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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

SEMIANNUAL TECHNICAL PROGRESS REPORT FOR PERIOD: JANUARY 1 - JUNE 30, 1983

Mechanical Technology Incorporated

August 1983

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN 3-32

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U.S. DEPARTMENT OF ENERGY
Conservation and Solar Applications
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Semiannual Technical Progress Report

Period Covered:

January 1 - June 30, 1983

Approved By: N.P. Nightingale 8/30/83
Noel P. Nightingale
ASE Program Manager

Approved/Ratified By NASA: William K. Tabata 8/31/83
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In March, 1978, a Stirling-engine development contract, sponsored by the Department of Energy and administered by NASA/Lewis Research Center, was awarded to Mechanical Technology Incorporated (MTI) for the purpose of developing an automotive Stirling engine and transferring Stirling engine technology to the United States. The program team consisted of MTI as prime contractor, contributing their program management, development, and technology-transfer expertise; United Stirling of Sweden (USAB) as major subcontractor for Stirling-engine development and AM General (AMG) as major subcontractor for engine and vehicle integration.

Most Stirling-engine technology previously resided outside of the United States, and was demonstrated for stationary and marine applications; therefore, the Automotive Stirling Engine (ASE) Development Program was directed at the establishment and demonstration of a base of Stirling engine technology for automotive application by September, 1984. The high-efficiency, multifuel capability, low-emissions, and low-noise potential of the Stirling engine made it a prime candidate for an alternative automotive propulsion system.

ASE Program logic called for the design of a Reference engine to serve as a focal point for all component, subsystem, and system development within the program. The Reference Engine System Design (RESD) was defined as the best-engine design generated at any given time within the program that would provide the highest possible fuel economy and meet or exceed all other program objectives while utilizing all new technologies that are reasonably expected to be developed by 1984 and are judged to provide significant improvements relative to the risk and cost of their development.

The Mod I and Mod II engines are experimental versions of the RESD. The Mod I was the first engine design that used existing technologies embodied in USAB's P-40 and P-75 engines. The Mod II (slated to be designed based on the RESD, the Mod I, and component and technology development improvements) is an engine design directed at meeting the final ASE Program objectives.

In March, 1981, the RESD was updated to predict a combined mileage of 41.1 mpg on unleaded gasoline (55% above projected internal-combustion-engine mileage) for a 1984 X-body vehicle with a curb weight of 2870 pounds; however, because of Government funding cutbacks, the Mod II design and associated development efforts were never started, making the Mod I the only experimental engine in the ASE Program.

Since the Mod II has been postponed, it was reasoned that the Mod I could be used to develop and demonstrate RESD technology through a design upgrade. As a result, the "Proof-of-Concept" logic evolved, and the Upgraded Mod I engine emerged as an improved engine system. Specific design goals were established for the Upgraded Mod I directed at demonstrating or "proving" the design "concepts" of the RESD. Inherent limitations were, however, recognized in the proof-of-concept approach, since Mod I hardware was larger and, in some cases, of a fundamentally different design than the RESD.

During this semiannual report period, the ASE Program directed its resources at the fabrication, assembly, and initial characterization of Upgraded Mod I engine systems, component and technology development, and RESD design substantiation. Mod I engine endurance life testing was also initiated. As a result of an intensive value-engineering effort, the RESD was updated incorporating major design changes, resulting in a more competitive manufacturing cost while maintaining the overall program objective of a 30% better fuel economy than a comparable internal-combustion engine. A design review of this updated RESD was held in May, 1983

The four current ASE Program Mod I engines have reached a total of 2523 operational hours, while two Upgraded Mod I engines have accumulated 166 hours, for a program total of 2689 hours. The AMC Lerma vehicle has accomplished 406 starts and has run for 298 hours of transient operation.

During the past six months, the ASE Program has been expanded to support an Industry Test and Evaluation Program (ITEP). NASA/Lewis Research Center has obtained agreements with specific industry companies (General Motors, John Deere, and Cummins) to test Mod I engines. Plans have been finalized to build and deliver two engines for the ITEP; delivery is scheduled for early 1984. The procurement of engine hardware and spares has been initiated, as well as the preparation of training documents.

I. SUMMARY

Since the inception of the ASE Program in 1978, 13 Quarterly Technical Progress Reports have been issued under NASA Contract No. DEN3-32, "Automotive Stirling Engine Development Program;" however, reporting was changed to a semiannual format in July, 1981. This report, the fourth Semiannual Technical Progress Report issued under the contract, and covering the period of January 1 to June 30, 1983, includes technical progress only. Although the program has been modified to a proof-of-concept format, the objectives described below still apply to the RESD. The upgraded version of the Mod I engine is not, however, required to demonstrate all these hardware objectives.

Overall Program Objectives

The overall objective of the ASE Program is to develop an automotive Stirling Engine System (SES) by September, 1984 which, when installed in a late-model production vehicle, will:

- demonstrate at least a 30% improvement in combined metro/highway fuel economy over that of a comparable spark-ignition, engine-powered production vehicle, based on EPA test procedures*; and,
- show the potential for emissions levels less than: NOx = 0.4 g/mi, HC = 0.41 g/mi, CO = 3.4 g/mi, and a total particulate level of 0.2 g/mi after 50,000 miles.

In addition to the above objectives, which are to be demonstrated quantitatively, the following system design objectives are also considered:

- ability to use a broad range of liquid fuels from many sources, including coal and shale oil;
- reliability and life comparable to current-market powertrains,
- a competitive initial cost and life-cycle cost comparable to a conventionally powered automotive vehicle;
- acceleration suitable for safety and consumer considerations; and,
- noise/safety characteristics that meet the currently legislated or projected Federal Standards for 1984.

Major Task Descriptions

The overall objectives of the major program tasks are described below as modified for the proof-of-concept program:

<u>Task 1 - Reference Engine</u> - This task, intended to guide component, subsystem, and engine system development, involves the establishment and continual updating of an RESD, which will be the best engine design that can be generated at any given time, and that can provide the highest possible fuel economy while meeting or exceeding other final program objectives. The engine will be designed for the requirements of a projected reference vehicle that will be representative of the class of vehicles for which it might first be produced, and it will utilize all new technology (expected to be developed by 1984) that is judged to provide significant improvement relative to the risk and cost of its development.

<u>Task 2 - Component/Technology Development</u> - Guided by RESD activities, this task will be conducted in support of various Stirling engine systems, and will include conceptual and detailed design/analyses, hardware fabrication and assembly, and component/subsystem testing in laboratory test rigs. When an adequate performance level has been demonstrated, the component and/or subsystem design will be configured for in-engine testing and evaluation in an appropriate engine dynamometer/vehicle test installation. The component development tasks, directed at advancing engine technology in terms of durability/

*Automotive Stirling and spark-ignition engine systems will be installed in identical model vehicles that will give the same overall vehicle driveability and performance.

reliability, performance, cost, and manufacturability, will include work in the areas of combustion, heat exchangers, materials, seals, engine drivetrain, controls, and auxiliaries.

Task 3 - Technology Familiarization (Baseline Engine) - The existing USAB P-40 Stirling engine will be used as a baseline for familiarization, as a test bed for component/subsystem performance improvement, to evaluate current engine operating conditions/component characteristics, and to define problems associated with vehicle installation. Three P-40 engines will be built and delivered to the United States' team members; one will be installed in a 1979 AMC Spirit. A fourth P-40 engine will be built and installed in a 1977 Opel sedan for testing in Sweden. The baseline P-40 engines will be tested in dynamometer test cells and in the automobiles. Test facilities will be planned and constructed at MTI to accommodate the engine test program and required technology development.

Two Mod I engines will be procured, assembled, and tested for delivery to automotive companies, who will provide independent test and evaluation.

Task 4 - Mod I Engine - A first generation automotive Stirling engine (Mod I) will be developed using USAB P-40 and P-75 engine technology as an initial baseline upon which improvements will be made. The prime objective will be to increase power density and overall engine performance. The Mod I engine will also represent an early experimental version of the RESD, but will be limited by the technology that can be confirmed in the time available. The Mod I need not achieve any specific fuel economy improvements. It will be utilized to verify concepts incorporated in the RESD, and to serve as a stepping stone toward the Mod II engine, thus providing an early indication of the potential to meet the final ASE Program objectives.

Three engines will be manufactured in Sweden and tested in dynamometer test cells to establish their performance, durability, and reliability. Continued testing and development may be necessary to meet preliminary design performance predictions. One additional Mod I engine will be manufactured, assembled, and tested in the United States.

A production vehicle will be procured and modified to accept one of the above engines for installation. Tests will be conducted under various steady-state, transient, and environmental conditions to establish engine-related driveability, fuel economy, noise, emissions, and durability/reliability.

The Mod I engine will be upgraded through design improvements to provide a "proof-of-concept" demonstration of selected advanced components defined for the RESD.

Task 5 - Mod II Engine - Postponed

Task 6 - Prototype Study - Postponed.

Task 7 - Computer Program Development - Analytical tools will be developed that are required to simulate and predict engine performance. This effort will include the development of a computer program specifically tailored to predict SES steady-state cyclical performance over the complete range of engine operations. Using data from component, subsystem, and engine system test activities, the program will be continuously improved and verified throughout the course of the program.

Task 8 - Technical Assistance - Technical assistance will be provided to the Government as requested.

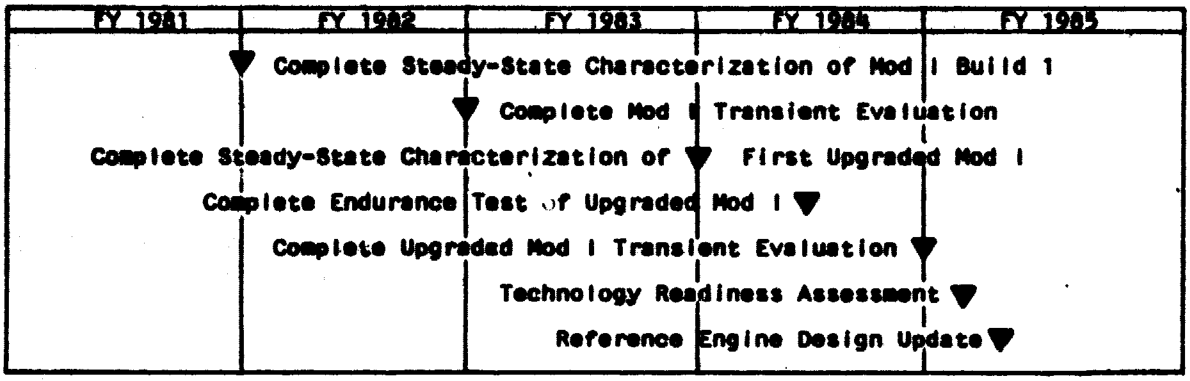
Task 9 - Program Management - Work under this task will provide total program control, administration, and management, including reports, schedules, financial activities, test plans, meetings, reviews, and technology transfer.

Program Schedule

A current schedule of the major milestones for the ASE Program is presented on the following page.

Stirling Proof-of-Concept Program

Milestones



Program Status and Plans

A summary of the accomplishments achieved in the ASE Program during this semiannual report period are presented in the following sections.

Component and Technology Development

The development of a dual-orifice Delavan fuel nozzle was discontinued after tests in the vehicle showed excessive carbon buildup and the need for a complex fuel control for emissions. As a result, emphasis has been placed on the development of an improved air-atomized nozzle having a number of advantages such as wide turndown, multifuel capability, and simple fuel control. Disadvantages include the need for an atomizing air compressor, a negative impact of cold atomizing air on the External Heat System (EHS) efficiency, and the potential to plug if small-diameter fuel orifices are used. These disadvantages originally influenced the decision to attempt the development of a dual-orifice Delavan nozzle; however, it is felt that reducing the required atomizing air, the impact on EHS efficiency can be reduced considerably from two percentage points to a negligible amount. By using a single large orifice in the nozzle, the tendency to form carbon can be eliminated.

As a result, both existing and new-designed air-atomized nozzles were reviewed during this semiannual report period. Spray Rig tests were conducted on nine different types of fuel nozzles. Based on the spray results, selected nozzles were evaluated in a Combustion Performance Rig. In order to provide a direct comparison of nozzle performance, all tests were conducted using a Mod I exhaust gas recirculation (EGR) combustor with no gas recirculation. Two versions of a radial air-atomized nozzle showed encouraging results. Heater tube temperature variations were comparable to the Mod I nozzle at fuel flows less than 2 g/s, and Lambdas (λ)* greater than 1.2. NOx emissions levels were lower than Mod I baseline data, and the nozzle demonstrated carbon-free operation during eleven hours of rig testing.

In order to reduce the manufacturing cost of the engine preheater, the use of ceramic material has been under consideration. After consulting with several ceramic vendors, an existing counterflow recuperative heat exchanger was selected for possible use as a preheater. A cordierite block manufactured by Coors Porcelain Company was chosen because its anticipated performance closely matched design requirements. Sample blocks were evaluated in a Heat-Transfer Rig using cold air on one side and heated air on

*air/fuel ÷ air/fue stoichiometric

the other. Both heat transfer and pressure loss were found to be comparable to existing metallic preheater performance. Durability of the test sections was acceptable up to and including 800°C inlet air temperature. A program has been initiated with Coors to supply as many as eight test sections (single blocks of the RESD design) for rig performance and durability tests. One of the ceramic blocks will be chosen for evaluation in a complete engine system during 1984.

As described in the previous semiannual report*, P-40 engines have been assigned for test to provide statistical data on the life of Pumping Leningrader (PL) main rod seals. Initial plans were to use three engines; however, two engines were finally utilized to reduce costs. These P-40 engines were modified to duplicate the Mod I seal (except for stroke) as closely as possible. Each engine was tested according to a prescribed duty cycle** to properly simulate cold starts and a high-pressure/velocity environment. If each set of seals (PL seals made from HABIA material) completed 500 hours of testing, they were to be removed, inspected, and replaced. Twelve seals completed 500 hours without failure or degradation in engine performance. During this period, however, premature seal failures were experienced on some Mod I engines. Since the Mod I seals were made from the same batch of HABIA material as used in the P-40 engines, it was felt that the difference in P-40 life versus Mod I life was not due to the material, but rather the basic design.

Figure 1-1 shows the demonstrated seal life in Mod I engines. The life is erratic - note that long life was demonstrated on certain engine builds, and very short life on others. In comparing the duty cycle used on the P-40 engine to the normal testing environment of Mod I development engines, it was felt that Mod I engines operate for longer periods at high power levels, and experience more frequent starts and stops. As a result, the duty cycle for the P-40 life engines was changed to a more accelerated one calling for longer operating times at high pressures and speeds and more frequent starts and stops***. During a 24-hour period, the seals experience two hot starts, two cold starts, and more than 16 hours at 12 MPa and 2500 rpm. At this point in time, one set of seals (quantity 4) had completed 500 hours without failure. Mod I engine No. 2 is scheduled for testing to this accelerated duty cycle during the latter half of 1983 to help identify any unique design difference that may exist between P-40 and Mod I seal hardware.

Continued development of a digital combustion Air/Fuel Control (AFC) System during this report period gave very encouraging results. An electronic AFC with low pressure drop and low minimum fuel flow is being developed to replace the existing Bosch K-Jetronic system. A digital AFC will allow improved λ control and, therefore, lower emissions levels across the entire engine operating range.

In May, a digital AFC was tested with the current Bill-of-Materials (BOM) air-atomized fuel nozzle on Mod I engine No. 1 in the Lerma Transient Test Bed (TTB). The results were very comparable to the emissions and fuel economy obtained with the Bosch K-Jetronic system. The tests were conducted simply to attempt to duplicate the performance of the Bosch K-Jetronic unit. No effort was made to make use of two potentially large advantages of the digital system, i.e.; the ability to vary the air-to-fuel ratio as a function of engine parameters other than just airflow, and the ability to initiate changes in airflow and fuel flow with appropriate delays to coordinate the changes in airflow/fuel flow rates to occur simultaneously at the fuel nozzle. Future tests to develop these attributes are scheduled in the TTB during the latter half of 1983. Currently, steady-state tests are being conducted on Upgraded Mod I engine No. 10.

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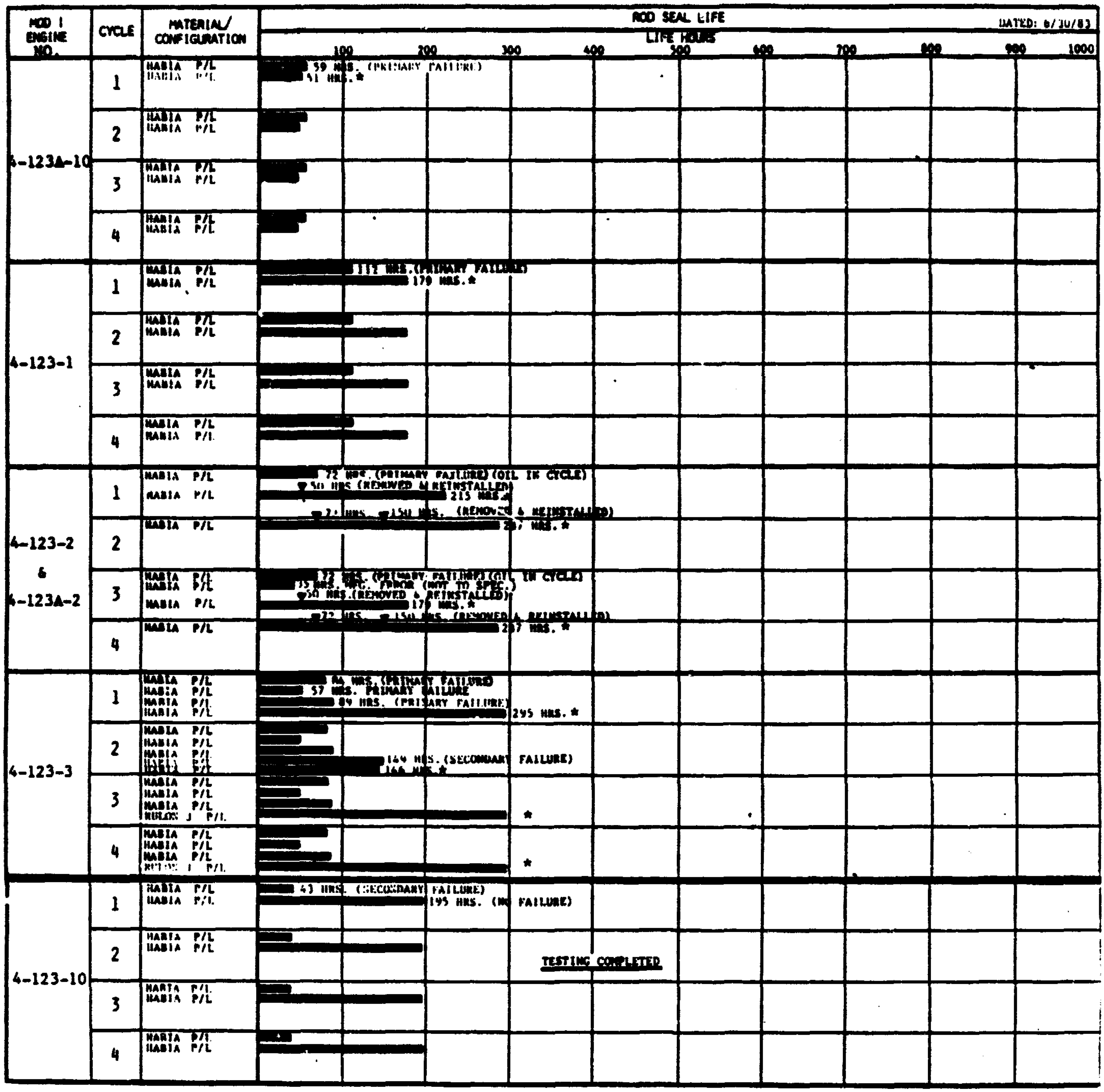
*MTI Report No. 83ASE308SA3
**Figure 3-43 shows the duty cycle used.
***Figure 3-44 shows the new accelerated duty cycle used.



*Still Testing

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Fig. 1-1 Demonstrated Seal Life in Mod I Engines

Reference Engine System Design (RESD)

As stated previously, the RESD represents the best engine design that meets ASE Program objectives while utilizing the technologies expected to be developed by the end of the program. Since the last RESD update in 1981, an intensive cost-reduction effort has been ongoing in an attempt to reduce the estimated manufacturing cost of the RESD without a sacrifice in performance. The results of this effort culminated in a design review update in May, 1983.

The manufacturing cost of the RESD was reduced from $1651 per unit to $1238 per unit. Major changes were made in the design; notably, going from a 'U'-drive, two-crankshaft arrangement to a 'V'-drive system, and from a canister hot engine system to an annular configuration. Significant changes were also made in the control and auxiliaries designs. A rotary power-control valve utilizing an electric motor as a power source replaced the current linear valve design, which calls for an oil-servo system.

Similarly, the current centrifugal air blower system, which uses a variator calling for an oil-servo system, was replaced by a lobe positive-displacement, variable-stroke blower design. A double-acting hydrogen compressor was incorporated to reduce the size of the start-up motor and, in turn, was combined with the starter motor to make a single unit. Ceramic material was used in the preheater, and as a thermal-barrier coating on the combustor liner.

At this point in time, analysis shows that the projected performance of the RESD has been reduced as a result of these design changes. Table 1-1 compares the 1981 RESD to the revised design, the engine efficiencies at several key operating points, and vehicle mileage projections.

In order to improve the engine performance, new technology development efforts are planned and ongoing in the area of annular heater heads and improved auxiliaries designs.

