B - REQUIREMENTS** (Continued)

5 - It shall conform to the following specifications: (Continued)

LABORATORY PROCEDURES

Flash Point, °F -----	Min. 275	ASTM D-92
Pour Point, °F -----	Max. -40	ASTM D-97
Copper Strip Corrosion -----	Max. 3b	ASTM D-130
(1 hour at 300°F)		
Corrosion Protection -----	No Corrosion	Chrysler LP 461H-91
Pentane Insolubles, %		
by Weight -----	Max. 0.01	ASTM D-893
Water Content, % by Volume -----	Max. 0.10	ASTM D-95
Foaming Characteristics		ASTM D-892

	<u>Foaming Tendency</u> <u>after 5 min., ml.</u>	<u>Foam Stability</u> <u>after 5 min., ml.</u>
Sequence 1	Max. 30	nil
Sequence 2	Max. 200	nil
Sequence 3	Max. 30	nil

* Effect on Nylon, MS-DB41, 300 hours at 300°F ASTM D638

Tensile, psi ----- No decrease
 Elongation ----- -20% Max.
 Impact, in/lbs. ----- Min. 40

Effect on Rubber Seals, 70 hours at 300°F ASTM D-471

<u>Seal Material</u>	<u>MS-BZ74</u>	<u>MS-BZ80</u>	<u>MS-BZ81</u>	<u>MS-BZ87</u>
Change in Durometer	±5 pts.	±5 pts.	±5 pts.	-25 pts. Max.
Change in Tensile	-30% Max.	-30% Max.	* -55 % Max.	-20% Max.
Change in Elongation	-50% Max.	-30% Max.	-70% Max.	-10 to +70%
Change in Volume	+1 to +8%	+1 to +8%	* 0 to +10%	+40% Max.

6 - This fluid must pass any additional laboratory or performance tests that may be deemed necessary to insure satisfactory performance of the fluid in the automatic transmissions in which it will be used.

C - CONTROL

1 - All tests shall be conducted according to the procedures designated in paragraph B-5.

(Continued)

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200-36-B (5/67)

VOLUME U

PAGE 291(2)

ENGINEERING OFFICE

MS -4228

CHRYSLER CORPORATION MATERIAL STANDARD

C - CONTROL (Continued)

- 2 - The formulation of approved materials shall not be changed without approval of the Organic Fluids and General Chemistry Department, Chrysler Corporation Engineering Office. For additional information, address inquiries to the Standards Engineer of the above department.
- 3 - Materials purchased according to this standard shall be obtained only from sources which appear on the Chrysler Corporation Engineering Approved Source List. This information may be obtained from the Chrysler Corporation Purchasing Department.

#

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200-36-B (5/67)

VOLUME U

PAGE
291(3)

FRONT COVER OF THE REDUCTION GEARS HOUSING
AND MAIN LUBRICATION LINES.

LUBRICATION SYSTEM-PTW SHAFT

Date 4/16/75

POWER TURBINE HIGH SPEED SHAFT
LAYOUT

(C)

HYDRAULIC SYSTEM - SCHEMATIC

Page 4.18

Date 5/15/75

DRAULIC SYSTEM - FLUIDDate 4/16/75

Engineering and Research Office

Material StandardMS -5931 UChange A

ASL Req'd.

FLUID - HYDRAULIC - STEERING - DUAL PURPOSE

Part No. 4006323

A - GENERAL

- 1 - This standard outlines a quality petroleum oil suitable for use as a hydraulic steering and brake assist fluid.
- 2 - This oil shall be used specifically for factory fill and field service of units fitted with "Hydroboost" systems.

B - REQUIREMENTS

- 1 - This material shall be essentially a well refined mineral oil. Small amounts of anti-wear additives and oxidation inhibitors may be used to meet the detailed requirements in paragraph B-2. Additives, if used, shall be selected to have a minimum effect on seal and hose materials.
- 2 - This oil shall conform to the following specification:

LABORATORY
PROCEDURE

☒ Viscosity at 99°C (210°F), cSt -----	Min. 7.25	ASTM D445
Viscosity Index -----	Min. 150	ASTM D2270
Viscosity at -28.8°C, (-20°F), cPs ---	Max. 6000	ASTM D2983
Pour Point, °C (°F) -----	Max. -42.7 (-45)	ASTM D97
☒ Flash Point, °C (°F) -----	Min. 193 (380)	ASTM D92
Phosphorus, % Wt. -----	Min. 0.02	ASTM D1091
Copper Strip Corrosion -----	Max. 1a	ASTM D130
☒ Aniline Point, °C (°F) -----	90 - 102.8 (195 - 220)	ASTM D611
Heat Test - 96 hours at 148.7°C (300°F) with aeration		Chrysler LP-461C-18
Evaporation Loss % Wt. -----	Max. 4.0	
Viscosity Increase at 99°C (210°F) % --	Max. 3.0	
Petroleum Ether Insoluble % (Wt.) ----	Max. 0.5	

Wear Test

Falex Load -----	Min. 2000 lbs	Chrysler LP-461C-84
Falex Wear (mg) -----	Max. 1.7	Chrysler LP-461C-84

(Continued)

* Denotes area of change from previous issue.

Date	Eff. Code	P C N No.	Change	Releases - Changes and Cancellations
3-4-74			A	PF-Safety shields added, omitted in error
1-22-74				Date Issued (Depr. 5120)

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Engineering and Research Office

Material Standard

MS -5931 U

B - REQUIREMENTS (Continued)

- 3 - The change in the properties of rubber hose and seal materials, after 70 hours immersion in this oil at 148.7°C (300°F) in accordance with ASTM D-471, shall not be greater than the following:

	MS-BZ74	MS-BZ81	MS-EA225
Durometer Hardness, Change -----	±3 points	±10	Max. -15
Tensile Strength, Change -----	Max. 5% loss	Max. 45% loss	Max. 20% loss
% Elongation, Change -----	Max. 25% loss	Max. 25% loss	Max. 5% Inc.
Volume Change % -----	±3	±4	Max. +20%

- 4 - When tested by the procedure outlined in the Rubber Material Standard at 135°C (275°F) and using a minimum of 16 hoses conforming to the indicated Material Standard, this fluid shall give a minimum average hose life as indicated below:

Material Standard	Average Cycle Life
MS-EA225	200,000
MS-EA229	200,000

C - CONTROL

- All tests shall be conducted according to the procedures designated in Paragraph B.
- Containers and shipping invoices must be plainly marked with the Material Standard Number and the Approved Source List Code Number. Containers must be marked in characters not less than 1-1/2 inches high on both the top and side. Proprietary names must not appear on the container.
- The formulation of approved materials shall not be changed without approval of the Organic Fluids and General Chemistry Department, Chrysler Engineering Office. For additional information, address inquiries to the Standards Engineer of the above department.
- Materials purchased according to this standard shall be obtained only from sources which appear on the Chrysler Engineering Approved Source List. This information may be obtained from the Chrysler Corporation Purchasing Department.
- This standard refers to the following standards:

Material Standard MS-BZ74
MS-BZ81
MS-EA225
MS-EA229

#

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Volume U

Page 341(2)