Table 1-1

Reference Engine System Design Comparison

Configuration	Current RESD	1981 RESD
Hot Engine System Engine Drive System Power-Control System Air Blower Auxiliaries	Annular 'V'-Drive Rotary Valve, Double-Acting Compressor Lobe, Positive-Displacement Variable-Stroke Electric Motor	Canister 'U'-Drive Linear Valve, Single-Acting Compressor Centrifugal with Variator Oil-Servo Powered
Manufacturing Cost (300,000 units/yr.)	$1238 (1983 dollars)	$1651 (1983 dollars)
Power Density	4.7 lb/hp	4.9 lb/hp
Max Power Point (15 MPa) Power Efficiency	60 kW (80 hp) (at 4000 rpm) 34.6%	60.1 kW (80.6 hp) (at 4000 rpm) 34.2%
Max Efficiency Point (15 MPa) Power Efficiency	25.5 kW (34.2 hp) (at 1350 rpm) 42.2%	22.1 kW (29.6 hp) (at 1100 rpm) 43.3%
Part-Power Point (5 MPa) Power Efficiency	12 kW (16 hp) (at 2000 rpm) 37.3%	12.2 kW (16.3 hp) (at 2000 rpm) 37.7%
Weight	173 kg	180 kg
Combined Fuel Economy* Unleaded Fuel Diesel Fuel	40.1 mpg 46.1 mpg	41.1 mpg 47.3 mpg

*2870-lb. curb weight GMC X-Body vehicle

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Upgraded Mod I

The Mod I engine design was upgraded to support the proof-of-concept program, whereby RESD technologies were to be evaluated in the Mod I engine system. As a result, the Upgraded Mod I provides an engine design capable of demonstrating a logical growth toward the RESD, and offers improvements in engine performance, cost, weight, reliability, and durability. Specific design goals established for the Upgraded Mod I have been described and compared to the Mod I and RESD in MTI Report No. 82ASE278SA2.

Table 1-2 summarizes the Upgraded Mod I design goals.

Table 1-2

Upgraded Mod I Design Goals

Power	69.0 kW	92.5 hp
Weight	263 kg	580 lbs
Specific Weight	4.5 kg/kW	7.5 lb/hp

Net Brake Efficiency	
Maximum	39%
Part-Power	33%

A design review of the Upgraded Mod I was held in February, 1983. Highlights of this engine configuration include:

- EHS preheater diameter reduction, and fewer and thinner heat exchanger plates;
- further part-power optimization was incorporated into the design of the Hot Engine System (HES), thus reducing the regenerator diameter;
- the heater head metal temperature was designed for 820°C using XF-818 casting material and CG-27 tube material;
- the cold connection ducts and cylinder liners of the Cold Engine System (CES) were integrated;
- the Engine Drive System (EDS) utilized a lighter weight design and roller bearings;
- the Power-Control System's blocks were redesigned for lighter weight, and the overall tubing arrangement/bracketry of the system was simplified; and,
- the oil-servo system was improved with simplified mounting bracketry, and the air-blower bearing and drive arrangement was redesigned for increased life.

Figures 1-2 and 1-3 show cross sections of the Upgraded Mod I and Mod I engines (respectively) for comparison purposes.

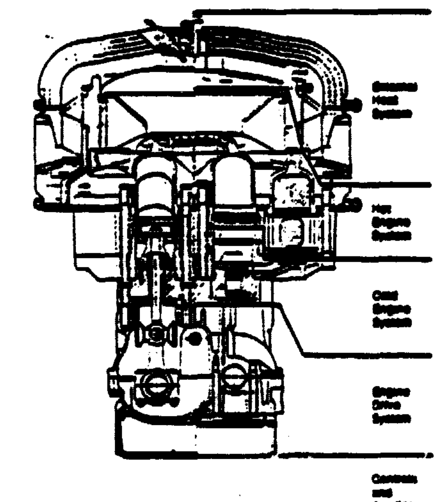

Fig. 1-2 Upgraded Mod I Engine Cross Section

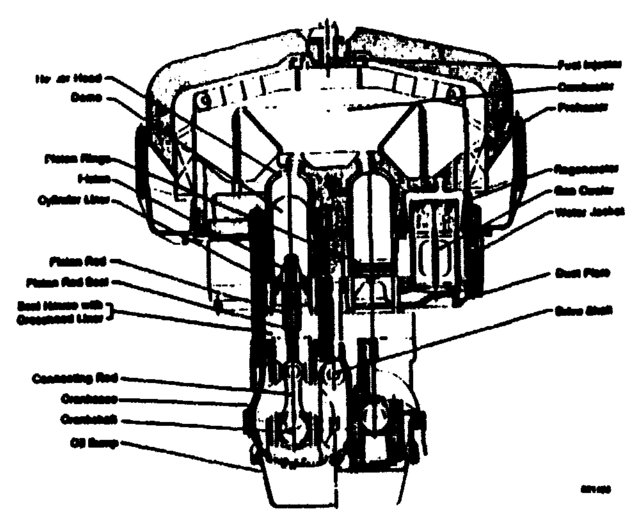

Fig. 1-3 Mod I Engine Cross Section

Mod I Engine Test Program

The four Mod I engines undergoing test in the ASE Program are identified as engines No. 1, 2, 3, and 10. During the past six months, engines No. 2 and No. 10 have been retrofit to an Upgraded Mod I configuration. In order to properly identify these engines from the original designs, they have been designated as 4-123A-2 and 4-123A-10, respectively. As shown in Figure 1-4, a total of 2689 test hours have been accumulated to date on all Mod I and Upgraded Mod I engines. Figure 1-5 shows an engine-by-engine summary of test hours to date.

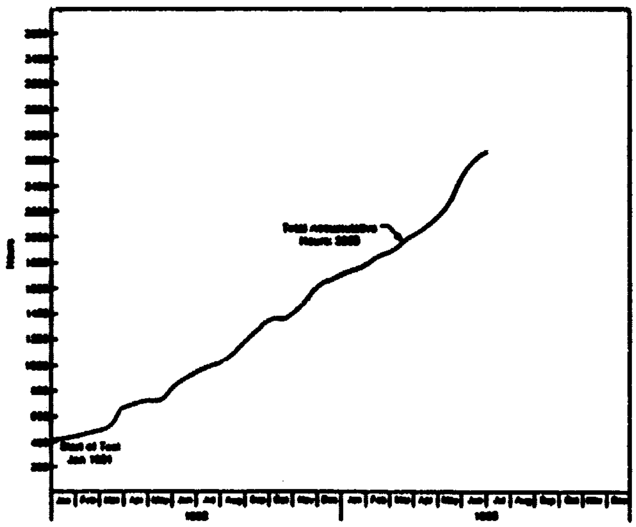

Fig. 1-4 Mod I Engine Test Hours as of June, 1983

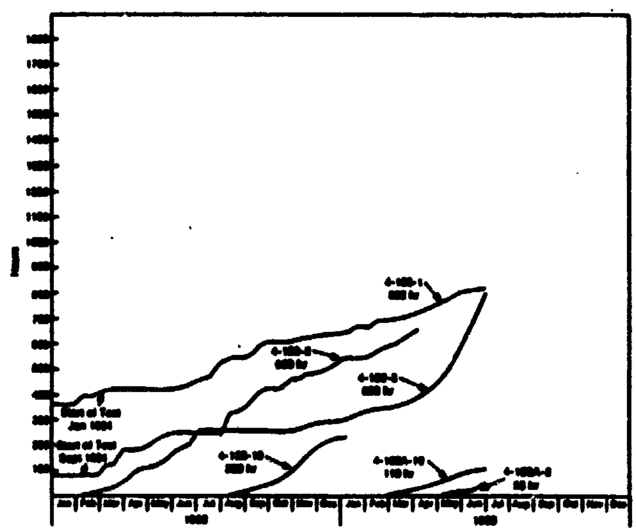

Fig. 1-5 Mod I Engine Test Hours by Engine as of June, 1983

Mod I engine No. 1, installed in the TTB Lerma vehicle, has accumulated 822 total test hours. The engine has experienced 644 stops and starts, and the vehicle has logged a total of 3403.73 miles. During this report period, a significant improvement in fuel economy was attained on the Lerma over that reported in the last semiannual report. Combined mileage using unleaded fuel was increased from 23.5 to 24.5 mpg. The comparable combined mileage for an AMC Spirit is 22.3 mpg. Changes made in the TTB were the installation of a high-stall torque converter and improved shift schedule. The engine oil-servo system was improved to provide steady operating pressure throughout the driving cycle, which is believed to have provided more stable control system operation. Also, during the 10-minute hot soak period, the air blower was shut off to minimize the cooling effect on the engine. During these latest CVS tests, 17 urban cycles and 28 highway cycles were completed, with more than 45 hours spent on the vehicle dynamometer. The acceleration of the Lerma, which is a 3750-lb vehicle, for 0-60 mph was measured at 25.7 seconds.

Mod I engine No. 3, as reported last, was being assembled to begin a 2000-hour endurance test program. During the past six months, the engine has successfully completed 100 starts and stops, and has started endurance testing. From the total time plot, it can be seen that engine No. 3 has accumulated a total of 803 test hours, 400 of which have been completed during the endurance test program. The duty cycle used is an extremely harsh cycle intended to accelerate any fatigue failures and expose the auxiliaries to high speeds and rapid changes in speed.

Both Mod I engines No. 2 and 10 have been retrofit to the upgraded configuration. While undertaking this, care was taken to make the designs interchangeable with existing Mod I hardware, thus making it possible to test and evaluate each system if necessary. This approach was used on Upgraded Mod I engine No. 2 as it was assembled initially with an Upgraded Mod I EHS (all remaining hardware was of the Mod I configuration). The engine was then fit with an Upgraded Mod I HES, integrated cold connecting duct, and cylinder liner. The engine, configured in this manner at the conclusion of this report period, was testing as a Basic Stirling Engine (BSE),* and has accumulated 56 test hours as an Upgraded Mod I, which was defined as the point in time when the Upgraded Mod I HES was installed.

The approach used in retrofitting Mod I engine No. 10 with Upgraded hardware was totally opposite from that used on engine No. 2. Engine No. 10 was completely disassembled and rebuilt as a Stirling Engine System (SES)** with Upgraded Mod I hardware. A roller-bearing drive system, integrated cold connecting duct and cylinder liners, HES, EHS, Power-Control System, and auxiliaries*** were installed, and the engine tested. The engine has accumulated 110 test hours as an Upgraded Mod I engine system since testing began in February.

Hardware development problems and poor engine performance were experienced on both Upgraded Mod I engines from the very beginning of testing. In the EHS, wide variations in combustion gas temperatures after the second row tubes and in the exhaust were observed. The heat absorbed by the HES was observed to be less than experienced in the Mod I, and thermal distortion was experienced in the preheater cover. In the HES, backing rings between the cylinder and cylinder liner were found to be melting after less than ten hours of testing. Regenerators rotated within the regenerator housings, wearing out the top surfaces of the coolers. During this early development phase, the engines were tested at 720°C, although they will eventually be tested at the design temperature of 820°C.

On Upgraded Mod I engine No. 10, the measured maximum engine power was 48.4 kW (64.9 hp), which was 10 kW (13.4 hp) below predictions. The measured maximum efficiency was 30.5%, which was seven percentage points below predictions. The distorted combustion gas temperatures were found to be caused by manufacturing flaws in the preheater matrix which induced a malflow distribution in the inlet air to the system that carries through the preheater heat exchanger process, combustion system, and exhaust system. Incorporation of a uniform preheater rectified this problem.

The heater heads on both engines were found to have excessive gaps in two major areas which allowed combustion gases to bypass the head and go directly to the exhaust. A shim stock strip was placed between the ends of the back-row tubes on each quadrant to close the gap between quadrants. A second problem area was a gap between the regenerator housing manifold and the bottom row of fins on the second-row tubes. This gap was also plugged with shim stock. Future heater heads will be manufactured with modifications to eliminate such gaps. When these modifications were made, the heat absorbed by the HES immediately increased to levels experienced in Mod I engines.

The thermal distortion problem experienced in the preheater cover was rectified by the installation of a heat shield between the cover and combustion liner, with a thermal-barrier coating applied to its surface.

The cause of melting backing rings between the cylinder housing and liner was due to a misdirected "jetting" effect of working gas exiting from the manifold. Flow from the manifold did not enter the cylinder at top dead center as in the Mod I, rather it impinged on the side wall of the cylinder, allowing hot gases to penetrate down the gap between the piston and cylinder into the area of the o-ring seal between the cylinder and cylinder liner. Temperatures were measured as high as 650°C in this area, which was 150°C above Mod I experience. The profile of the entrance from the manifold to cylinder was modified (Figure 1-6) to redirect flow to the middle of the cylinders. Subsequent tests late in the report period showed that temperature levels and profiles returned to normal. Figure 1-7 shows the current power level and measured efficiency with the above modifications incorporated.

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*complete engine less auxiliaries and control system
**complete engine with auxiliaries and control system
***The fuel nozzle atomizing air was supplied from the test stand.

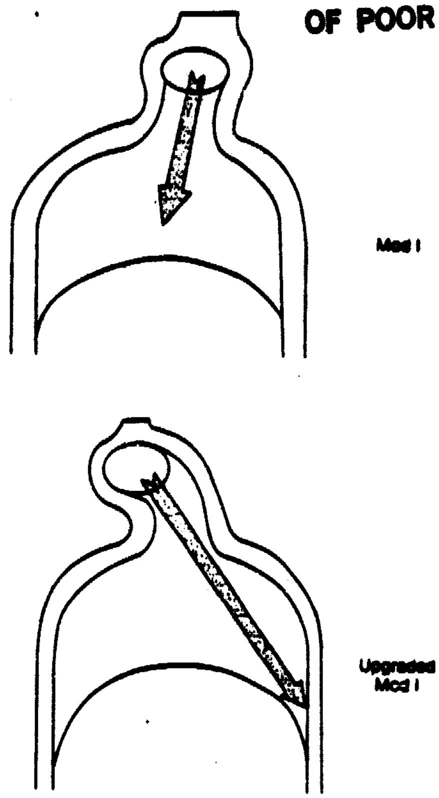

Fig. 1-6 Cylinder Housing with Piston at BDC - Arrows Depict Gas Flow from Manifold

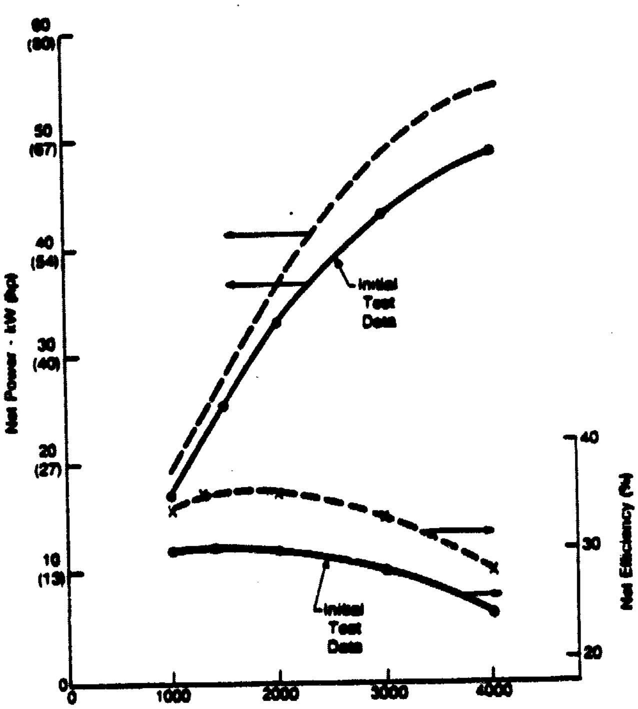

Fig. 1-7 Upgraded Mod I SES* Test Data - ASE 4-123A-10 (15 MPa/720°C Set Temperature)

The data are for Upgraded Mod I engine No. 10 testing as an SES* at 720°C. Note that maximum power has increased 6.1 kW (8.2 hp) to 54.5 kW (73.3 hp), and peak efficiency has increased 5 percentage points to a 35.5% level; however, these data fall below predictions (see Table 5-1) because of poor regenerator performance due to a major flaw that occurred in the fabrication of the regenerator. The flaw was committed during the manufacture of the mesh, resulting in less material mass than was designed for the engine. This allowed heat to "jet" through the regenerator heat exchanger and escape through to the cooling water. This mechanism has been verified through measurement. As a result, new regenerator matrices are being manufactured by a vendor to the correct porosity. Tests scheduled to be conducted in September are expected to bring the measured data into agreement with predictions.

Industry Test and Evaluation Program

The basic objective of the Industry Test and Evaluation Program (ITEP) is to provide an independent evaluation of the technology level of the ASE program, and an opportunity for U. S. automotive/engine manufacturing inputs for improving the design and manufacturability of Stirling engines.

The NASA/Lewis Research Center has negotiated directly with automotive and engine-manufacturing companies to determine their interest in conducting a test program with a Mod I Stirling engine. Public notice was given of the intent to conduct such a program. Several companies responded and, to date, official agreements have been reached between NASA and three interested companies: General Motors, John Deere, and the Cummins Engine Company.

In light of these positive commitments, the ITEP was officially started midway through this report period. Under the program, MTI will procure, assemble, and test two Upgraded Mod I engines with appropriate spares for industry test and evaluation. Certain "hands-on" training will be provided, as well as required drawings, schematics, system descriptions, and parts' lists. MTI will also provide test site support at the

*without air atomizing compressor as an auxiliary individual company locations while they test and evaluate the Mod I Stirling engines. Engine deliveries are scheduled for early 1984, and tests will be conducted at company sites during mid 1984. Currently, all engine hardware is on schedule to initiate assembly of the first engine in September, 1983.

Work Planned for Next Report Period

The following major activities are scheduled for the ASE Program during the latter half of 1983:

- the Upgraded Mod I engine will be fully characterized at steady-state conditions;
- a manufacturing cost study of the V-4 annular RESD will be completed by an independant, outside firm;
- an air-atomized fuel nozzle will be developed for use in the Mod I engine;
- a digital Air/Fuel Control System will complete development for use in the Mod I engine;
- the two ITEP engines will be fully assembled;
- Mod I engine No. 3 will complete 1000 hours of endurance testing; and,
- Upgraded Mod I engine No. 2 will be tested on endurance.

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Design and Performance Summary

The RESD was updated during this report period. Several engine concepts were considered*, and a single design selected as the preferred concept. Emphasis on this update was to reduce the engine's manufacturing cost while maintaining performance that would meet the program goals. Careful value engineering of the RESD has reduced its manufacturing cost from $1651 to $1238 (1983 dollars) - a reduction of $413, or 25%. The selected design (Figure 2-1) contains the following characteristics:

- equal-angle V;
- annular regenerator/cooler;
- rolling-element drive unit;
- ceramic preheater;
- simplified CGR combustor;
- lightweight piston/rod unit; and,
- simplified auxiliaries and control system.

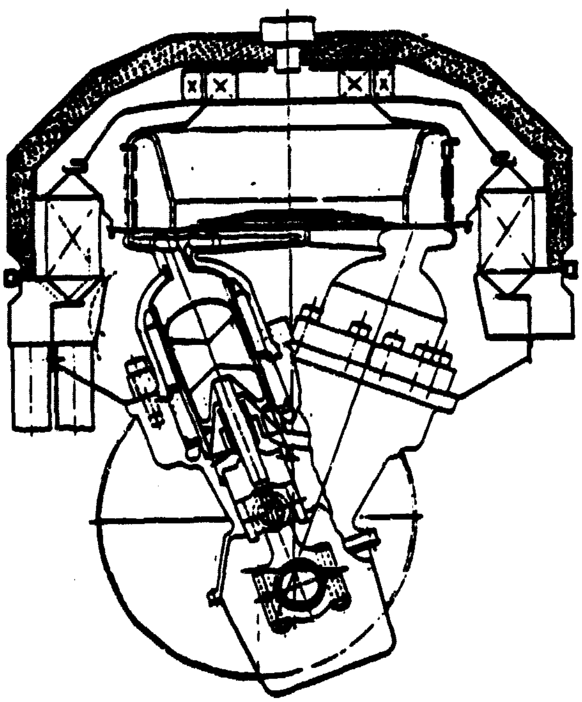

Fig. 2-1 V-4 RESD (Front View)

The engine and vehicle performance summary is shown in Table 2-1. The program goal, in terms of vehicle fuel economy, is to provide a 30% improvement over a comparable IC-engine-powered vehicle with the same acceleration characteristics. The RESD, sized to meet the acceleration criteria of 0-60 mph in 14.7 seconds, 50-70 mph in 9.9 seconds, and 0-100 ft in 4.2 seconds, will provide a combined fuel economy of 40.1 mpg on unleaded fuel and 46.1 mpg on diesel fuel.

Table 2-1

RESD Performance Change Summary

	1983 RESD V-4 Annular
Max Power Point	
Power	60 kW
	80 hp
Speed	4000 rpm
Working Gas Pressure	15 MPa
Efficiency	34.6 %
Max Efficiency Point	
Power	25.5 kW
	34.2 hp
Speed	1350 rpm
Working Gas Pressure	15 MPa
Efficiency	42.2 %
Part-Load Point	
Power	12 kW
	16.1 hp
Speed	2000 rpm
Working Gas Pressure	5 MPa
Efficiency	37.3 %
Weight	173 kg
	381 lbs
Power Density	4.7 lb/hp

RESD Vehicle Fuel Economy Using Unleaded Gasoline

Urban Without Cold-Start Penalty	41.3 mpg
Urban With Cold-Start Penalty	32.3 mpg
Highway	56.9 mpg
Combined	40.1 mpg
Comparable IC	27.0 mpg
% Improvement Relative to IC	48.5%

*see MTI Report No. 82ASE278SA2

Design Description

The specific design technologies embodied in the RESD are presented herein, grouped by major engine systems.

EXTERNAL HEAT SYSTEM

Preheater

The RESD preheater will be a block-type, straight-plate, ceramic recuperative system. The final material, which will be developed during the component technology program, is expected to be similar in nature to cordierite, a material successfully rig-tested. The blocks will be rectangular in shape, and will surround the heater head in an annular arrangement (Figure 2-2). The blocks, designed with angled end surfaces, will be clamped in a spring-loaded metallic fixture that will allow for the difference in thermal growth between the ceramic and the fixture (Figure 2-3). The primary advantage of this design, relative to the current metallic ASE preheater, is a cost reduction on the order of a factor of four.

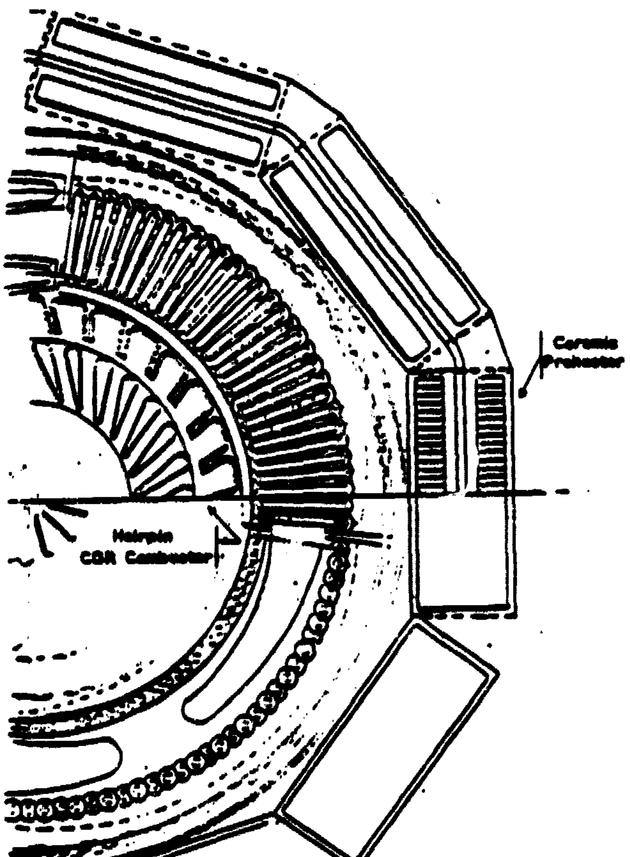

Fig. 2-2 RESD Ceramic Preheater Arrangement (Top View)

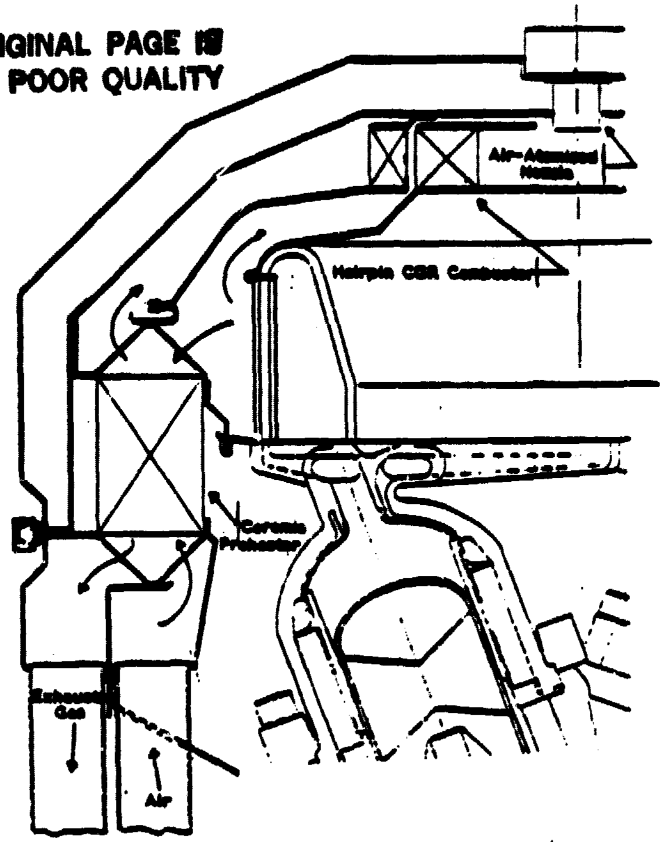

Fig. 2-3 Side View of RESD Showing Preheater Clamping

Combustor/Fuel Nozzle

The combustor is a simplified combustion gas recirculation (CGR) design (see Figures 2-2 and 2-3) that addresses specific areas of concern experienced with previous CGR combustors. Differential thermal growth in Mod I ASE CGR combustors has resulted in durability problems evidenced by cracking and distortion in the swirler and preheater mounting areas. The RESD combustor is flexible in these areas to reduce these problems.

Circumferential combustion-gas temperature nonuniformity with the Mod I system has been a continual problem, most likely caused by uneven ejector performance and inadequate swirl and mixing. The RESD combustor addresses this aspect by utilizing several ejectors in direct combination with the turbulator, which has demonstrated good swirl and mixing characteristics in the P-40 and Mod I engines. This combustor is a very simple, low-manufacturing-cost design. The fuel nozzle, a sonic nozzle driven by a piezoelectric crystal (shown in Figure 2-4), eliminates the need for an atomizing air compressor, thereby decreasing performance penalties. The large flow passages of the nozzle will reduce the plugging problems experienced with the Mod I nozzle.

HOT ENGINE SYSTEM

<u>Heater Head</u>

The heater head, an annular configuration (Figure 2-5), features a ceramic stuffer piece above the regenerator to fill dead volume and minimize stress concentrations due to cross-section changes in the pressure vessel shell (Figure 2-6). The manifold entry, a necked design that provides low stresses at the tube/manifold junction, has proven to be readily castable in the Upgraded Mod I configuration (Figure 2-7). The partition wall, separating the cylinder and regenerator spaces, is a hybrid metal/ceramic configuration utilizing a metal shell into which a ceramic sleeve is compression-fitted. This arrangement combines the benefits of a metal-to-metal fit between the partition wall and heater housing, along with the reduced conduction losses associated with ceramics (Figure 2-8).

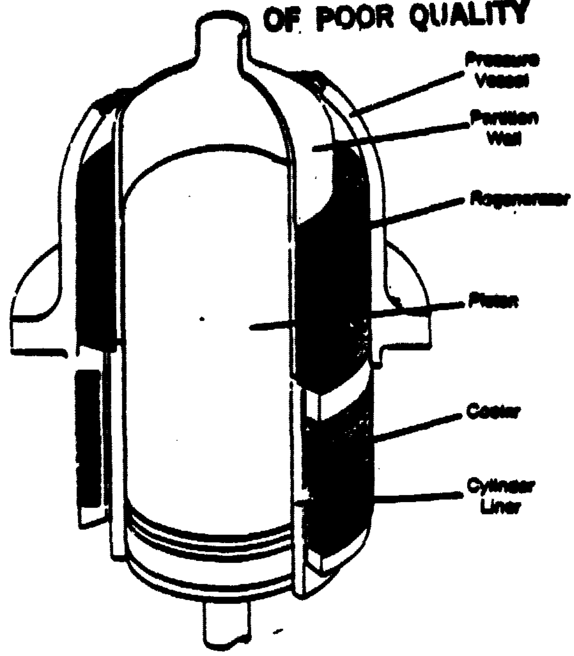

Fig. 2-5 Annular RESD Heater Head Geometry

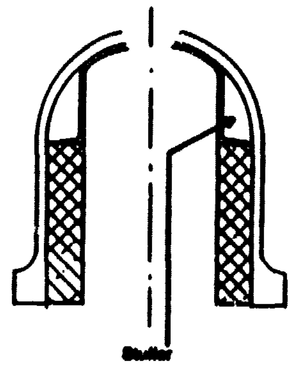

Fig. 2-6 RESD Regenerator Arrangement

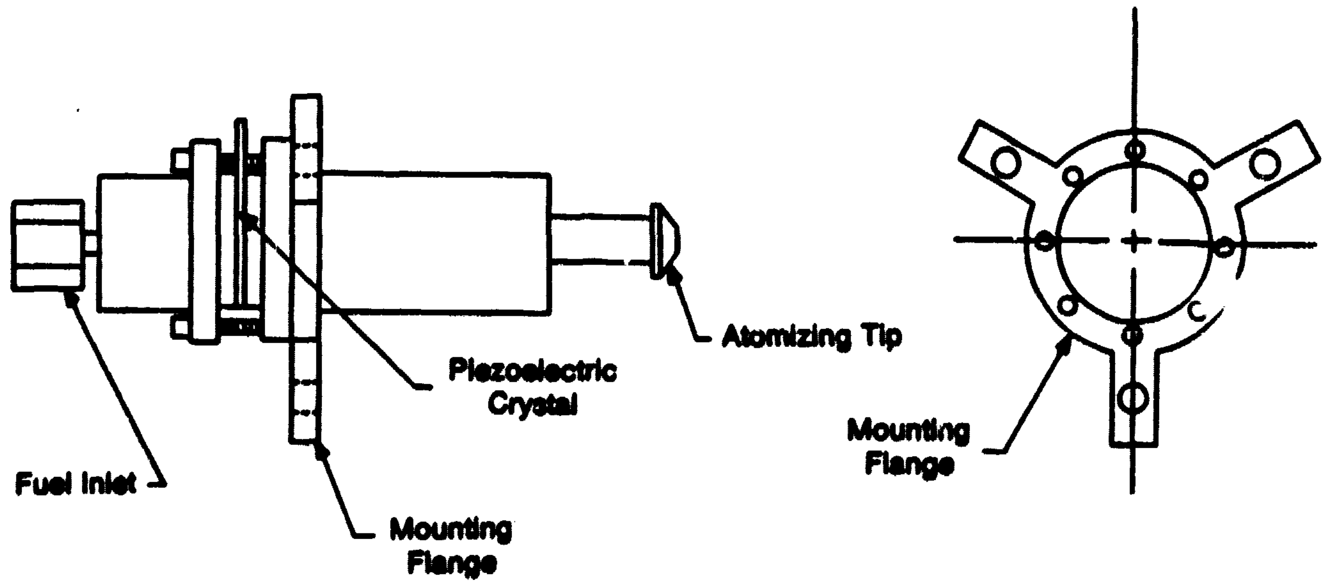

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Fig. 2-4 Outline of Sono-Tek Nozzle and Mounting Flange

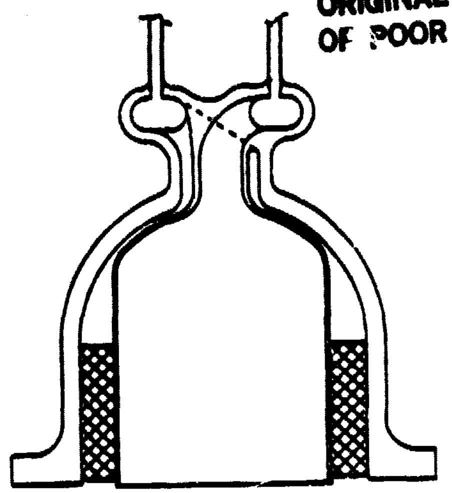

Fig. 2-7 Annular Heater Head Necked Manifolds

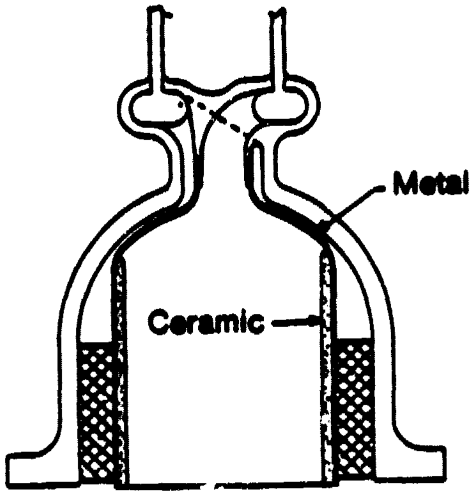

Fig. 2-8 Partition Wall Arrangement

Regenerator

The regenerator is formed of sintered screens similar to the Mod I configuration. A slight increase in wire diameter relative to the Mod I (0.0024" versus 0.0020") has been selected to provide lower manufacturing costs with no perceptible change in engine performance.

Gas Cooler

The gas cooler is a tube- and shell-type similar to the Upgraded Mod I; however, coated carbon-steel tubes instead of the stainless steel tubes used in existing Mod I engines will be used to reduce manufacturing costs.

COLD ENGINE SYSTEM

Block

The RESD integrates the previously separate Cold Engine System parts; namely, water jacket, crosshead guides, cold connecting ducts, and crankcase/bedplate units. The design provides improved tolerance control due to reduction in the number of parts in any stack up, and is much simpler and less costly to manufacture. The block is shown in Figure 2-9 with a piston, rod, and gas cooler in place.

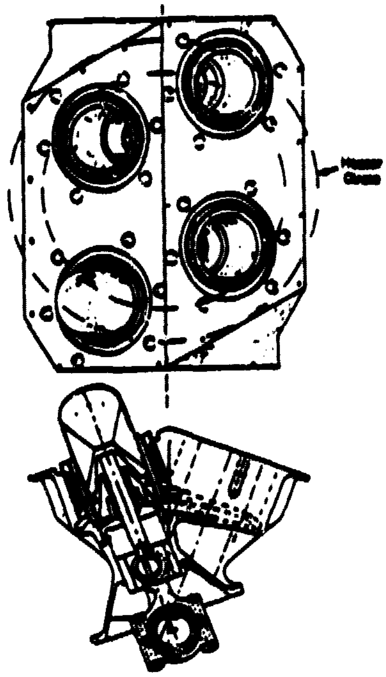

Fig. 2-9 RESD V-Engine Block

Drive Unit and Piston

The RESD drive unit utilizes rolling-element bearings to provide reduced friction losses and utilize the full low-speed torque capability of a Stirling cycle. This technology is based on experience gathered with the Upgraded Mod I engine. The piston/rod combination is a lightweight, unified design in which the piston, dome, and rod are a one-piece welded unit. Attachment at the crosshead is achieved via a threaded joint. This arrangement, in addition to reducing weight, minimizes the overall stack-up tolerances, both axially and radially. Precise control is achieved axially since final machining of a shoulder on the rod at the crosshead joint is measured as a single unit to the dome top. Similarily, concentricity of the dome to the rod is precise. Figure 2-10 shows the piston/rod assembly including the rolling-element bearings.

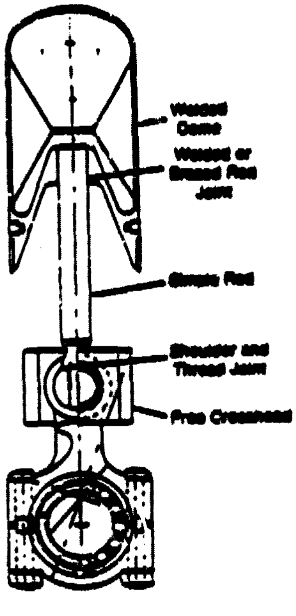

Fig. 2-10 RESD Piston/Rod Assembly

CONTROLS AND AUXILIARIES

Controls

Power Control - A mean-pressure-control system is maintained for the RESD. A rotary hydrogen control valve, which will be electric motor-driven, eliminating the need for servo-oil actuation, has been selected to replace the linearly-actuated spool valve of the Mod I system. This valve, via placement close to the cycles, will also reduce the number of check valves in the system.

Combustion Control - Combustion control of fuel and airflow rates is achieved via a closed-loop system. Airflow is controlled via speed and eccentricity sensing of the positive-displacment air blower, while fuel metering is controlled via a stepper-motor actuated valve. Fine tuning of the system is achieved via an oxygen sensor at the combustion-gas preheater inlet. Heater temperature control will continue to utilize tube-placed thermocouples.

Auxiliaries

Combustion Air Blower - A positive-displacement, variable-stroke blower will be utilized, replacing the variator-driven centrifugal unit of the Mod I engines. This unit (with the rotary hydrogen control valve) eliminates the need for the servo-oil system.

Combined Starter Motor/Upstart Motor - Current Mod I engines use both an upstart motor to drive the combustion air blower, and a starter motor to rotate the engine during the start procedure. Motoring tests of the Mod I engine system indicated that, at low charge pressures, the motoring power required to crank the engine at 400 rpm is quite low (0.65 kW (0.87 hp) at 1 MPa), and the starter motor could be used to crank the engine and drive the combustion air blower. This would eliminate the need for an upstart motor, and also decrease thermal shock during starting, since the working gas would be circulating from the beginning of the start procedure. A revised hydrogen compressor will be used to pump the engine down to 1 MPa on shutdown to permit engine starting with the starter alone, eliminating the need for the upstart motor.

Electronics - The electronics of the control system will continue to be microprocessor-based as in the Mod I.

Manufacturing Cost Analysis

A prime objective of the RESD update was to reduce manufacturing costs to provide an engine competitive with the IC engine. As stated previously, a reduction of $413 was achieved, bringing the manufacturing cost of the RESD down to $1238. A summary of the production costs by major units for the engine and controls is shown in Table 2-2.

During the latter half of 1983, the manufacturing costs will be verified by an independent firm, as was done for the 1981 RESD effort.

Engine Performance

The emphasis on reduced manufacturing costs has been the prime objective of this design update. Incorporation of the reduced cost concepts has, in some cases, resulted in degradation in engine performance.

A comparison of performance at key operating points for the V-4 RESD and the 1981 RESD (shown in Table 2-3) shows that efficiency levels are lower for the V-4. The resultant impact on vehicle mileage, however, is small (~1 mpg).

Table 2-2

RESD V-4 Preliminary Estimated Manufacturing Costs

Title	Cost*	Remarks
Cylinder Housing	$ 56	Material - XF-818 Castings
Heater Tubes	123	Material - CG-27
Regenerator Matrix	125	Stainless-Steel, Square-Mesh Wire Cloth (0.0024" Diameter)
Gas Cooler Assembly	92	Carbon-Steel Tubes - Modified
Preheater Assembly	70	Ceramic Preheater Blocks - Modified
Combustion Chamber Assembly	46	
Piston/Piston Rod Assembly	58	Domes CG-27, Rods 4340 Stainless Steel
Water Jacket, Crosshead Guides, and Crankcase	62	Integrated One-Piece Modular Cast Iron
Oil Sump	3	Low Carbon-Steel Stamping
Main Bearings	14	Three Main Bearings
Crankshaft	30	One Crankshaft
Step-Up Gear Set Assembly	15	Two Helical Gears, Rolled Teeth, Induction-Hardened
Contingencies	175	
TOTAL BASIC ENGINE COST	$ [illegible]9	
Control Block Assembly	62	No Moog Valve or Hydraulic System; Four Control Blocks Reduced to Two Simplified Control Blocks
H2 Storage Vessel	35	Capacity - 4 liters
Electronic Package	62	Simplified Design for Automated Volume Production
Starter/Upstart Motor	38	Starter Motor and Upstart Motor Combined
Alternator	66	Commercial Alternator
Lube Oil Pump	3	Commercially Available Item
Water Pump	17	No Special Fabricated Stainless-Steel Pump; Use Commercial Pump
Combustion Air Blower	54	Modified High-Efficiency Design for Automated Volume Production
Air/Fuel Control	37	Modified Version for Easy Manufacturing
H2 Compressor	35	Commercially Available Item
Transducers	131	Five Pieces Required; Buy Commercially Available Items
Simplified Pressure-Control System	27	Modified Design for Automated Volume Production
Contingencies	20	
TOTAL CONTROLS/AUX. COST	587	
TOTAL SYSTEM COST	1456	
NEW PRODUCT COST REDUCTION (15%)	218	
FINAL MANUFACTURING COST	$1238	

*1983 dollars

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Table 2-3

RESD Performance Comparison

Operating Point		1981 RESD	V-4 1983 RESD	Operating Point		1981 RESD	V-4 1983 RESD
Full-Load	15 MPa			Max. Efficiency	15 MPa		
Engine Speed	rpm	4000	4000	Engine Speed	rpm	1100	1350
Indicated Power	kW	73.3	69.8	Indicated Power	kW	24.8	28.1
	hp	98.3	93.6		hp	33.3	37.7
Friction	kW	9.8	6.3	Friction	kW	2.3	2.0
	hp	13.1	8.4		hp	3.1	2.7
Auxiliaries	kW	3.4	3.4	Auxiliaries	kW	0.4	0.6
	hp	4.6	4.6		hp	0.5	0.8
Net Power	kW	60.1	60.0	Net Power	kW	22.1	25.5
	hp	80.6	80.5		hp	29.6	34.2
EHS Efficiency	%	90.5	90.3	EHS Efficiency	%	92.4	92.8
Net Efficiency	%	34.2	34.6	Net Efficiency	%	43.5	42.2
Part-Load	12 kW 16(hp)			Low-Load	5 MPa		
Engine Speed	rpm	2000	2000	Engine Speed	rpm	1000	1000
Indicated Power	kW	14.9	14.3	Indicated Power	kW	7.9	7.2
	hp	20.0	19.2		hp	10.6	9.7
Friction	kW	2.1	1.4	Friction	kW	0.9	0.6
	hp	2.8	1.9		hp	1.2	0.8
Auxiliaries	kW	0.8	0.9	Auxiliaries	kW	0.4	0.4
	hp	1.1	1.2		hp	0.5	0.5
Net Power	kW	12.0	12.0	Net Power	kW	6.6	6.2
	hp	16.0	16.0		hp	8.9	8.3
EHS Efficiency	%	91.7	92.3	EHS Efficiency	%	89.8	90.7
Net Efficiency	%	37.6	37.3	Net Efficiency	%	36.4	33.9
Heat Conduction Loss	kW	1.6	2.6				

Vehicle Packaging

The program vehicle for the RESD installation is the General Motors X-Body car. All vehicle auxiliaries (i.e., air-conditioning compressor, power-steering pump, power brakes, and cooling fan) are retained. A side view of the installation is shown in Figure 2-11, including the auxiliaries positioning, vehicle hood, frame, fire wall, and radiator mounting. The auxiliaries, as shown, are driven by a single, serpentine-ribbed belt. Top and front views of the installation are shown in Figures 2-12 and 2-13. The engine packages in the X-Body engine compartment with an existing production torque converter/transaxle, with no modifications to the vehicle.