UPGRADED ENGINE SPECIFICATIONS

SECTION 5

AEROTHERMO COMPONENTS

<u>Contents</u>	<u>Page</u>	<u>Revision Date</u>
Compressor - General	5.1	4/8/76
- Operating Characteristics	5.1.1	5/2/75
- Augmented and Overspeed Performance	5.1.2	7/31/75
- Match Point Information	5.1.3	7/22/75
- Design Velocity Triangles	5.1.4	12/19/74
- Thrust Load	5.1.5	1/10/77
- Structural Summary	5.1.6	10/25/76
- Impeller, Radial Diffuser	5.1.7	4/29/77
Burner	5.2	3/10/76
- Combustor Assembly	5.2.1	10/25/76
Vortex	5.3	7/18/75
Compressor Turbine - Stage	5.4	1/10/77
- Eff. Versus Corr. Work	5.4.1	7/22/75
- Corr. Work Versus Flow Speed Param.	5.4.2	7/22/75
- Match Point Information	5.4.3	7/22/75
- Design Velocity Triangles	5.4.4	8/20/74
- Wheel Rotor	5.4.5	3/19/75
- Wheel	5.4.6	10/31/75
- Structural Analysis	5.4.7	10/31/75
- Blade Vibration	5.4.8	5/30/75
- Blade Profile	5.4.9	4/29/77
Power Turbine - Stage	5.5	7/18/75
- Efficiency	5.5.1	7/22/75
- Performance Map	5.5.2	7/22/75
- Vector Diagram Information	5.5.3	7/22/75
- Design Velocity Triangles	5.5.4	12/19/74
- Thrust Loads	5.5.5	7/22/75
- Wheel	5.5.6	10/31/75
- Structural Analysis	5.5.7	10/31/75
- Blade Vibration	5.5.8	1/24/75
- Rotor Assy.	5.5.9	10/15/75
- Nozzle Vane Angle	5.5.10	1/31/77
- Blade Profile	5.5.11	4/29/77
Regenerator - General	5.6	7/9/75
- Drive Mount	5.6.1	7/25/75

UPGRADED ENGINE SPECIFICATIONS

Page 5.1

COMPRESSOR - GENERAL

Date 4/8/76

The compressor consists of a single stage radial impeller with backward curved blades, a bladed diffuser and variable impeller inlet nozzle blades.

Impeller:

Aluminum hub (2618-T61)

Integral steel inducer (17-4 PH)

24 blades

30° backswept blading at exit

Polar moment of inertia 0.000836 kg-m^2 ; Mass 0.624 kg
($0.0074 \text{ in-lb-sec}^2$) (1.38 lbs)

Outer diameter, 166.37 mm (6.55 in.)

Diffuser:

14 Radial vanes

60 Axial vanes

Variable Inlet Guide Vanes:

Modulated pre-whirl angle at 50% gas generator speed for efficient compressor air flow reduction at low output power without reducing engine operating temperature or minimum gas generator speed.

Power augmentation by increasing air flow and pressure by inducing negative pre-whirl at maximum gas generator speed.

13 Vanes; stationary portion at -10° vane angle, rear portion movable:

60° Vane angle pre-whirl at 50% speed

30° Vane angle negative pre-whirl at 100% speed

COMPRESSOR OPERATING CHARACTERISTIC

COMPRESSOR - OPERATING CHARACTERISTIC

COMPRESSOR MATCH POINT INFORMATION

<u>% Ngg</u>	<u>RPM</u>	<u>Mass Flow (Lb/Sec)</u>	<u>VIGV Angle (Deg.)</u>	<u>Inlet Swirl (Deg.)</u>	<u>P_{T IN} (PSIA)</u>	<u>T_{T IN} (°R)</u>	<u>Rotor Total Pressure Ratio</u>	<u>Work Coeff.</u>	<u>H₂O Inj. (Yes/No)</u>
100	58500	1.335	0	0	14.48	545	4.89	.7430	No
100	58500	1.450	-30	-24	14.48	545	5.283	.7407	No
100	58500	1.521	-30	-24	14.48	511*	5.78	.7409	Yes
50	29250	.433	0	0	14.674	545	1.590	.7320	No
50 ⁽¹⁾	29250	.377	0	0	14.674	545	1.597	.7600	No
50 ⁽²⁾	29250	.409	+60	+48	14.674	545	1.525	.7370	No

* Wet Bulb Temperature for 545° Day

(1) Surge Point

(2) Idle Point

COMPRESSOR-DESIGN VELOCITY DIAGRAMS

Date 12/19/74

COMPRESSOR, GAS GENERATOR - THRUST LOADDate 1/10/77

<u>Gas Generator Speed, Ngg Percent</u>	<u>Compressor Thrust, Lbs.</u>	<u>Compressor Turbine Thrust, Lbs.</u>	<u>Net Thrust, Lbs.</u>	
50	- 30.5	10.9	- 19.6	
60	- 51.0	11.5	- 39.5	
70	- 78.5	14.7	- 63.8	
80	-110.0	17.9	- 92.1	
90	-148.0	32.9	-115.1	
95	-164.0	39.5	-124.5	
100	-180.0	70.8	-109.2	(0° VIGV)
100	-196.0	112.8	- 83.2	(-30° VIGV)

UPGRADED ENGINE IMPELLER
STRUCTURAL SUMMARY - CENTRIFUGAL LOAD ONLY

INDUCER:

MATERIAL - 17-4 PH STEEL
 MECHANICAL SPEED - 6126 RAD/SEC (58,500 RPM)
 AVE. TANGENTIAL STRESS - 306 MPa (44.4 KSI)
 BURST MARGIN - 1.59
 BURST SPEED - 9739 RAD/SEC (93,000 RPM)

IMPELLER:

MATERIAL - 2618-T61 ALUMINUM
 MECHANICAL SPEED - 6126 RAD/SEC (58,500 RPM)
 AVE. TANGENTIAL STRESS - 105.5 MPa (15.3 KSI)
 BURST MARGIN - 1.53
 BURST SPEED - 9372 RAD/SEC (89,500 RPM)

LOCATION	ESTIMATED TEMP °C (°F)	*MINIMUM MATERIALS PROPERTIES-MPa (KSI)		MAXIMUM APPLIED DISC STRESS - MPa (KSI)			MAXIMUM APPLIED BLADE STRESS - MPa (KSI)		
		.2% YIELD	ULTIMATE	RADIAL	TANGENTIAL	EQUIVALENT	MIN. PRINCIPAL	MAX. PRINCIPAL	EQUIVALENT
A	66 (150)	875.5 (127.0)	1027.2 (149.0)	24.8 (3.6)	405.4 (58.8)	389.4 (56.5)			
B	52 (125)	881.0 (127.8)	1034.1 (150.0)	44.8 (6.5)	304.0 (44.1)	288.9 (41.9)			
C	116 (240)	281.3 (40.8)	351.6 (51.0)	28.3 (4.1)	231.6 (33.6)	223.4 (32.4)			
D	93 (200)	293.0 (42.5)	364.7 (52.9)	171.0 (24.8)	193.0 (28.0)	125.5 (18.2)			
E	182 (360)	234.4 (34.0)	281.3 (40.8)				95.8 (13.9)	205.4 (29.8)	99.3 (14.4)
F	132 (270)	252.3 (36.6)	346.1 (50.2)				30.3 (4.4)	171.7 (24.9)	125.5 (18.2)
G	66 (150)	875.5 (127.0)	1027.2 (149.0)				139.0 (20.2)	423.3 (61.4)	363.3 (52.7)

* Minimum Properties = 85% Typical Properties

COMPRESSOR-IMPELLER, RADIAL DIFFUSER

Page 5.1.7

Date 4/29/77

Vehicle emission target: 1978 Federal Emission Standards;
0.41 gram/mile HC, 3.4 grams, mile CO, 0.4 gram/mile NOx.

The combustor is a fixed geometry, single stage configuration with a torch-type igniter.

An electronic programmable analog computer control system is used to minimize stoichiometric operation:

- steady stage operation is lean
- transient operation is either very lean or rich, depending upon severity of acceleration.

Non-leaded gasoline is the basic fuel for development but the control system must be able to use any generally available liquid hydrocarbon fuel without substantial effects on the control outputs.

VORTEX

One-piece casting - transition from burner exit oval to first stage turbine inlet annulus.

Material, CRM18D

Wall thickness, 1.91 to 3 mm (.075 to .12 in.)

COMPRESSOR TURBINE-STAGEDate 1/10/77

Nozzle:

15 Vanes

MAR-M-509, MM-002, IN 792+Hf; candidate materials,
diffusion coated

Wheel:

62 Blades

MAR-M-246

Work-speed parameter ($\Delta H/U^2$): 2.1

Design speed: 58,500 rpm

Minimum blade thickness: 0.38 mm (0.015 in.)