Vehicle Performance

RESD vehicle performance has been estimated using the MTI Vehicle Simulation Computer Code. This code has been verified via comparison to the Mod I Lerma TTB. Table 2-4 compares actual test results to those predicted via the code. Using this verified code, the V-4 RESD engine maps and the RESD vehicle characteristics (shown in Table 2-5) were utilized to develop RESD mileage projections. The resultant fuel economy projections are depicted in Table 2-6. Note that the combined mileage of the RESD-powered vehicle (40.1) compared to the baseline IC mileage (27) represents a mileage improvement of 48.5% relative to the program goal of 30%.

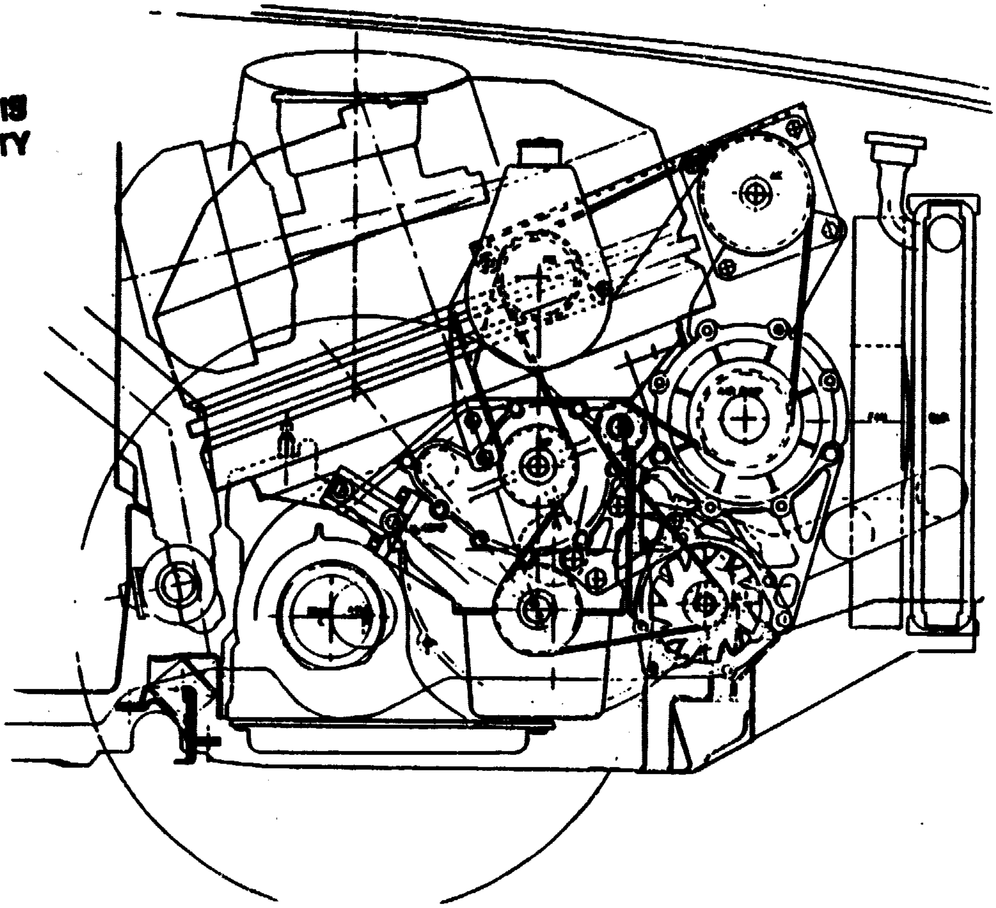

Fig. 2-11 RESD Vehicle Installation (Side View)

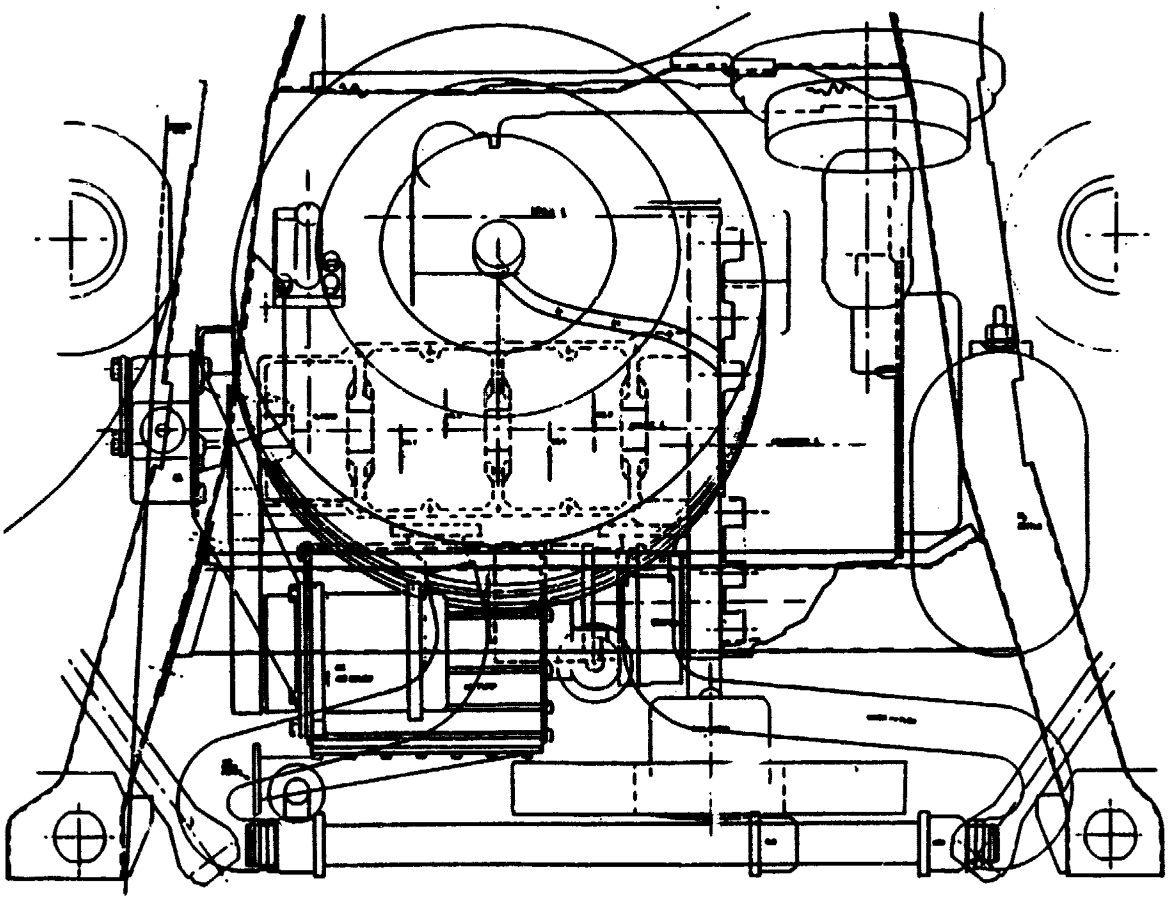

Fig. 2-12 RESD Vehicle Installation (Top View)

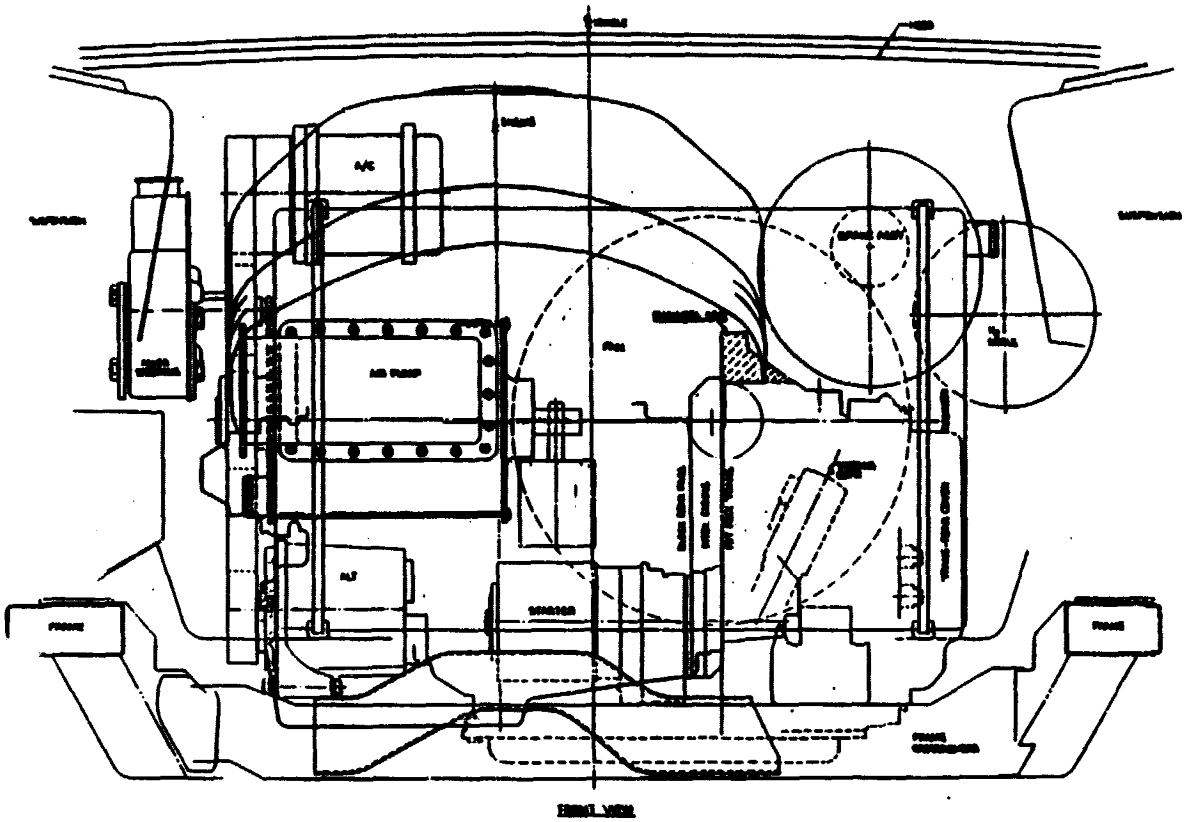

Fig. 2-13 RESD Vehicle Installation (Front View)

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Table 2-4

Mod I Lerma Test Data Compared to Predictions

	Urban				Highway	Combined
Description	HC g/mi	CO g/mi	NOx g/mi	Mileage mpg	mpg	mpg
Test No. 1	.230	3.40	.960	19.90	31.70	23.90
Test No. 2	.290	3.30	.900	18.80	32.10	23.20
Test No. 3	.250	3.20	.840	19.20	32.40	23.50
Standard Deviation	.025	0.082	.049	0.45	0.29	0.29
Mean	.260	3.30	.900	19.30	32.10	23.50
Projections	--	--	.840	19.10	30.00	22.80

11.1 hp at 50 mph Road Load Setting
3750-lbs IW Setting
2.73 Axle Ratio

Table 2-5

RESD Vehicle Specifications

Complete vehicle specification published as MTI Report No. 79ASE34TS1 Rev. 4

GENERAL

Model, Year	1983 Pontiac Phoenix 4-Door Hatchback
EPA Class	Mid-Size
Max. Passengers	2 Front/3 Rear
Vehicle Weights	
Curb Weight	1304.5 kg (2870 lb)
Test Weight	1440.9 kg (3170 lb)
EPA Inertia Weight	1420.4 kg (3125 lb)

ROAD LOAD

Rolling Resistance Coefficient	0.0095 (road) 0.0110 (dyno)
Rolling Radius @ 72.4 km/h(45 mph)	302.7 mm (0.993 ft) (846 rev/mile)
Air Drag	5.37 kW @ 80.4 km/h) (7.20 hp @ 50 mph)
Drag Coefficient	0.417
Frontal Area	21.34 ft2
Total Road Load	8.37 kW @ 80.4 km/h (11.22 hp @ 50 mph)
EPA DPA Setting (includes 10% A/C penalty)	5.414 kW @ 80.4 km/h (7.26 hp @ 50 mph)

TRANSAXLE TYPE	Modified GM THM 440 (4-speed automatic w/ lockup in 3rd and 4th)

GEAR RATIOS

Engine/Torque Converter	1.40
First	2.92
Second	1.55
Third	1.00
Fourth	0.704
Final Drive	3.64

SHIFT SCHEDULE	Optimized for Stirling Engine
TORQUE CONVERTER	Chrysler 241 mm (9-1/2 in.)
Stall	k = 210
Lockup Speed	30 mph
Unlock Speed	25 mph
ACCESSORIES	Power Steering/ Brakes, A/C
Accessory Loads	P/S, B, Electric Cooling Fan, Transaxle Front Pump

ACCELERATION PERFORMANCE

0-60 mph Time	14.7 sec
50-70 mph Time	9.9 sec
0-100 ft Time	4.2 sec
Top Speed	>100 mph

Table 2-6

RESD Vehicle Fuel Economy Performance

		IC	V-4 1983 RESD
Urban w/o Cold-Start Penalty	(mpg)	--	41.3
Cold-Start Penalty	(g)	--	141
Urban	(mpg)	--	32.3
Highway	(mpg)	--	56.9
Combined Mileage	(mpg)	27	40.1
Combined Mileage with Diesel Fuel	(mpg)	--	46.1

Component development activity is organized on an engine subsystem basis with developmental emphasis on: 1) External Heat System (combustor, fuel nozzle, igniter, preheater); 2) Hot Engine System (heater head, regenerators, coolers); 3) Materials (heater head casting/ tube materials); 4) Cold Engine System (piston ring, main seal/capseal systems, piston domes, cylinder liner); 5) Engine Drive System (crankcase, crankshaft, bearings, connecting rods); and, 6) Control System (mean pressure, combustion, temperature, and microprocessor-based controls).

During this semiannual report period, development activity focused on design substantiation through Upgraded Mod I engine testing, initiation of further Upgraded Mod I component development consistent with the proof-of-concept program, and preparation for the RESD update. With the fabrication complete, testing is underway on the Upgraded Mod I, the technology development cycle has now been completed for an engine design, and technology transfer is complete.

During the latter half of 1983, primary emphasis will be placed on the development of technology justified in the updated RESD, and further design substantiation based on Upgraded Mod I testing.

External Heat System (EHS)

The primary goal of the EHS is low emissions while maintaining high efficiency for an 18:1 fuel turndown ratio in a minimum volume. The design must consider durability, heater head temperature profile, and expected use of alternate fuels, while recognizing the significant cost impact of system size and design.

Development activity during the first half of 1983 concentrated on the design and selection of an improved fuel nozzle for the Upgraded Mod I, CGR combustor development, analysis of Upgraded Mod I emissions, development of a technique to predict transient vehicle emissions from steady-state engine data, and ceramic preheater performance and cost studies.

Primary objectives during the latter half of 1983 will be the development and selection of a reduced-flow, air-atomized nozzle and CGR combustor for Upgraded Mod I engine evaluation, improving combustor durability through the use of thermal-barrier coatings, and the initiation of fabrication and rig testing of RESD ceramic preheater matrices.

UPGRADED MOD I FUEL NOZZLE DEVELOPMENT

The Upgraded Mod I fuel nozzle must be capable of operation over a turndown ratio of 18:1, allow the engine to idle at 0.25 g/s fuel flow, demonstrate an improvement in cycle efficiency through reducing or eliminating atomizing air, prevent the formation of carbon (adversely affects engine maintenance intervals), and have a neutral or beneficial influence on emissions and heater head temperature profile.

Initially, a Delavan nonair-atomized nozzle was selected for the Upgraded Mod I engine based on spray pattern, ignition performance, and steady-state Combustor Rig emissions/temperature profile. This piloted air-blast nozzle (Figure 3-1), originally developed for gas-turbine use, has two flow passages. The secondary passage uses combustor pressure drop to induce airflow through the nozzle for atomization. Since the turndown ratio of the secondary passage is ~6:1, a pressure-atomized pilot (primary) is utilized for the low fuel flows characteristic of ignition and idle conditions.

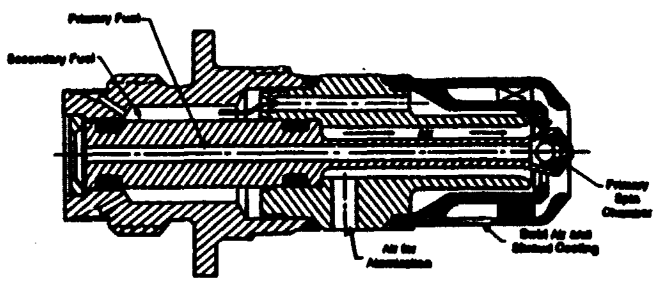

Fig. 3-1 Delavan Air-Blast Nozzle

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Experience in the rig indicated a tendancy to form carbon and plug the pilot zone orifice at idle where fuel cooling is at a minimum and heat load is high due to the 750°C combustor inlet air temperature. The nozzle also adds complexity to the Air/Fuel Control*(Figures 3-2 and 3-3), and makes the purging of the nozzle upon shutdown more difficult since blower pressure is inadequate and there is no atomizing air compressor. Subsequent testing of the nozzle in the TTB verified the adverse effects of carbon buildup and a complex fuel control on emissions and engine efficiency. Although a water-cooled nozzle housing was designed to eliminate plugging, the purge and control complexity imposed by the air-blast nozzle were deemed too detrimental to engine cost and reliability to be feasible for the Upgraded Mod I engine. As a result, emphasis was placed on the development of an "improved" air-atomized nozzle for the engine.

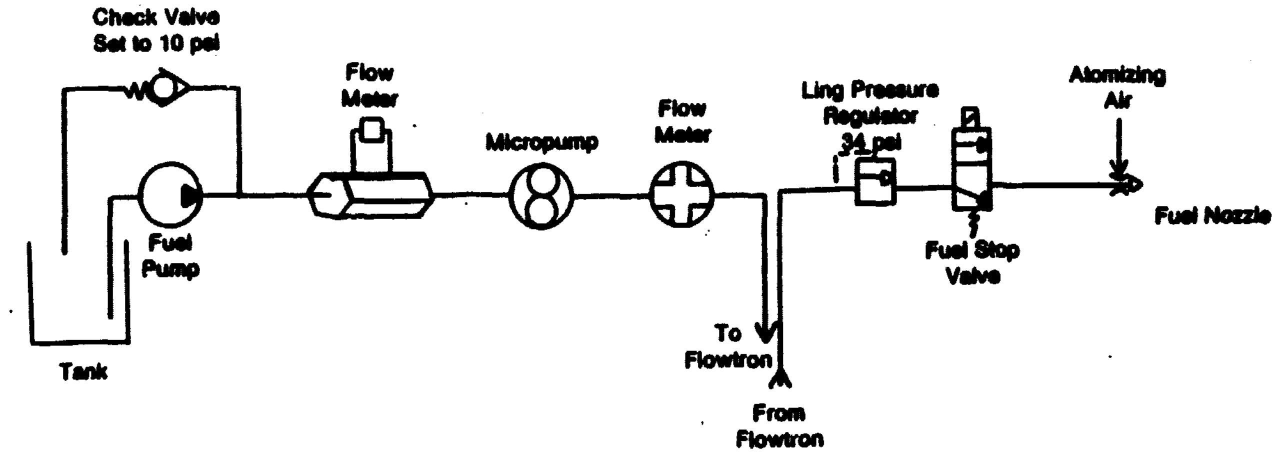

Fig. 3-2 Fuel System for an Air-Atomized Nozzle

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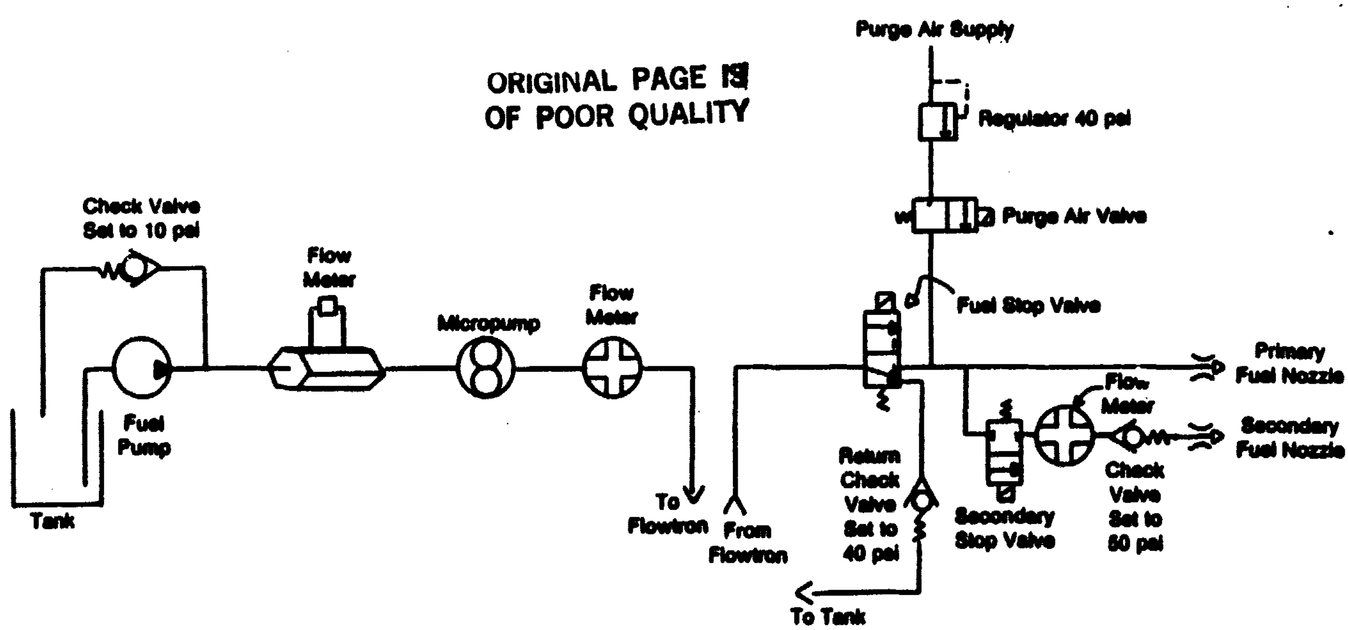

Fig. 3-3 Fuel System for Piloted Air-Blast Nozzle

*two as opposed to one fuel flow

An air-atomized nozzle has a number of advantages such as wide turndown, multi-fuel capability, and simple fuel control. Disadvantages include the need for an atomizing air compressor, the negative impact of cold atomizing air on EHS efficiency (η_B), and the potential to plug if small-diameter fuel orifices are used. The "improved" nozzle refers to reducing the atomizing airflow and pressure requirements relative to the current BOM Mod I nozzle, as well as eliminating carbon buildup, e.g., plugging. By reducing the required atomizing air, a smaller, less expensive compressor can be used, and the impact on η_B at low power can be reduced from two percentage points to negligible. Use of a single large orifice*, either alone or in combination with a water-cooling jacket, will eliminate the detrimental (to heater head temperature distribution) internal carbon buildup characteristics of the Mod I design.

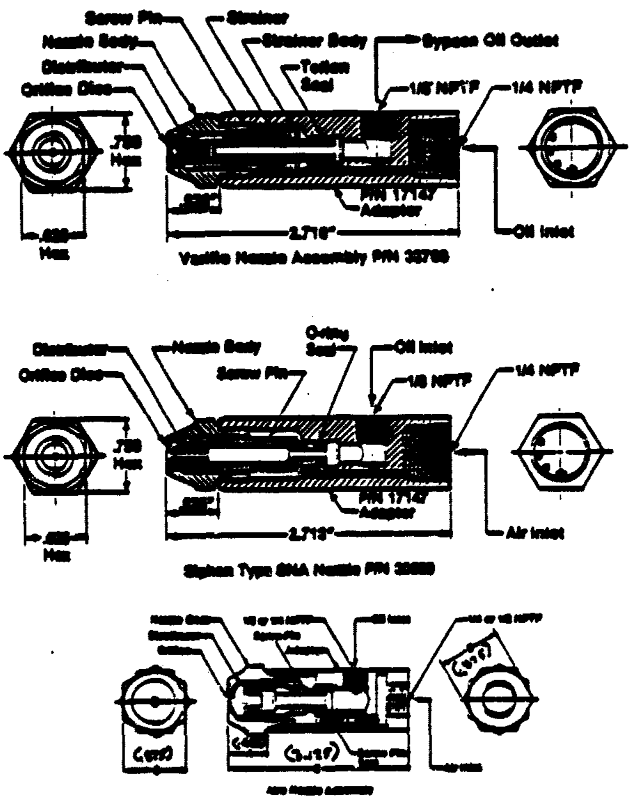

Fig. 3-4 Delavan Industrial Fuel Nozzles

Both existing and new design air-atomized nozzles were considered. The former category includes three nozzles manufactured by Delavan (see Figure 3-4 and Table 3-1). The Variflo nozzle, although not air-atomized, was included because of its uniform spray pattern. The latter category consists of two nozzles that were designed and manufactured to incorporate radial and axial air swirl (Figure 3-5). A housing was also made for the radial nozzle with a water-cooling passage (Figure 3-6).

Table 3-1

Candidate Upgraded Mod I Nozzles

Delavan Nozzle	Type
Variflo	Spill-Flow Pressure-Atomizing
Siphon	External Air-Atomizing
Airo	Internal Air-Atomizing

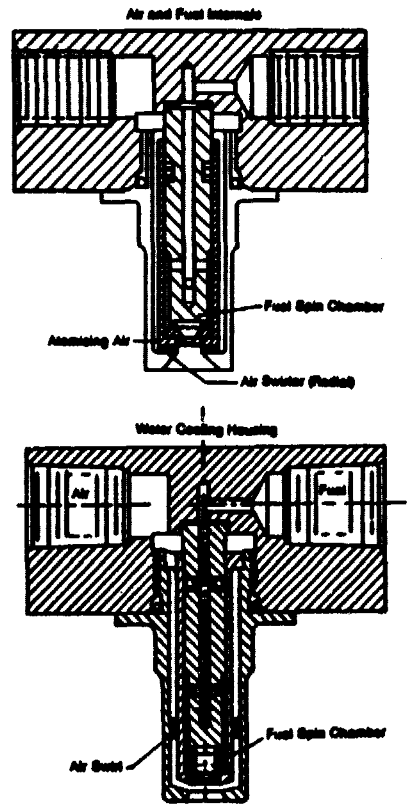

Fig. 3-5 MTI Air-Atomized Fuel Nozzles

*The Mod I nozzle has twelve small-diameter orifices.

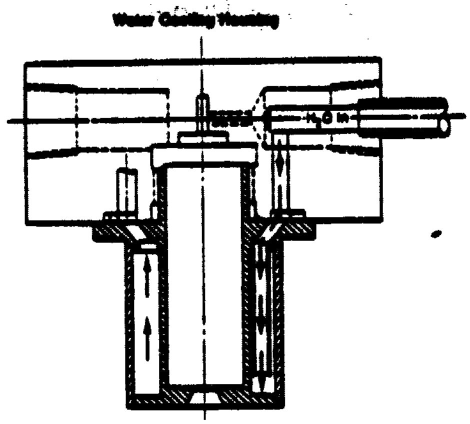

Fig. 3-6 MTI Fuel Nozzle Water-Cooled Housing

Table 3-2

Nozzle Flow Rates

	Mod I	Upgraded Mod I Goal	Radial
Atomizing Airflow (g/s)	1.5	<0.75	0.35
Atomizing Air Pressure (psi)	10.0	5.00	3.00
Maximum Fuel Pressure (psi)	30.0	30.00	15.00

The first stage of the evaluation involved spray tests conducted in the Fuel Flow Rig with diesel fuel. By observing the spray pattern, an estimate of atomizing air and fuel pressure requirements was obtained (Figures 3-7 and 3-8). One version each of the Variflow and Siphon nozzles, three different size Airo nozzles, and four each of the radial and axial nozzles* were sprayed. The radial nozzle demonstrated excellent atomization from idle to maximum power with substantially reduced atomizing air (see Table 3-2).

Based on the spray tests, the Variflow, one version of the Airo, and several radial nozzles were selected for further evaluation in the Combustor Performance Rig. In order to provide a direct comparison of nozzle performance, all tests were conducted with an increased ΔP Mod I EGR combustor previously developed for the air-blast nozzle** without EGR. Both the Variflo and large Airo exhibited poorer performance than the Mod I nozzle.

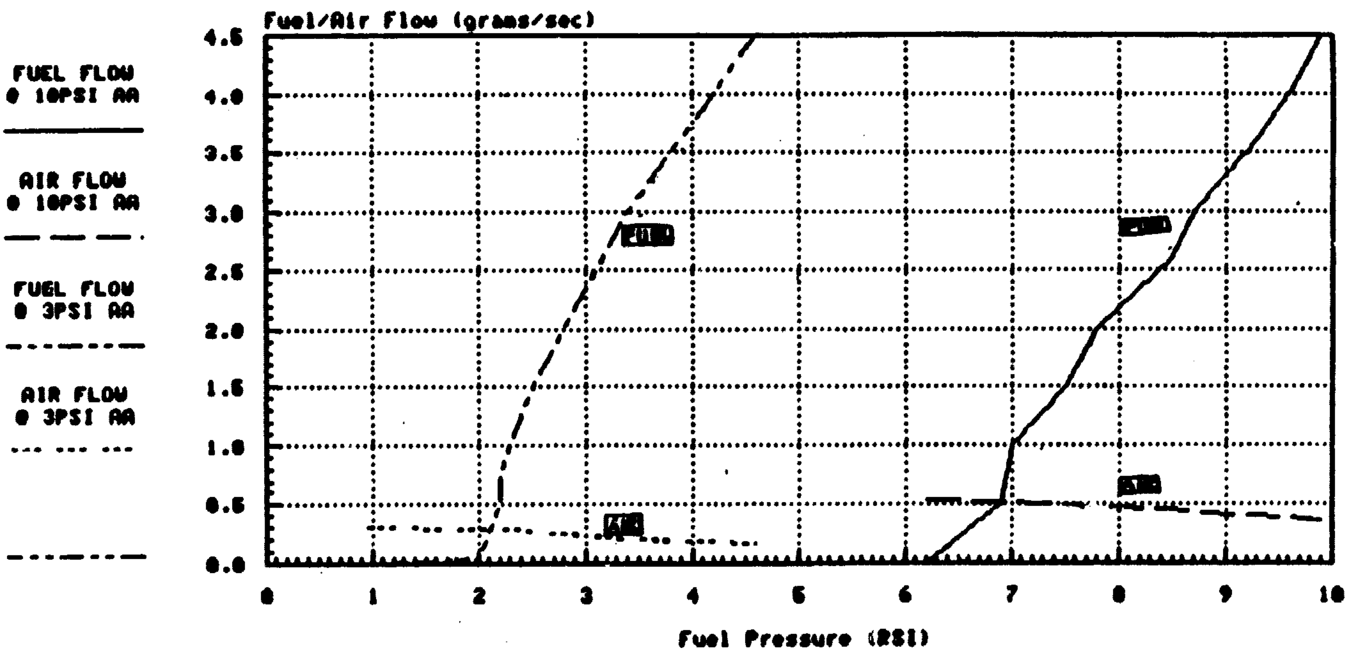

Fig. 3-7 Delavan Airo Nozzle - Fuel and Airflow Versus Fuel Pressure

*radial and axial versions involved variations in sleeves and spin chambers, e.g., atomizing air/fuel passage geometry.
**see MTI Report No. 83ASE308SA3

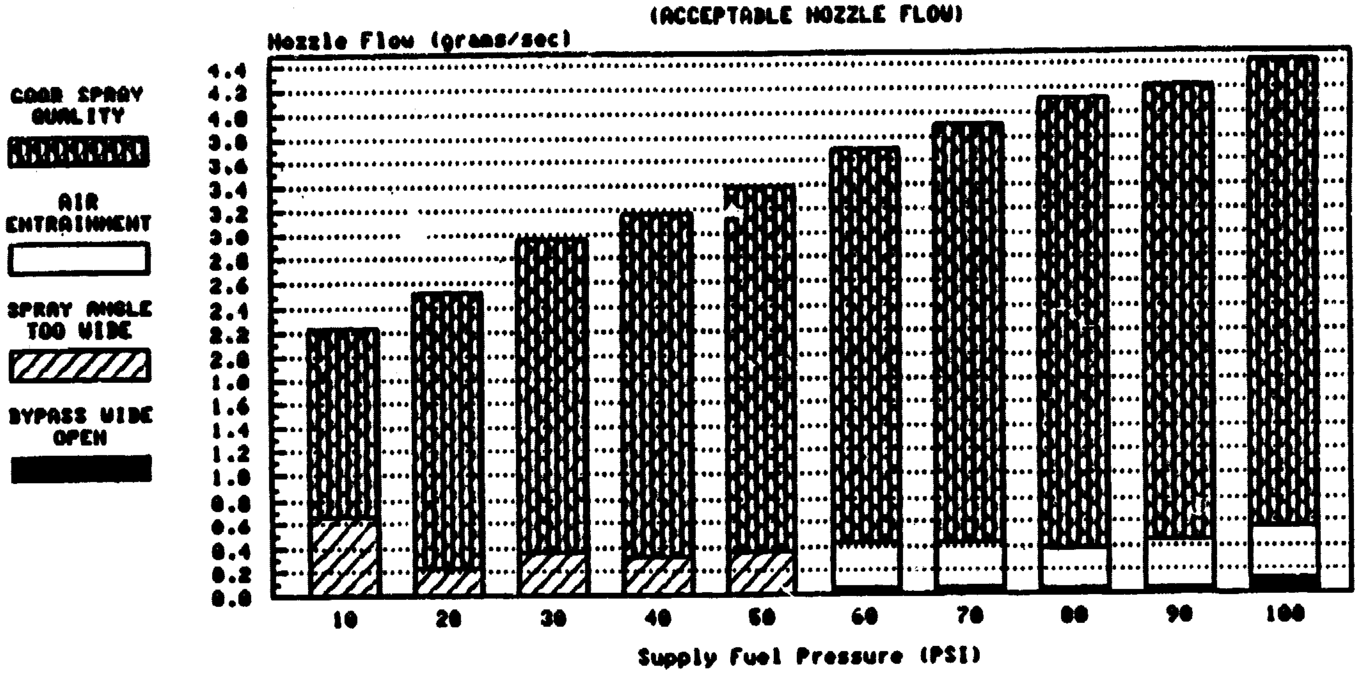

Fig. 3-8 Delavan Varifio Nozzle - Nozzle Flow Versus Supply Fuel Pressure

In the former case, high heater tube temperature spreads were obtained, as well as high CO and HC emissions. In the latter case, temperature spread (Figure 3-9), CO (Figure 3-10), and HC were considerably higher, and NOx (Figure 3-11) was lower than the Mod I rig baseline. The two versions of the radial air-atomized nozzle tested in the rig demonstrated encouraging results (Figures 3-12 to 3-15). Heater tube temperature variations are comparable to the Mod I nozzle at fuel flows less than 2 g/s and λ greater than 1.2. CO and HC emissions are comparable if λ is greater than 1.2, and NOx emissions are lower than the Mod I baseline. These data were obtained at atomizing airflows equal to 25% of the Mod I.

The high temperature spreads obtained at fuel flows above 2 g/s are believed to be due to a narrowing of the nozzle spray angle (60°). The radial nozzle has also demonstrated carbon-free operation during the eleven hours of rig testing without the use of water-cooling.

During the latter half of 1983, a reduced-size Airo nozzle and redesigned version of the radial nozzle having a wider spray angle (80°) water cooling and a reduced face (to eliminate external carbon buildup) will be fabricated and tested in the Performance Rig. These tests will be conducted with EGR upon completion of rig modifications to provide this capability.

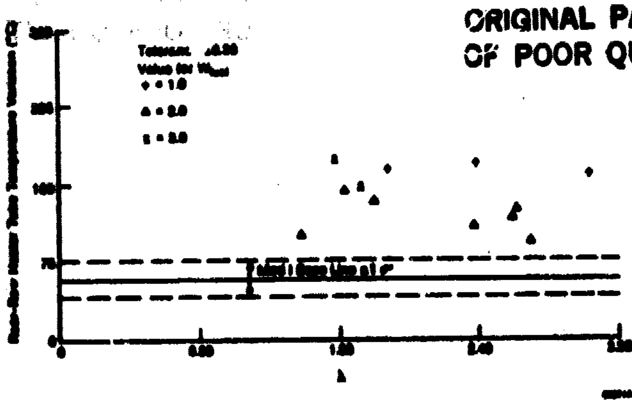

Fig. 3-9 Comparison of Airo and Mod I Nozzle Heater Head Temperature Variation

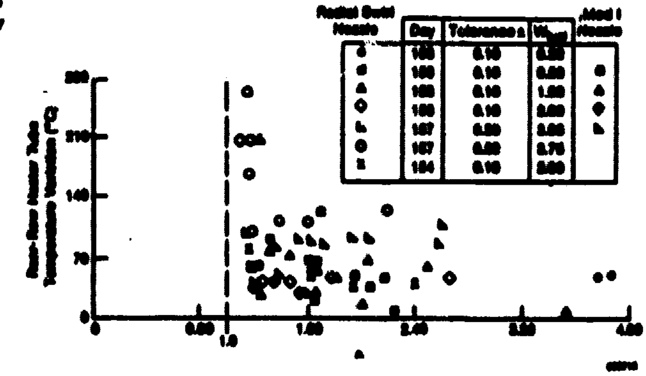

Fig. 3-12 Comparison of Radial and Mod I Nozzle Tube Temperature Variation

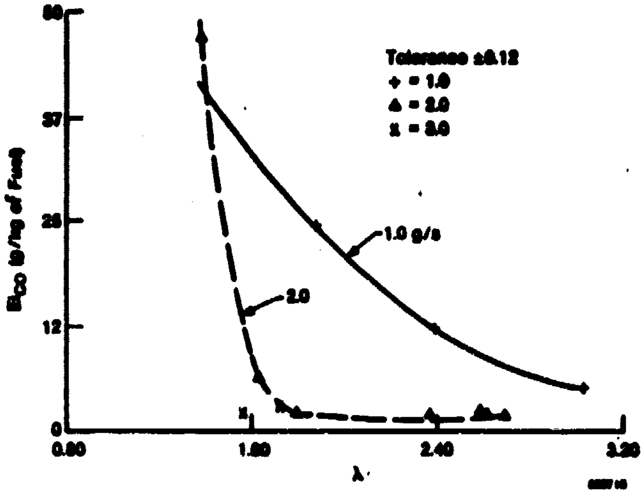

Fig. 3-10 Airo Nozzle CO Emissions

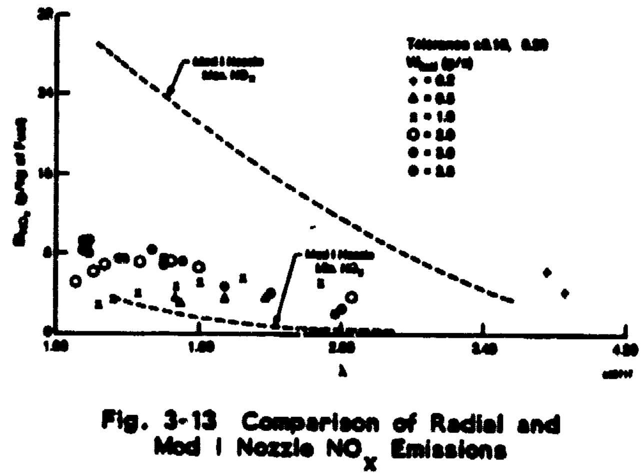

Fig. 3-13 Comparison of Radial and Mod I Nozzle NO_x Emissions

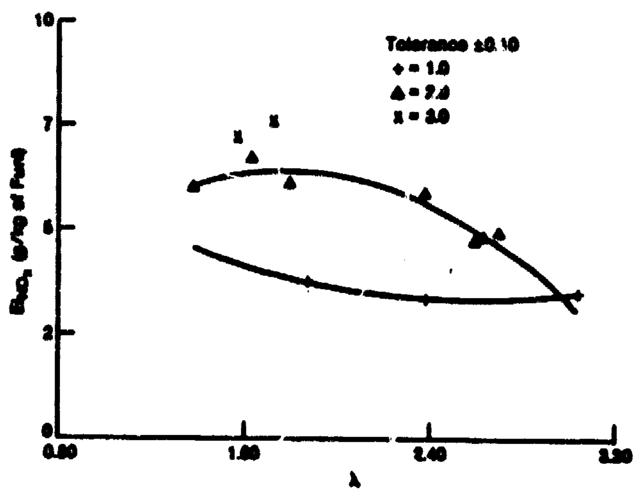

Fig. 3-11 Airo Nozzle NO_x Emissions

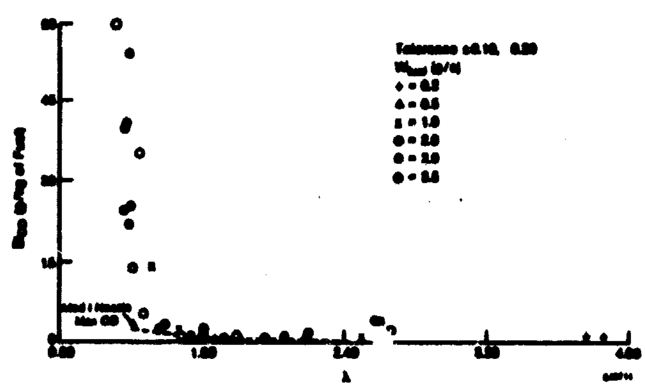

Fig. 3-14 Comparison of Radial and Mod I Nozzle CO Emissions

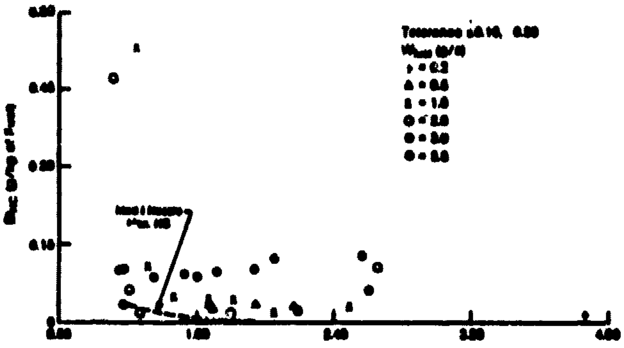

Fig. 3-15 Comparison of Radial and Mod I Nozzle HC Emissions

UPGRADED MOD I CGR COMBUSTOR DEVELOPMENT

The initial Upgraded Mod I combustor is a turbulator design that utilizes EGR for emissions control. This will be replaced with a CGR combustor that simplifies engine packageability and preheater maintenance requirements. The CGR combustor must fulfill the ASE Program emissions requirements of NOx = 0.4 g/mi, CO = 3.4 g/mi, HC = 0.41 g/mi, and particulates = 0.2 g/mi, minimize heater tube temperature variation for high efficiency and long life, operate with low excess air and pressure drop to limit parasitic blower power consumption, be stable at off-design conditions, and be durable.