Rotor blade taper ratio: 1.65

Rotor inertia: 0.00044 kg-m^2 ($0.0039 \text{ in-lb-sec}^2$)Blade natural frequency: ≥ 1.15 design speed

Maximum root blockage: 15%

Nozzle - outer wheel shroud: One-piece casting

Integral cast ribbed disc

Rib to blade thickness ratio: 1.2

Alternating shallow and deep pockets; 1 per blade

UPGRADED COMPRESSOR TURBINE PERFORMANCE

Efficiency vs. Corrected Work

UPGRADED C.T. PERFORMANCE
EFFICIENCY VS. CORR. WORK

227

UPGRADED ENGINE COMPRESSOR TURBINE PERFORMANCE GOAL

CORRECTED WORK VS. FLOW SPEED PARAMETER

COMPRESSOR TURBINE MATCH POINT INFORMATION

Station	% Ngg	RPM C.T.	Mass Flow (Lbs/Sec)	VIGV Angle (Deg.)	H ₂ O Inj. (Yes/No)	ΔH (BTU/Lb)	T _T °R	P _T PSIA	Pressure Ratio (Nozzle In) (Rotor Out)	η Bld. T-T %
Nozzle In	---	---	1.2961			87.57	2384.7	59.134		
Rotor Out	100	58500	"	0	No		2074.0	29.72	1.99	84.57
Nozzle In	---	---	1.4096			93.63	2384.7	64.29		
Rotor Out	100	58500	"	-30	No		2052.3	30.48	2.11	83.95
Nozzle In	---	---	1.518				2384.7	69.38		
Rotor Out	100	58500	"	-30	Yes	98.72	2037.6	31.10	2.198	83.2
Nozzle In	---	---	.4243				1996.2	21.452		
Rotor Out	50	29250	"	0	No	22.42	1915.0	17.53	1.224	83.21
Nozzle In	---	---	.4013				1975	20.62		
Rotor Out	50	29250	"	+60	No	21.04	1898.7	17.04	1.210	83.51

Compressor Turbine - Match Point Information

Date 7/22/75

Page 5.4.3

COMPRESSOR TURBINE-DESIGN VELOCITY DIAGRAMS

Date 8/20/74

TIP ($R = 2.19''$)

MEAN ($R = 1.9725''$)

HUB ($R = 1.755''$)

	Inertia kg-m ² (in-lb-sec ²)	Weight kg(lb)
Turbine Wheel Disc	0.00044 (0.0039)	.428 (.943)
Integral Shaft	---	.263 (.58)
TOTAL	0.00044 (0.0039)	.691 (1.523)

Compressor Turbine Wheel

UPGRADED ENGINE COMPRESSOR TURBINE WHEEL Structural Analysis Summary

Mechanical Speed - 6130 Rad/Sec (58,500 RPM)
Avg. Tangential Stress - 232,000 kPa (33,600 psi)
Burst Speed - 10,000 Rad/Sec (95,500 RPM)
Burst Factor - 1.63
Material - IN-792 + Hf

COMPRESSOR TURBINE - BLADE VIBRATION

Date 5/30/75

TITLE COMPRESSOR TURBINE - BLADE PROFILE

NOZZLE

ROTOR

TIP

MEAN

HUB

POWER TURBINE - STAGEDate 7/18/75

Interstage:

Area ratio: 2:1

Nozzle:

23 vanes

Variable nozzle vane angle for cycle temperature control and to provide engine braking:

28° Maximum closed

150° Maximum Braking

Wheel:

53 blades

MM-002

Work speed parameter ($\Delta H/U^2$): 1.247

Design speed, power: 46,150 rpm

Design speed, stress: 70,000 rpm

Minimum blade thickness: 0.38 mm (0.015 in.)

Rotor blade taper ratio: 1.88

Rotor inertia: 0.0028 kg-m² (0.024 in-lb-sec²)

Integral cast ribbed disc

Maximum root blockage: 15%

Zero absolute rotor exit swirl

Exhaust Diffuser:

Area ratio: 1.9:1

Power Turbine - EfficiencyDate 7/22/75UPGRADED POWER TURBINE EFFICIENCY

Efficiency Based on Interstage
Diffuser Inlet Total Pressure and
Exhaust Diffuser Discharge Static
Pressure. 3.55:1 Rear Axle