During this report period, developmental testing of the radial CGR concept was conducted in the Performance Rig, a tubular CGR combustor was designed for Upgraded Mod I engine evaluation, and a new hairpin CGR combustor was conceptualized

The radial CGR combustor (Figure 3-16) utilizes a continuous annulus, as opposed to descrete ejectors, to induce recirculating flow. Eleven radial CGR combustor configurations (Figure 3-17) were tested in the rig. None of these geometries proved capable of generating sufficient CGR for NOx control, contradicting earlier cold-flow test results (10% hot versus 40-45% cold). The discrepancy in test results is more than one would expect due to temperature difficiencies, and may be due to fabrication difficulties encountered with the Performance Rig combustors. Due to the unfavorable test results and the complexity of the design, radial CGR was eliminated as a candidate for the Upgraded Mod I engine.

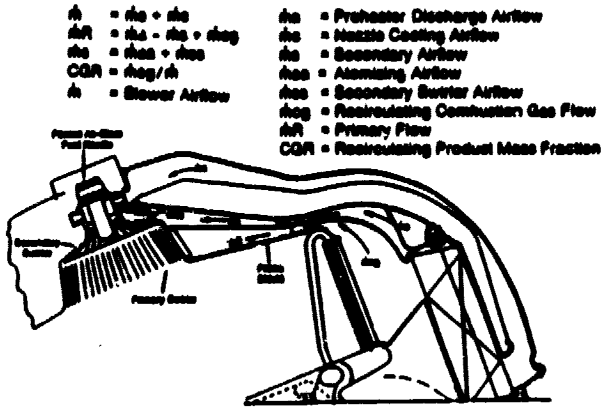

Fig. 3-16 Radial CGR Combustor

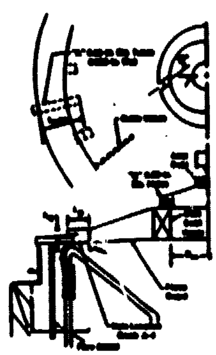

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Fig. 3-17 Radial CGR Configurations Tested in the Combustor Performance Rig

The tubular CGR combustor (Figure 3-18) designed for the Upgraded Mod I has a different geometry and number of ejector/CGR tubes compared to earlier Mod I designs (see Table 3-3). Previous Mod I testing of tubular CGR combustors demonstrated the superiority of the smaller ejector diameter and circular versus square tubes from a temperature profile and emissions standpoint. The smaller ejector, however, has a slightly negative effect on engine efficiency (more blower power), while the all-circular tube necessitates an additional support ring in the combustor assembly. The Upgraded Mod I combustor uses an increased number of tubes and length/diameter for improved combustor mixing and recirculation, respectively. The new tubular combustor will be tested in an Upgraded Mod I engine during the latter half of 1983.

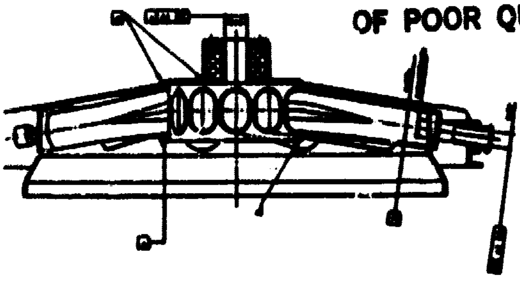

Fig. 3-18 Tubular CGR Combustor

The hairpin CGR combustor (Figures 3-19 and 3-20) incorporates the features of the EGR turbulator combustor with the ejector-flow-inducing technique necessary for CGR. The compact, inexpensive design combines the swirler (turbulator) and ejector into one piece. Air flowing into the combustor through the minimum hairpin ejector area induces combustion products to flow through the recirculation channel connected to the base of the hairpin. A hairpin combustor will be designed, fabricated, and tested in an upgraded Mod I engine during the latter half of 1983.

A program was initiated to improve combustor durability through the use of oxidation-resistance and/or thermal-barrier coatings. These ceramic coatings were considered for application to an EGR combustor flame shield. A three-layer graded thermal-barrier coating by Jetcom Inc. was selected (see Table 3-4).

Table 3-3

Tubular CGR Combustor Comparison

		Ejector		CGR Tube Geometry		
Design	Engine	Diameter	No.	L/D	Inlet	Exit
1-S-2074	Mod I	12 mm	8	4.33	Circular	Circular
1-S-2074	Mod I	13 mm	8	4.33	Circular	Circular
1-S-2134	Mod I	13 mm	8	3.25	Circular	Circular
1-18199	Upgraded Mod I		12	4.60	Circular	Circular

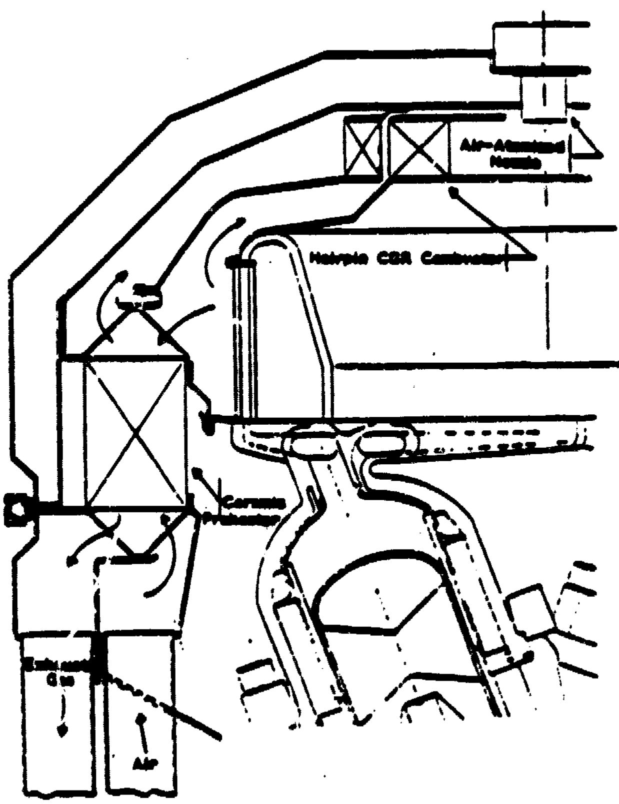

Fig. 3-19 RESD External Heat System

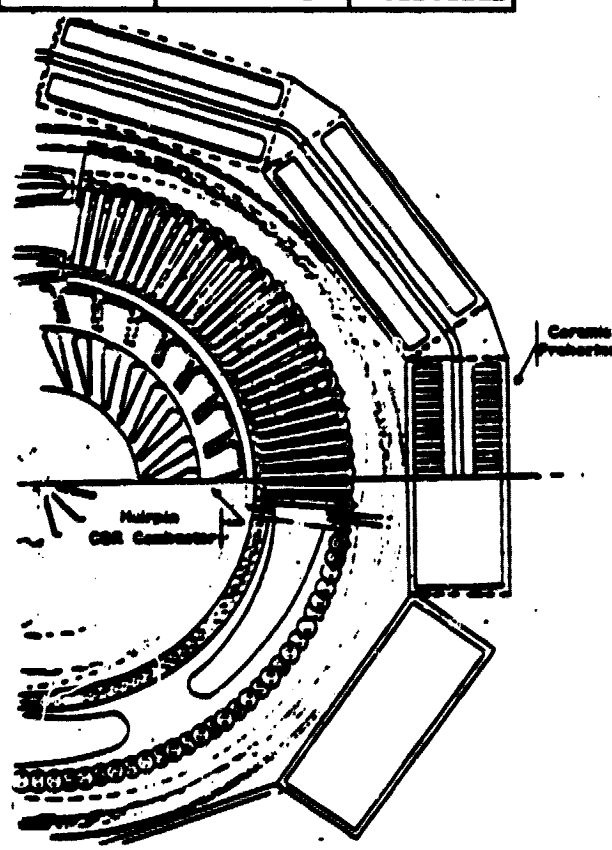

Fig. 3-20 RESD External Heat System (Top View)

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Table 3-4

Jetcom Coatings

Layer	Thickness	Constituent
Bond Coat	4 mils	Ni-22Cr-10Al-1Y
Mited Coat	8 mils	NiCrAlY/ZrO_2
Oxide Coat	8 mils	ZiO_2-20Y/Z_2O_3

Durability will be evaluated in the Combustor Performance Rig during the latter half of 1983.

ENGINE/VEHICLE EMISSIONS ANALYSIS

Since low emissions is one of the prime goals of the ASE Program, the development of analytical techniques to predict transient vehicle CVS cycle emissions from steady-state engine data is necessary so that refinements to emissions control technology can be made early in the development stage. Activity during this semiannual report period centered on improving CO and HC predictions using Mod I engine and vehicle data, and comparing Mod I/Upgraded Mod I emissions with EGR. A technique had been previously developed that divided the CVS cycle into discrete time-weighted engine fuel flow ranges (Figure 3-21). This was applied to steady-state engine emissions data (Figure 3-22) to predict vehicle emissions. During this time, additional vehicle CVS cycle measurements obtained with the Mod I TTB allowed further refinements to that technique.

Comparing predicted versus measured vehicle results for two different EGR levels (Table 3-5), a conclusion was reached that NOx predictions were accurate, while CO and HC were not. By including the effect of combustor ignition delay (~0.2 seconds), the discrepancy in HC emissions was drastically reduced. Current efforts are being made to quantify the effects of transient λ variation (Figure 3-23) and cold heater tube temperature (Figure 3-24) following start-up. Qualitatively, the influence of starts and transients is illustrated by comparing Urban and Highway emissions (Table 3-6). In the latter case, there are no starts and fewer transients, CO decreases an order of magnitude, and HC decreases by a factor of 50 to 100.

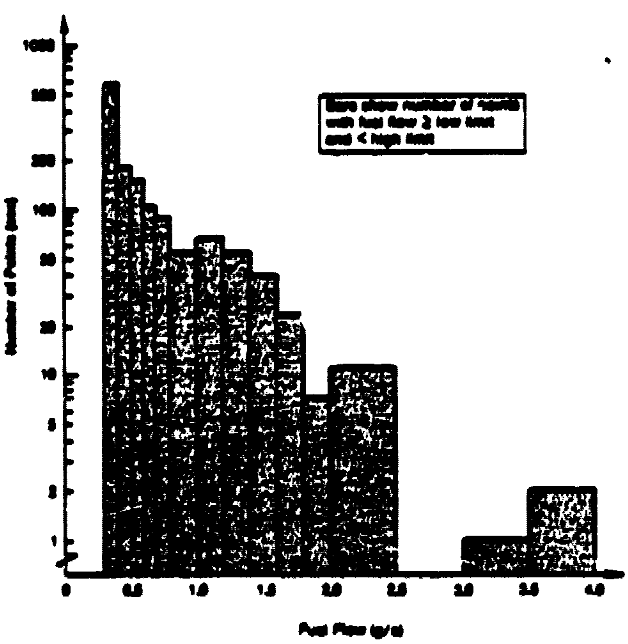

Fig. 3-21 Fuel Flow Distribution for Mod I in CVS Cycle

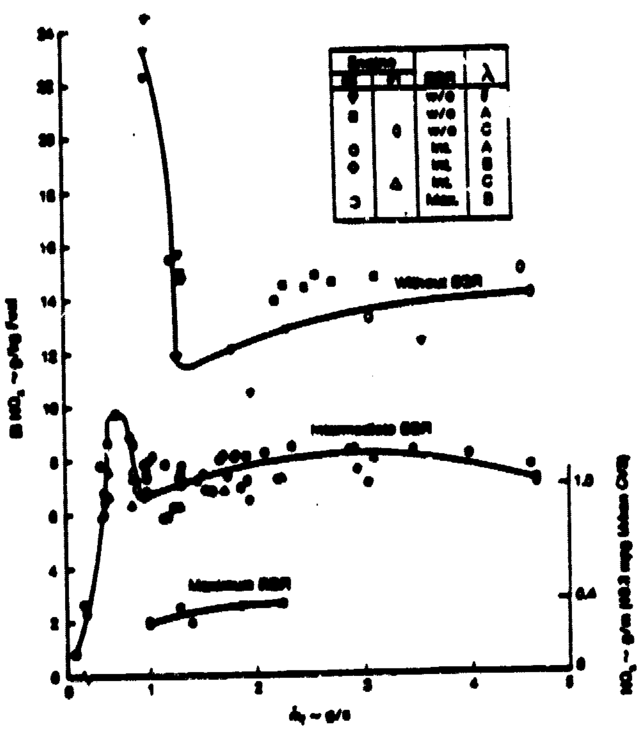

Fig. 3-22 Mod I Engine NO_x Emissions With/Without EGR

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Table 3-5

Measured/Predicted Urban Cycle Emissions and Fuel Economy
for the Mod I Lerma

	Intermediate EGR				Maximum EGR			
	NOx g/mi	CO g/mi	HC g/mi	Fuel Economy mpg	NOx g/mi	CO g/mi	HC g/mi	Fuel Economy mpg
Measured	0.96	3.43	.225	19.90	0.41	1.91	.369	20.50
Measured	0.90	3.27	.286	18.80	0.55	1.31	.718	19.40
Measured	0.84	3.21	.247	19.20	0.54	0.84	.910	19.20
Measured					0.65	0.72	.710	18.90
Measured					0.46	1.45	.490	20.60
Measured					0.46	1.72	.720	19.80
Average	0.90	3.30	.253	19.30	0.51	1.32	.653	19.70
Predicted w/o Ignition Delay	0.88	0.058	.026	19.10	0.41	0.104	.026	19.10
Predicted with Ignition Delay	--	--	.166	--	--	--	.166	--

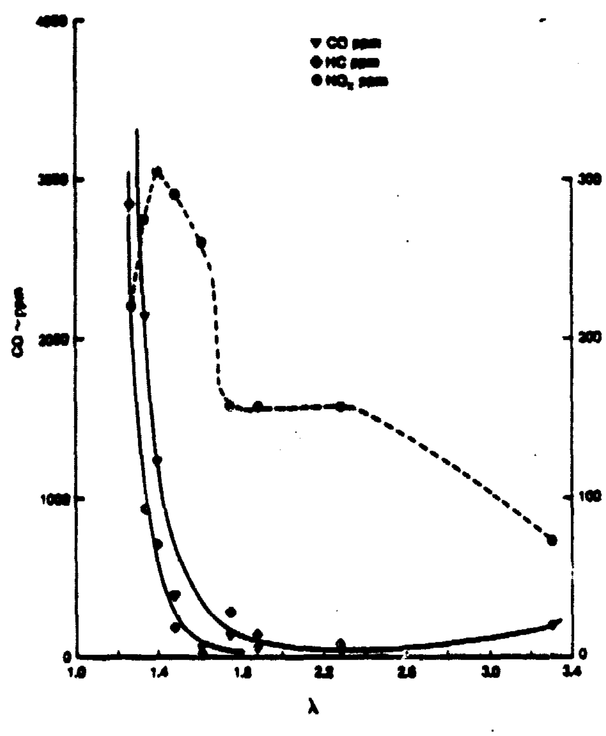

Fig. 3-23 Transient Test Bed Emissions at Quasi-Steady-State Conditions

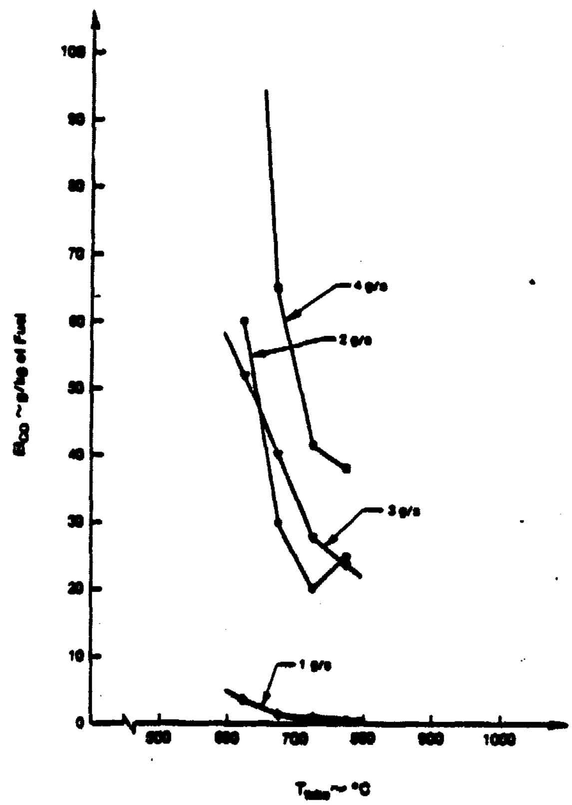

Fig. 3-24 Effect of Heater Tube Temperature on Rig CO Emissions (λ = 1.15, CGR Combustor)

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Table 3-6

Mod I Lerma* CVS Cycle Emissions and Fuel Economy Comparison

EGR Schedule	Cycle	No. of Tests	NOx (g/mi)	CO (g/mi)	HC (g/mi)	Fuel Economy (mpg)
Intermediate	Urban Highway Combined	3 3	0.90 0.66	3.30 0.311	0.253 0.0043	19.3 32.1 23.5
Maximum	Urban Highway Combined	6 10	0.51 0.32	1.32 0.12	0.653 0.007	19.7 35.8 24.7

An analysis of Upgraded Mod I steady-state emissions data with EGR has been completed. Compared to the Mod I, the Upgraded Mod I turbulator combustor has ~30% greater flow area. NOx emissions vary with fuel flow and EGR (Figure 3-25), while CO emissions are strongly dependent on λ (Figure 3-26). HC emissions were extremely low at all test conditions. Due to the lower pressure drop of the Upgraded Mod I combustor, CO and HC emissions were expected to be higher than the Mod I; however, that was not the case, as NOx (Figure 3-27), CO, and HC were comparable to Mod I levels. Since the Upgraded Mod I pressure drop is lower, there will be a reduced blower power consumption for equivalent emissions with EGR if differences in fuel economy are ignored.

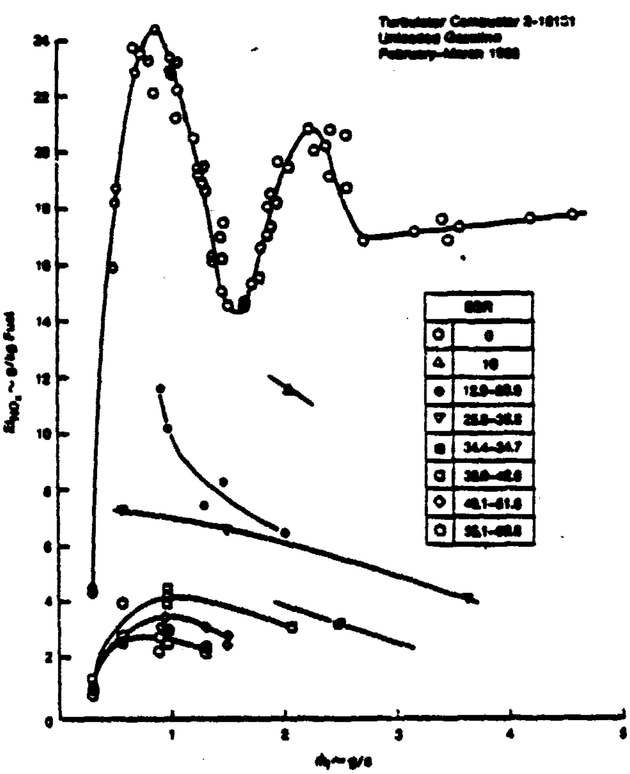

Fig. 3-25 Upgraded Mod I Engine No. 2 NO_x Emissions With/Without EGR

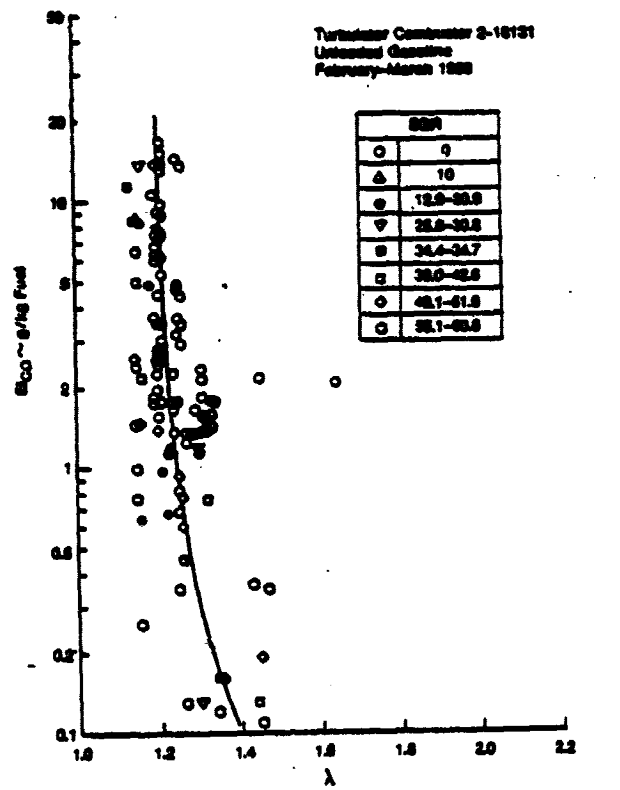

Fig. 3-26 Upgraded Mod I Engine No. 2 CO Emissions With/Without CGR

*3750-lb. inertia weight; 11.1-hp road load (improvement made in transmission shift schedule and torque converter after intermediate EGR tests)

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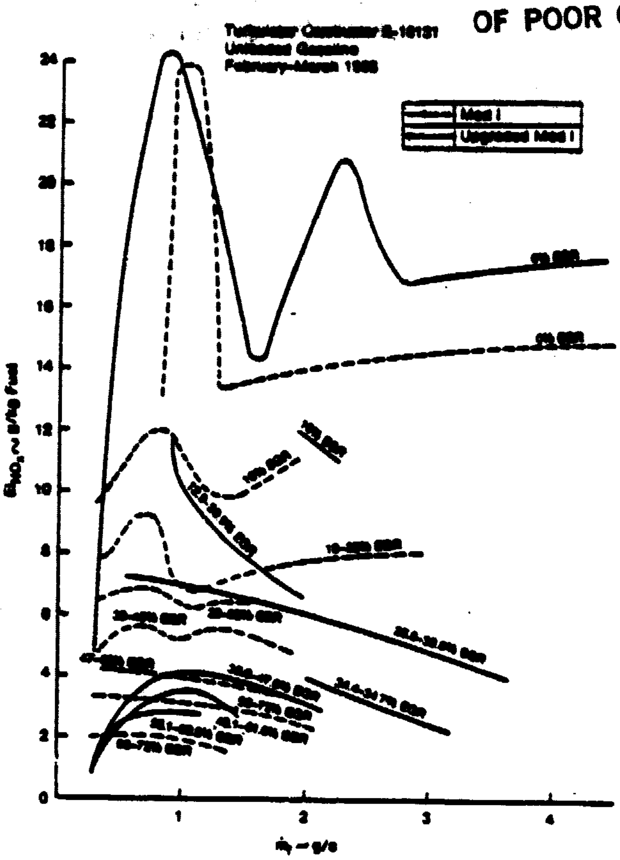

Fig. 3-27 Comparison of Mod I and Upgraded Mod I NO_x Emissions With/Without EGR

After consulting with several ceramic vendors, an existing counter-flow recuperator developed for gas turbine use (Figure 3-28) was selected for performance and durability evaluation. This cordierite block (manufactured by Coors Porcelain Company) was chosen because its anticipated performance closely matched ASE Program requirements. Evaluation was conducted in a Preheater Rig using air on the cold side and heated air, or natural-gas combustion products, on the hot side. Both heat transfer (Figure 3-29) and pressure loss (Figure 3-30) were found to be comparable to the Mod I metallic preheater. The slight deterioration in heat transfer at low flows, and the increased friction factor were due to the restrictive manifolding of the gas turbine test section, and not inherent to the heat exchanger design. Durability of the two test sections utilized was acceptable up to and including 800°C. Further increases in temperature resulted in increased leakage due to crack formation. Since temperatures up to 900°C were encountered in the Upgraded Mod I, durability was unacceptable. The cracking was judged to be geometry, as opposed to material, related.

CERAMIC MATRIX PREHEATER

The metallic preheater represents one of the most expensive components of the engine due to the inherent expense of fabricating individual metallic platelets into a matrix. In order to reduce this cost, ceramic material has been evaluated for use in the preheater. In addition, a ceramic preheater must be packageable, and must perform and have low thermal mass to yield high EHS efficiency and minimal cold-start penalty. These requirements imply an extremely high surface/volume ratio (300 ft^2/ft^3), thin walls, and low heat capacity (mass times specific heat). In order to determine the feasibility of using a ceramic preheater, performance, durability, and cost evaluations were conducted. The conclusions showed that performance is acceptable, durability needs to be improved, and there is potential for significant cost savings.

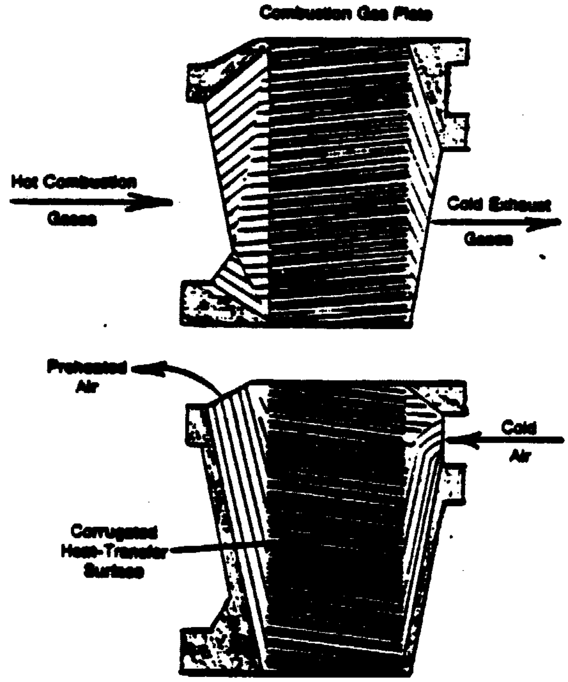

Fig. 3-28 Ceramic Preheater Test Section

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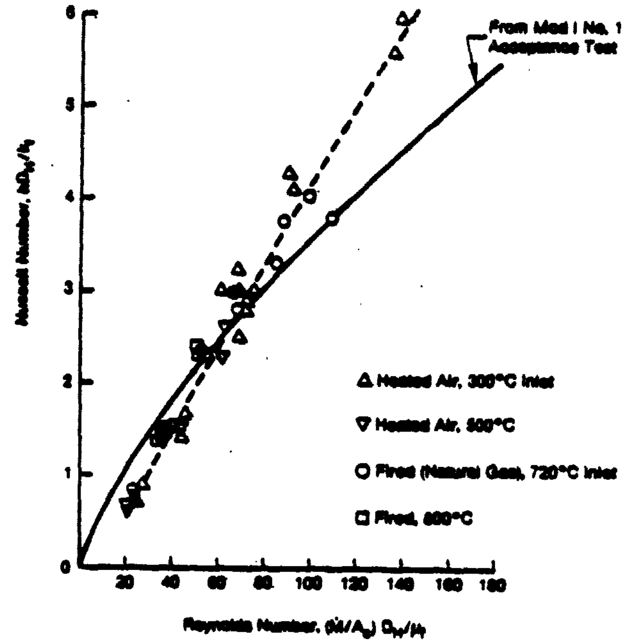

Fig. 3-29 Ceramic Preheater Heat-Transfer Test Results

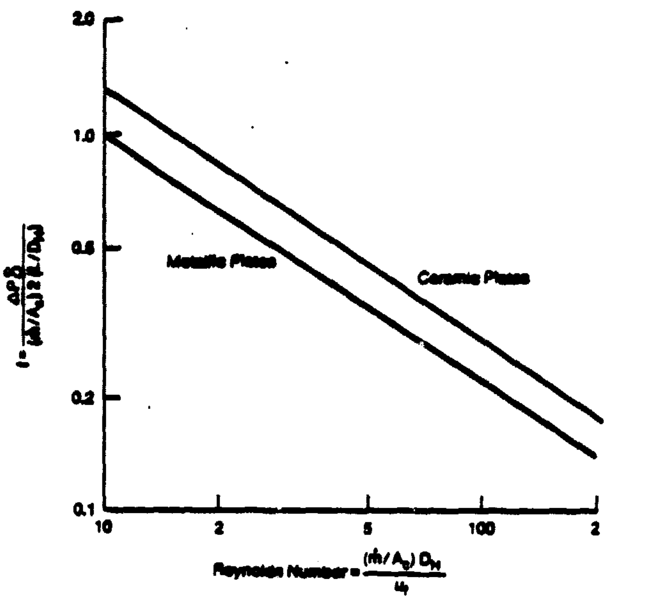

Fig. 3-30 Ceramic Preheater Pressure Drop Test Results

Based on the encouraging performance results, a mass-production RESD cost study was undertaken with Coors. The preheater design (refer to Figures 3-19 and 3-20) is simpler and more thermally stress-resistent than the gas-turbine test sections evaluated in the rig, offering ease of maintenance through the use of ten modular blocks. The study concluded a per engine cost of \$80-85 (1984 dollars) in quantities of 300,000 engines/year (a factor of 4 less than the comparable cost of a brazed metallic preheater).

Having demonstrated the cost benefits of a ceramic matrix preheater, a program will be initiated with Coors during the latter half of 1983 to supply up to eight test sections (two each of four different materials) for rig performance/durability evaluation. These sections will be single blocks of the RESD design. Initial sections will be cordierite; the remaining ceramics have yet to be determined. If successful, one of the ceramic blocks will be chosen for evaluation in an Upgraded Mod I engine during 1984.

Hot Engine System (HES) Development

The primary goals of this task are to provide a low-cost regenerator design with mileage performance comparable to the current design, and to evaluate the flow distribution of the asymmetrical heater head manifolds for the V-4 RESD.

Good regenerator performance is the key to good Stirling-engine performance; therefore, the effectiveness, pressure drop, and thermal conductance of this component must be extremely favorable for optimum engine power and efficiency. To date, these requirements have resulted in the use of a high-cost matrix material (a high mesh count, sintered wire cloth).

Another key area for engine performance is the heater head manifold. Since uniform flow within the tubes is needed for optimum performance, concern was expressed about the lack of symmetry of the V-4 RESD manifolds.

Activity during the first half of 1983 was aimed at evaluation of alternative regenerator matrix materials in both the rigs and the engine, and evaluation of the flow characteristics of the RESD heater head manifolds.

Primary objectives during the second half of 1983 will focus on further rig evaluation of alternative regenerator matrix material, testing of ceramic regenerators in a P-40 engine, and fabrication of a regenerator for the annular P-40R engine utilizing the most promising alternative regenerator material.

REGENERATOR DEVELOPMENT

Regenerators fabricated for the Mod I engine during this report period utilized Metex knitted wire in porosities of 55% to 70%. A set of 70%-porosity regenerators were tested in Mod I engine No. 10. Performance predictions were also made by incorporating the results of the Single-Blow Rig into USAB's Engine Performance Code. The results of both tests and calculations are shown in Figures 3-31 and 3-32. There was a drop of ~6 kW (~8 hp) in power and 5-1/2 points in efficiency when switching from screen to knitted wire regenerators. The calculated data matched test data quite well, especially considering that the calculation did not use actual engine conditions from the test. This test served to validate regenerator matrix Single-Blow Rig tests.

Silicon carbide (SiC) foam regenerators were designed and released for fabrication during this report period. Delays in obtaining these regenerators from the vendor resulted in postponing the P-40 engine endurance testing to August, 1983. Due to the use of incorrect values of specific heat supplied by the vendor, predicted values of engine performance were found to be in error on the optimistic side; however, since SiC regenerators offer a significant cost advantage, and would be required if an all-ceramic engine were to be developed, the P-40 endurance tests will still be performed.

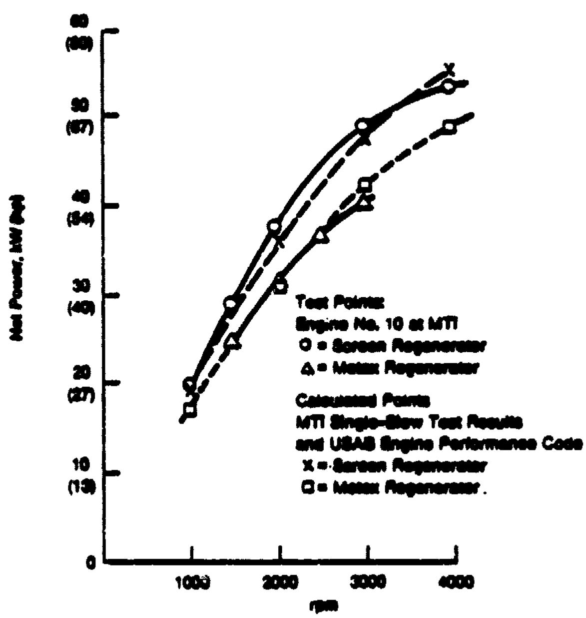

Fig. 3-31 Metex Versus Sintered Screen - Mod I Net Power Versus rpm

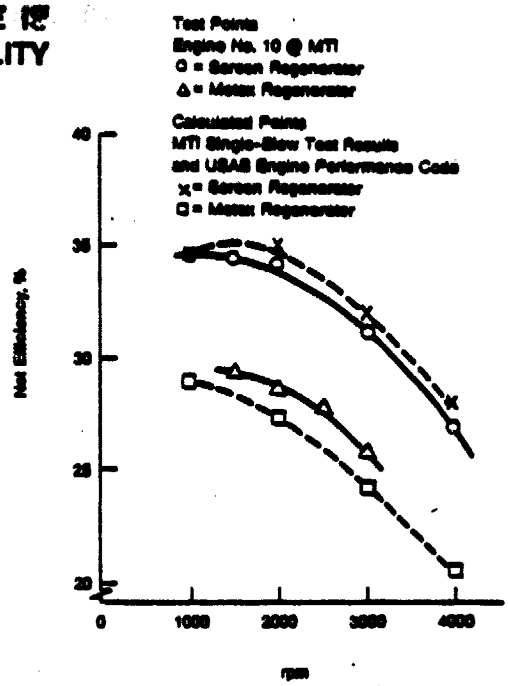

Fig. 3-32 Metex Versus Sintered Screen - Mod I Net Efficiency Versus rpm

Powdered metal test sections of two different particle sizes were also tested in the Single-Blow Rig. The results of these tests are shown in Figure 3-33. Also shown on this graph is the relative ranking of most of the test sections evaluated to date. One can see that the fine-particle-sized powdered metal sample shows good heat-transfer potential, but at the expense of significantly higher pressure drop than the baseline sintered screen. The coarse-particle sample had an acceptable pressure drop, but lower heat-transfer potential. Further samples will be evaluated to determine an optimum-particle-size test sample that should yield performance better than any alternative materials tested to date.

Further testing of alternative matrix materials is planned for the second half of 1983. Additional powdered metal, expanded metal, carbon-carbon composite material, and filament-wound wire materials will be evaluated. Further evaluation of sintered wire screen materials of differing wire sizes and porosities will also take place.

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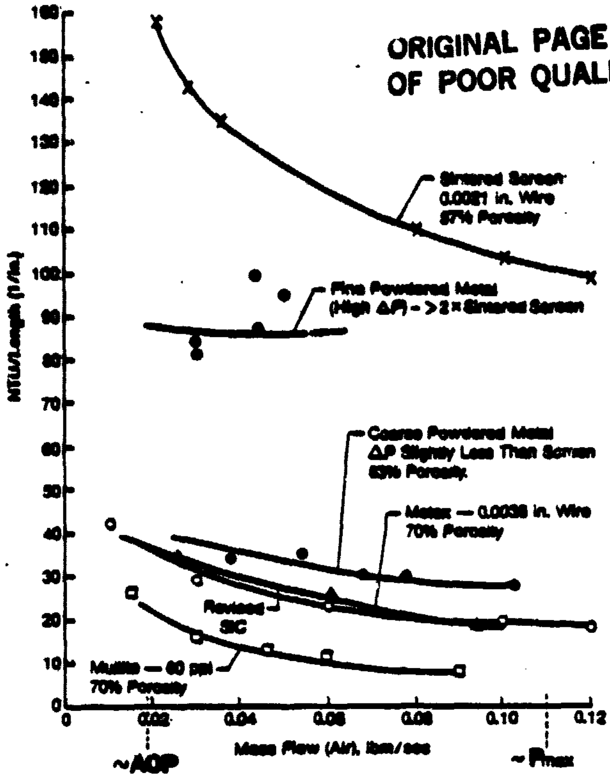

Fig. 3-33 NTU per Length Versus Mass Flow (Air) for a 3" Diameter Matrix

HEATER HEAD DEVELOPMENT

A test rig was assembled to evaluate the flow distribution of the heater head manifolds of the V-4 RESD. Flow distribution is determined by flowing water through manifold models and collecting the water in separate tubes. Water height in each tube is measured, representing the flow distribution of that manifold. Several manifold models were constructed and tested. The asymmetric V-4 manifolds had a fairly good distribution after the original geometry was slightly modified. A plot of the water height is shown in Figure 3-34, on which the regenerator and cylinder manifolds' flow distribution is averaged to give a combined effect.

In order to validate these tests, manifold models of the Upgraded Mod I heater head were constructed and tested (results are shown on Figure 3-35). This distribution is consistent with back calculated values determined from engine test data for the temperature difference between a center and end tube on the Upgraded Mod I heater head. The center tubes ran ~40°C cooler than the end tubes, indicating higher flows in the center tubes. A conclusion that can be drawn from this test is that the V-4 RESD manifolds should yield better distribution than the present Upgraded Mod I manifolds.

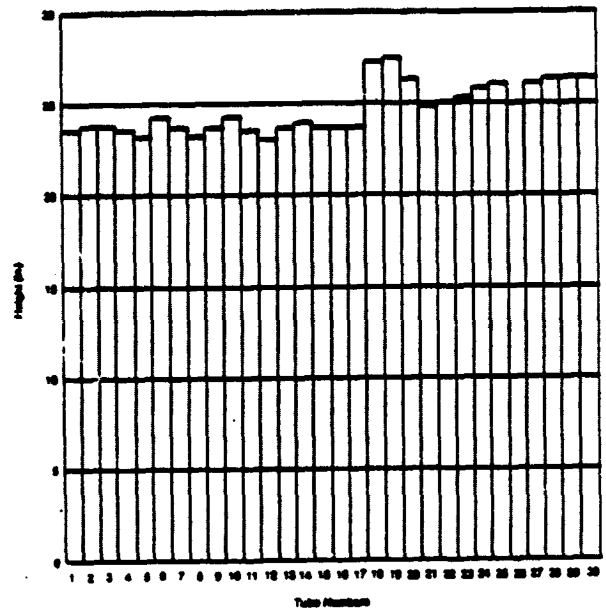

Fig. 3-34 V-4 RESD: Combined Regenerator and Cylinder Manifolds' Flow Distribution

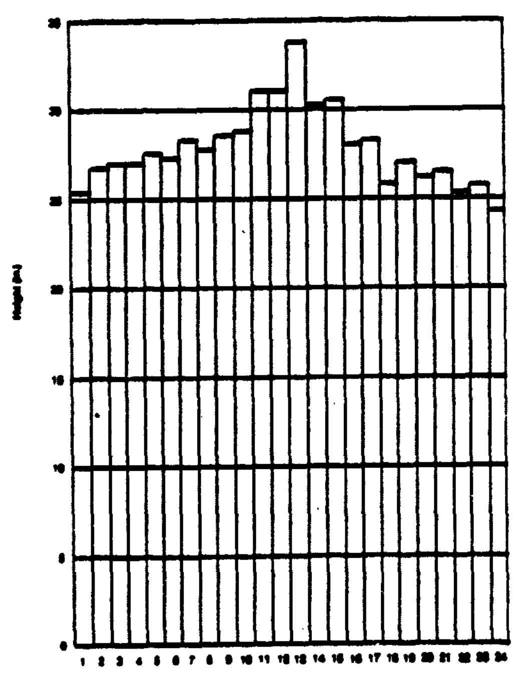

Fig. 3-35 Test Results of Upgraded Mod I Heater Head Manifold Models

Materials and Process Development

The main objective of this task is the utilization of low-cost, nonstrategic heater head materials that can survive the automotive duty cycle. The high-temperature/pressure environment, as well as the presence of high- and low-cycle cyclic stresses, have contributed to the difficulty of meeting this objective.

Developmental efforts during this report period included establishing base specifications for seal material, reducing the strategic-material content of the RESD, fatigue-testing of alternative heater head casting alloys, the manufacture of a test rig to hydraulically fatigue-test heater head castings, continuation of the creep rupture testing of alternative heater tube materials, and refitting the quadrant for the HTP-40 testing of alternative casting alloys with new heater tubes.

During the second half of 1983, material development efforts will focus on completing the fatigue testing of welded alloy XF-818, establishing a microstructure calibration chart to help determine time at temperature of heater tubes, fatigue-testing heater head castings and a crosshead/piston rod joint, determining the thermal conductivity of XF-818, continuing the creep rupture testing of alternative heater tube materials, and experimenting with thermal-barrier coatings on External Heat System components.

DESIGN PROPERTIES TESTING

The mean stress effects and hold time, or frequency, effects on the fatigue life of XF-818 were determined during this report period. Figure 3-36 is a plot showing the cycles to failure versus strain amplitude percentage for both the previously done Phase I and the current Phase II fatigue testing.

A Haigh Diagram (shown in Figure 3-37), constructed from the results in Figure 3-36, shows the effect of the mean stress for a given alternating load. The heater head castings operate at different R ratios (R = minimum stress/maximum stress). These data will be useful in optimizing heater head casting designs.

The effects of frequency are shown in Figure 3-38, which is a plot of frequency versus cycles to failure for XF-818. The stress amplitude temperature and R ratio were constant for all tests. The results show a decrease in low-cycle fatigue life of over two, with a decrease in frequency of two orders of magnitude (significant when considering thermally-induced stresses and mean working gas pressure changes in design of the heater head castings).

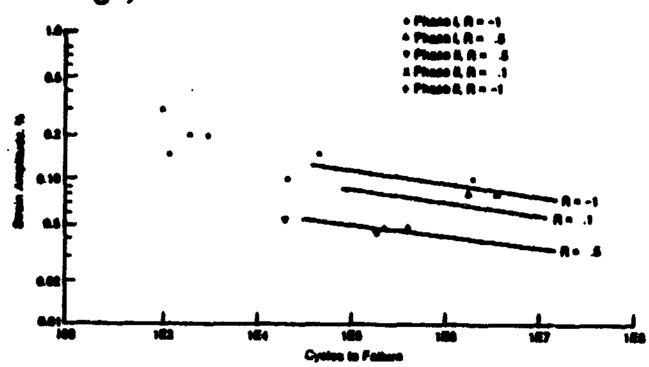

Fig. 3-36 Mean Stress Effects - XF-818 (800°C)

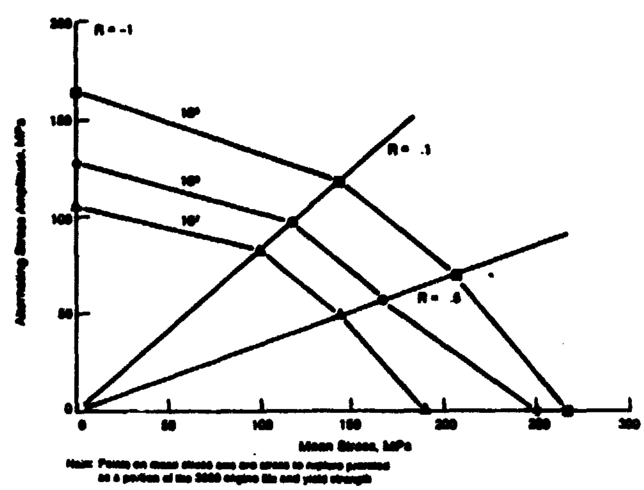

Fig. 3-37 Haigh Diagram - XF-818 (800°C) Air

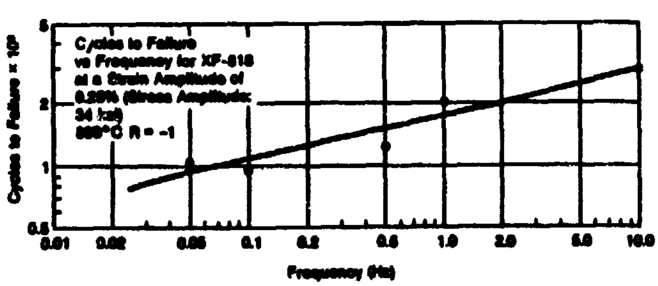

Fig. 3-38 Frequency Dependence of Low-Cycle Fatigue Life for XF-818

To examine the effects of high-pressure hydrogen on the high-cycle fatigue life of alloys XF-818 and CRM-6D, a special hollow fatigue test specimen (Figure 3-39) was designed that was internally pressurized to 19.65 MPa (2.85 ksi) with either hydrogen or helium.* The results are shown in Figure 3-40. This test showed no apparent degradation due to high-pressure hydrogen.