Power Turbine - Performance Map

Date 7/22/75

POWER TURBINE VECTOR DIAGRAM INFORMATION

Station	% Ngg	VIGV Angle (Deg.)	H ₂ O Inj. (Yes/No)	Mass Flow (Slugs Sec.)	RPM	P _T (Lbs/Ft ²)	T _T (°R)	rVu * (Ft ² /Sec)	77 Bld. Row T-T (%)	77 Interstage In Rotor Out T-T (%)	77 Interstage In Exh. Diff. Out T-S (%)
Stator In	100	0	No	.04028	---	4134.04	2074	36.3	86.88	81.83	75.96
Rotor In	"	"	"	"	46150	4020.17	"	353.51			
Rotor Out	"	"	"	"	"	2447.44	1854.7	39.48			
Stator In	100	-30	No	.04382	---	4207.06	2052	60.58	87.3	81.25	74.9
Rotor In	"	"	"	"	46150	4101.06	"	342.50			
Rotor Out	"	"	"	"	"	2476.80	1831.4	26.72			
Stator In	100	-30	Yes	.047293	---	4295.91	2037.6	76.80	87.54	79.29	72.61
Rotor In	"	"	"	"	46150	4201.93	"	328.94			
Rotor Out	"	"	"	"	"	2540.16	1812.48	14.62			
Stator In	50	0	No	.01319	---	2505.36	1915.0	8.97	66.58	63.27	59.97
Rotor In	"	"	"	"	18000	2470.30	"	87.56			
Rotor Out	"	"	"	"	"	2175.84	1870.9	22.25			

* Positive in the direction of rotation.

POWER TURBINE-DESIGN VELOCITY DIAGRAMS

Date 12/19/74

Power Turbine - Thrust Loads

UPGRADED ENGINE POWER TURBINE WHEEL Structural Analysis Summary

Mechanical Speed - 7330 Rad/Sec (70,000 RPM)
Avg. Tangential Stress - 440,000 kPa (63,700 psi)
Burst Speed - 8,800 Rad/Sec (84,000 RPM)
Burst Factor - 1.20
Material - NM-002

A-926 POWER TURBINE
BLADE VIBRATION
Campbell Diagram

POWER TURBINE ROTOR ASSEMBLY

POWER TURBINE - NOZZLE VANE ANGLEDate 1/31/77Nozzle Angle at Design ConditionsGas Generator
Speed, PercentNozzle Vane
Angle, Degrees

50	30.2	(+60° VIGV)
50	29.1	
60	29.6	
70	30.2	
80	31.9	
90	33.4	
95	34.6	
100	36.5	
100	38.2	(-30° VIGV)

POWER TURBINE - BLADE PROFILE

NOZZLE

TIP

ROTOR

MEAN

HUB

APPROX. 1.5x SIZE

PROVISIONAL

- I. Vendor:
- II. Overall Physical Dimensions
 - A. Diameters:
 1. Overall O.D.: 540 mm (21.25 in.)
 2. Effective Matrix O.D.: 502 mm (19.75 in.)
 3. Effective Matrix I.D.: 77 mm (3.00 in.)
 4. Hub I.D.: 36 mm (1.40 in.)
 - B. Length: 83 mm (3.25 in.)
- III. Matrix Characteristics:
 - A. Vendor Designation:
 - B. Material:
 - C. Passage Geometry:
 1. Shape: Triangular
 2. Hydraulic Diameter: 0.46 mm (.018 in.)
 3. Stock Thickness: .089 mm (.0035 in.)
 4. Open Frontal Area: 68%
 - D. Fanning Friction Factor: $f = 8.15 \text{ Re}^{-.9322}$
 - E. Colburn Modulus: $j = 1.26 \text{ Re}^{-.8380}$
- IV. Support: Center axle rotating on contoured graphite bearing
- V. Drive Gear Mount: Elastomeric (See 5.6.1)

APPENDIX C

PROGRAM PLAN PROGRESS CHARTS

BASELINE GAS TURBINE DEVELOPMENT PROGRAM PLAN

TASK/DESCRIPTION	1976				1977												1978			
	O	N	D	J	F	M	A	M	J	J	A	S	O	N	D	J	F	M		
Task 1 Furnish 7 Baseline Engines	(Completed April 1975)																			
Task 2 Furnish 3 Turbine Powered Vehicles	(Completed July 1974)																			
Task 3 Provide Service & Material Support for Program																				
Task 3.1 Support to ERDA (O=Major Shows)	O O O																			
Task 3.2a NASA Baseline Support																				
Task 3.2b NASA Upgraded Test Rigs																				
Task 4 Baseline Engine Endurance Tests	(Completed May 1976)																			
Task 5 Component Development & Evaluation	(Completed Except for Task 5.7b)																			
Task 5.7b Continuously Variable Hydromechanical Transmission																				
• Hydraulic Pump & Motor Design	(Completed December 1975)																			
• Procure & Assemble	(Completed June 1976)																			
• Evaluate Hydraulic Unit @ MTI																				
• Chrysler Evaluation & Transmission Definition																				
• Transmission Analysis & Report																				
Task 6 Verify Task 5 Improvements in Vehicles	(Completed July 1976)																			
Task 7 Furnish Upgraded Engines																				
Task 7.1 Design & Specify Upgraded Engine	(Completed August 1976)																			
Task 7.2-7.4 Build & Checkout 10 Upgraded Engines																				
• Procure & Inspect Parts	1&2 3 4																			
• Build 10 Engines																				
• Conduct Reference Tests																				

254

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TASK 7.5 (Con't)

- 6. Determine Parasitic Losses
 - a. Rib Wheel Pumping, Bearings and Accessory Drive
 - 1. Prepare P.T. Rotor: a) No Disc; b) No Blades
 - 2. Engine test at temperature driving with motoring dynamometer
 - 3. Analyze and project to C.T. wheel
 - b. Impeller Backface Pumping
 - 1. Conduct design analysis
- 7. Contingency Efforts (As required per above results)

B. CORRECT FOR HIGH FUEL FLOW

1. Develop Regenerator Flow Uniformity
 - a. High pressure side flow system (Station 3 to 4)
 1. Assemble cold flow rig (Spencer blower, plus engine parts)
 2. Test and develop required baffling
 - b. Low pressure side flow system (Exhaust diffuser plus station 8 to 9)
 1. Develop ducting for optimum diffuser recovery and flow uniformity on dynamic similarity cold test rig (Creare, subcontract)
 - c. Incorporate baffling/ducting into engine and evaluate
2. Bulkhead Cooling
 - a. Reduce cooling (most areas 100-200°F cooler than design)
 1. Devise baffling per thermal mapping tests
 2. Adapt and test

C. CORRECT FOR OTHER SYSTEM DEFICIENCIES

1. Burner (Reduce flashback and torch, carboning; evaluate coated and ceramic parts)
2. Control
 - a. Power turbine nozzle actuator (build and checkout redesign)
 - b. Controller (optimize programming logic)
3. Regenerator (simplify processing of flat crossarm seals)

4 Additional Component Development Tasks were conducted from 1974 through 1976.

BASELINE GAS TURBINE DEVELOPMENT PROGRAM PLAN

5

TASK/DESCRIPTION	1976						1977												1978		
	O	N	D	J	F	M	A	M	J	J	A	S	O	N	D	J	F	M			
Task 8 Furnish Upgraded Vehicles																					
Task 8.1 Design Vehicle Installation of Upgraded Engine, Cost Estimate (Completed November, 1976)																					
Task 8.2 Procure & Modify 6 Vehicles, Install Upgraded Engine Checkout																					
• Procure 6 Compact Vehicles (3 procured in 1976)																					
• Procure Parts																					
• Modify Chassis & Body																					
• Develop & Procure Vehicle Systems																					
• Final Assembly & Checkout																					
Task 8.3 Conduct Vehicle Test & Deliver																					
• Measure Performance & Emissions																					
• Conduct Ambient Tests																					
• Run Endurance Tests & Deliver																					
Task 9 Prepare Reports & Reviews																					
• Plan of Performance																					
• Monthly Reports																					
• Quarterly Reports																					
• Final Report																					
• Oral Presentation & Documentary Film																					

BASELINE GAS TURBINE DEVELOPMENT PROGRAM PLAN

6