Long-term creep rupture tests are being conducted on the five alternative heater tube alloys. Table 3-7 lists these alloys and their nominal composition. The latest 850°C creep rupture data have been added to the curves shown in Figure 3-41. The stress to rupture at 3500 hours has been estimated for each alloy based on this latest data (shown in Table 3-8). The estimate is a best straight-line fit through the data points using trend analysis. The estimate does not take into account any scatter of the data; it considers only the solution-annealed condition, except for CG-27, which is solution-annealed and precipitation-hardened.

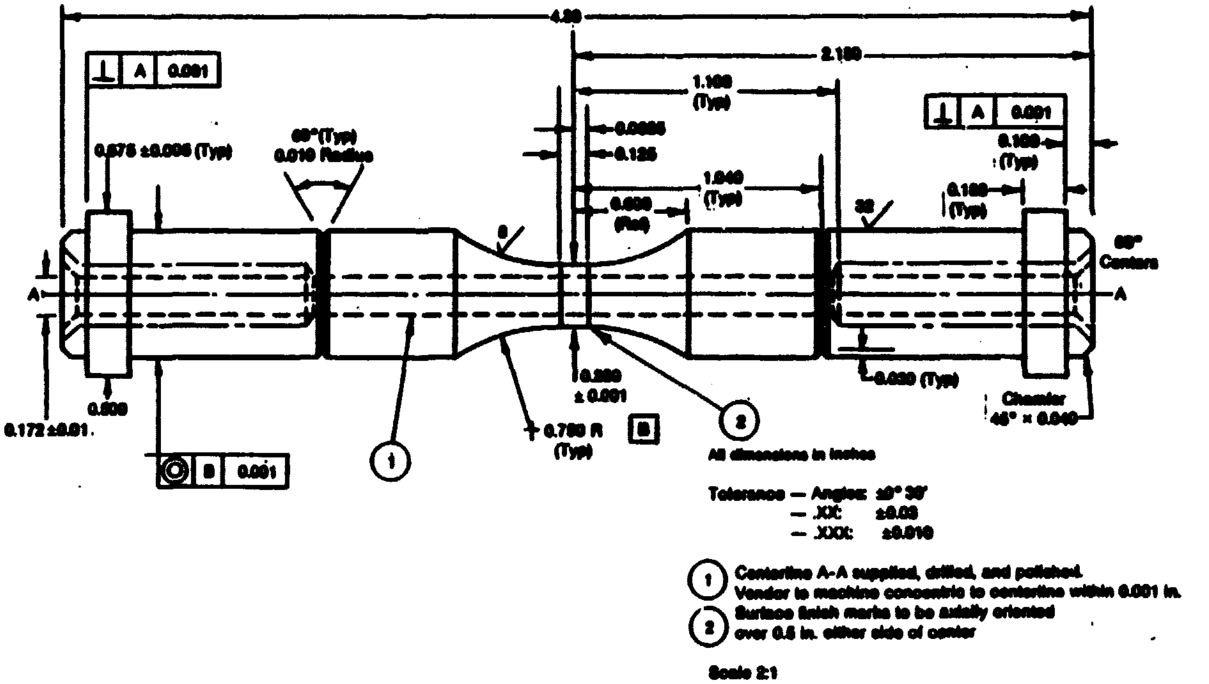

Fig. 3-39 Fatigue Test Specimen

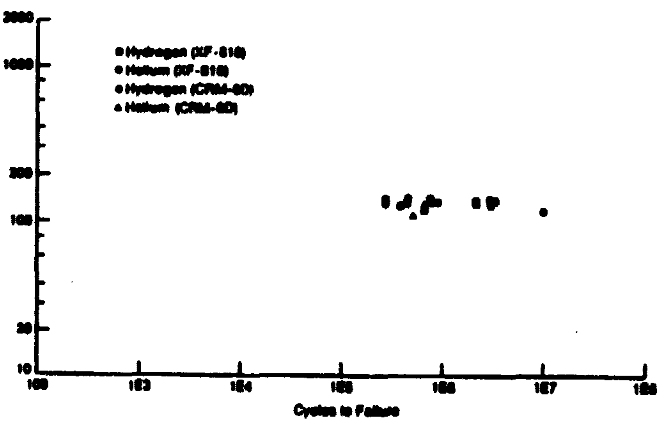

Fig. 3-40 Fatigue in Hydrogen and Helium - XF-818 and CRM-6D (800°C)

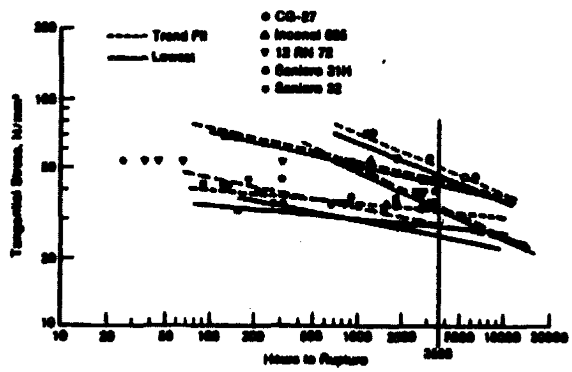

Fig. 3-41 Creep Rupture Testing (850°C) Argon - Sanicro 31H, Sanicro 32, CG-27, Inconel 625, 12RN72

*used as a reference to equate to air properties

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Table 3-7

Alternative Heater Tube Materials' Nominal Chemistry

Alloy	Co	Cr	Ni	Mo	W	C	Al	Ti	B	Cb	Mn	Fe	Si	N
Multimet[1] (N155)	19.75	21.25	20	3.00	2.5	.12	-	-	-	1.0	1.50	29.70	1.0	.15
Alternatives:														
CG-27[2]	None	13.00	38	5.75	-	.05	1.6	2.5	.010	0.7	-	38.00	-	-
Inconel 625[3]	None	21.50	61	9.00	-	.05	0.2	0.2	-	-	0.25	2.50	0.2	-
Sanicro 32[4]	None	21.00	31	-	3.0	.09	0.4	0.4	-	-	0.60	42.80	0.6	
Sanicro 31H[4]	None	21.00	31	-	-	.07	0.3	0.3	-	-	0.60	46.13	0.6	
12RN72[4]	None	19.00	25	1.40	-	.10	-	0.5	.006	-	1.80	51.80	0.4	

[1]Base Material
[2]Crucible Steel Corporation
[3]International Nickel
[4]Sandvik Alloys

Using the same slope as the trend-fitted line, and drawing a line through the lowest data point, yields the values listed in Table 3-8 as the lowest estimate stress. The design stress, based on an equivalent operational pressure of 7.5 MPa and a safety factor, is 28.125 N/mm^2. Considering this, Sanicro 31H and 32 will not meet the requirements; all the other alloys appear to meet the requirement for creep rupture at this time.

Table 3-8

Estimated Stress to Rupture at 3500 Hours (850°C)

Alloy	Estimated Stress Trend Fit (N/mm^2)	Lowest Estimated Stress (N/mm^2)
CG-27	52.46	46
Inconel 625	45.31	43
12RN72	34.34	32
Sanicro 31H	33.25	27
Sanicro 32	28.07	24

HIGH-TEMPERATURE ENGINE TESTING

The alternate heater head casting and heater tube materials are being subjected to high-temperature engine testing on an HTP-40 engine to rank the alternate materials in an engine environment. Eight quadrants have been manufactured with the combinations of casting and tube materials shown in Table 3-9.

Table 3-9

Tube/Casting Materials

Quadrant No.	Casting Material	Tube Material
1	HS-31	N-155
2	CRM-6D	N-155
3	XF-818	Inconel 155
4	SAF-11	CG-27
5	HS-31	N-155
6	CRM-6D	Inconel 155
7	XF-818	N-155
8	SAF-11	Sanicro 31H

To date, 1650 test hours have been accumulated against the test goal of 2000 test hours at 820°C.

A heater tube (made with Sanicro 32) failure interrupted the last 500-hour phase of the 2000-hour test plan. After reviewing the engine test results and the creep rupture test results of the alternative heater tube materials, it was decided to replace all the remaining Sanicro 32 tubes on the affected quadrant with N-155. Since the primary goal of this task is to test the alternate cast materials, the weaker alternate tube materials on three other quadrants will also be replaced with N-155. The quadrant material combinations will then be as shown in Table 3-9.

RESD STRATEGIC-MATERIAL CONTENT

One of the efforts in this task has been to reduce the amount of strategic elements* in the Stirling engine. By substituting alternative heater head casting alloy XF-818 for HS-31, and alternative heater tube alloy CG-27 for N-155, cobalt has essentially been eliminated. Chromium has also been substantially reduced. Table 3-10 compares the amount of strategic elements in the Mod I, the Upgraded Mod I, and the RESD. As a proof test of design and manufacturing capability, the heater head castings will be subjected to a full-scale fatigue test. The internal pressure will be applied hydraulically through the testing shown schematically in Figure 3-42. The test will be conducted at room temperature, and will apply a mean pressure of 5 MPa for 10^7 cycles.

*cobalt, chromium, columbium, tantalum

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Table 3-10

Strategic Materials

	Mod I		Upgraded Mod I		V-4 RESD	
	kg	%*	kg	%*	kg	%*
Co	10.15	4.9	0.063	-	-	-
Cr	16.40	8.0	12.90	6.5	6.047	7.0
Cb	0.095	0.1	0.134	0.1	0.108	0.1
Ta	-	-	-	-	-	-
Co = cobalt, Cr = chromium, Cb = columbian, Ta = tantalum						

*percent of total basic engine weight

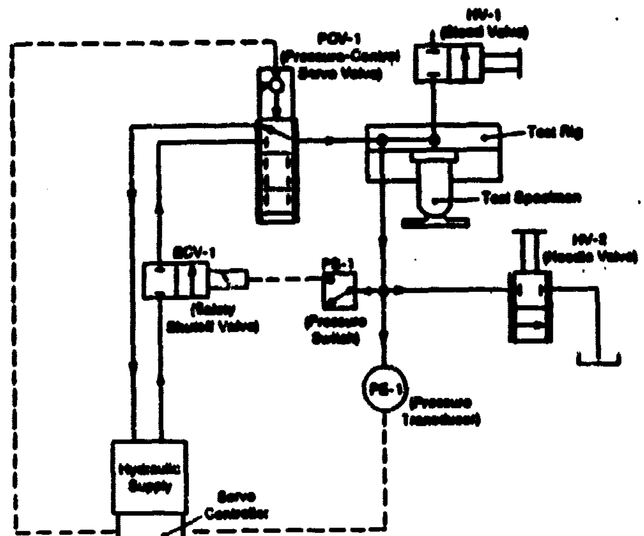

Fig. 3-42 Schematic of Hydraulic Fatigue Tester

Cold Engine System (CES) Development

The objective of CES activity is to develop reliable, low-friction, long-lived seals. Development activity covers piston rings and main (piston rod) seals.

During the first half of 1983, efforts to develop low-friction piston rings were concentrated in the Mod I Motoring Rig. Evaluation and development of advance-design main seals has been carried out in the Exploratory Seals Test Rig, and a statistical evaluation of Pumping Leningrader (PL) seal life was initiated using two P-40 engines.

The goal for the remainder of 1983 is to advance the development of main seals and piston rings through rig and engine testing.

MAIN SEALS

USAB experience has shown that main seals might fail after a short period of operation; however, if the seals do not fail prematurely, there is a high probability that they will have a life exceeding 1000 hours. In the early stages of Mod I testing, main seal failures occurred regularly after short periods of operation. Evidence suggested that this might have been the result of changes in the HABIA seal material properties caused by inconsistencies in the manufacturing process.

In an attempt to clarify this and generate seal life data under representative controlled conditions, a series of tests were initiated using two P-40 engines that were run for a nominal 500 hours using a duty cycle representative of CVS conditions (shown in Figure 3-43).

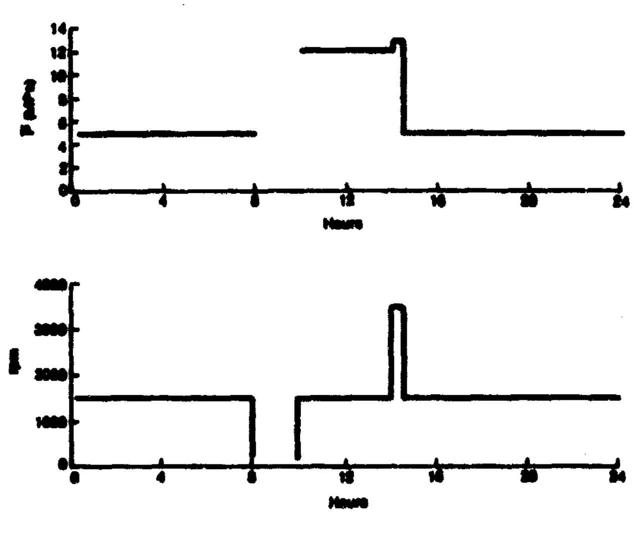

Fig. 3-43 Initial Main Seal Duty Cycle (P-40 Engines)

The first tests were carried out with PL seals made from HABIA material manufactured by a process specified by USAB. Three sets each of these seals completed 500 hours without a single seal failure or degradation in engine performance. At the end of testing, traces of oil were present in the seal housings, but no oil had entered the working cycles of the engines. In all cases, a feather edge had formed on the seals where they had been extruded between the piston rod and seal seat (cracks were also found in this region). During this time period, seals made from the same batch of HABIA material failed after short intervals in Mod I engines, indicating the differences were not the result of material properties.

The Mod I engines are all basically performance engines operating under conditions that are much harder on the seals than the CVS mode of operation used in the P-40 tests. The Mod I engines operate for longer periods at high power levels, and experience more frequent starts and stops. To investigate the effect of this type of operation on seal life, an accelerated duty cycle was devised (Figure 3-44). During each twenty-four-hour period, the cycle had two hot starts (engine is restarted after a ten-minute stop) and two cold starts (engine is restarted after a one-hour stop). For more than sixteen hours of every twenty-four, the engine operated at 12 MPa/2500 rpm. With this accelerated duty cycle, a set of HABIA seals completed 500 hours without failure or degradation in engine performance. At the end of the test, oil leakage and seals' conditions were the same as those seen with the original duty cycle. While the P-40 tests yielded valuable information, they did not impact the prime objective, i.e., they did not provide any explanation for the short-seal life experienced with the Mod I engines. At this stage, it would appear that the differences between the P-40 and Mod I engine designs may be significant in terms of seal life. Testing a Mod I engine with the same duty cycles should resolve this question. Mod I engine No. 2 is currently under assembly for this purpose.

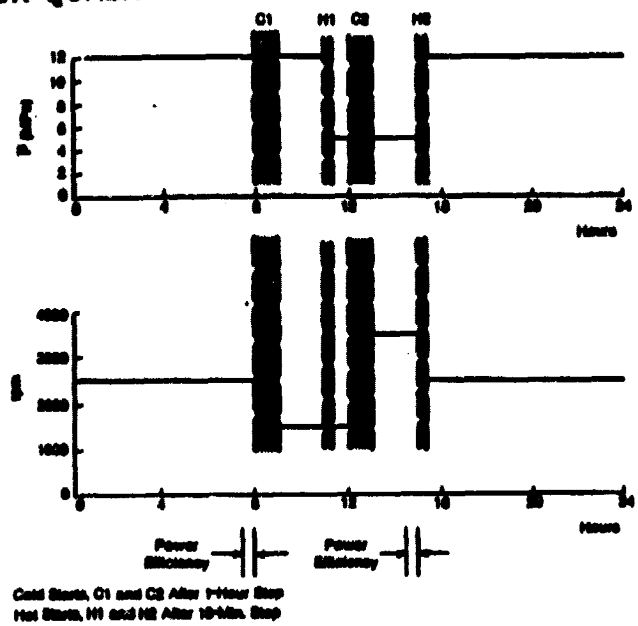

Fig. 3-44 Accelerated Main Seal Duty Cycle (P-40 Engines)

Exploratory Seals Rig testing during this report period concentrated on evaluating seal designs based on the double-angle principle. These tests are summarized in Table 3-11, and the different forms of seals are shown in Figures 3-45 through 3-49. All seals were tested with intermittent operation using the new test head with Mod-I-type hardware.

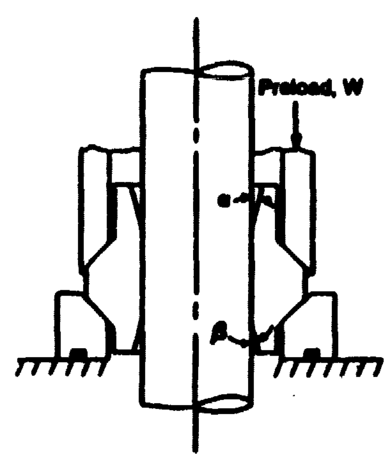

Fig. 3-45 Double-Angle Seal Design Variation

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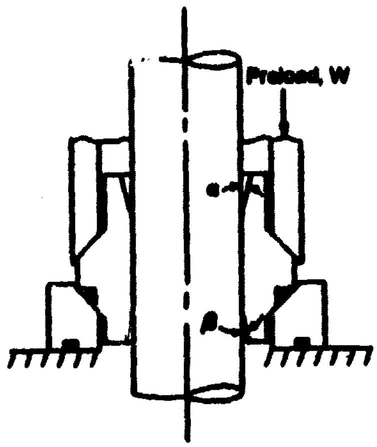

Fig. 3-46 Double-Angle Seal Design Variation

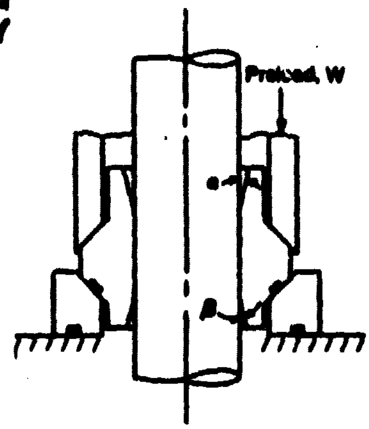

Fig. 3-47 Double-Angle Seal Design Variation

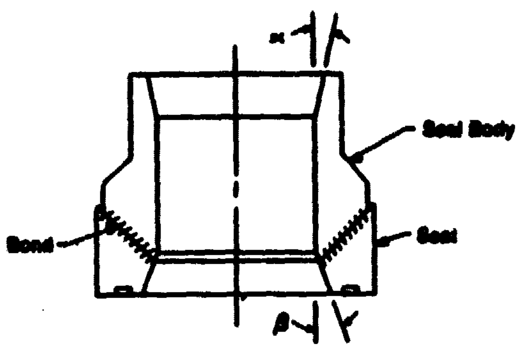

Fig. 3-48 Double-Angle Seal Design with Conformable-Filled PTFE Materials

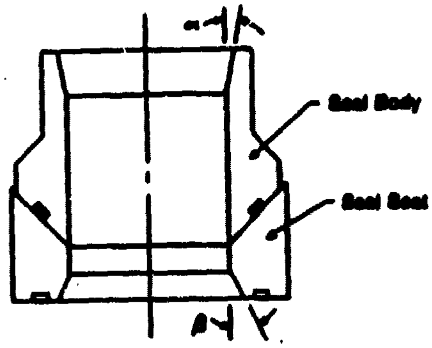

Fig. 3-49 Double-Angle Seal Design Variation

Table 3-11

Exploratory Seals Rig Testing Summary

Seal* Set	Seal** Type	Seal Body Material	Seal Seat Material	Preload (lb)	Test Duration (hrs)
22	1	Rulon J	Brass	40	72
23	2L	HABIA	Brass	80	514
24	2	Torlon 4301	Brass	80	24
25	4	Rulon J	Brass	80	10
26	4	Rulon J	Brass	80	1
27	4	Rulon J	Brass	80	2
28	3	Torlon 4203	Brass	100	181
29	3	Torlon 4203	Brass	120	73
30	3	Torlon 4203	Brass	120	3
31	5	Rulon J	Torlon 4203	80	1
32	5	Torlon 4203	Brass	100	5
33	5	Rulon J	Torlon 4203	100	11
34	5	Rulon J	Torlon 4203	120	8

*two seals per set
**refer to Figure 3-49 for all double-angle seals tested $\alpha = 5°$, $\beta = 15°$

To provide baseline data, a pair of standard HABIA PL seals were tested (Seal Set 23). Hydrogen leakage decreased during the initial test period until it reached a low equilibrium level (see Figure 3-50). The test was terminated after 514 hours. During this period, there was a small leakage of oil past the upper seal; both seals had worn as indicated by a loss in weight, but neither seal had extruded.

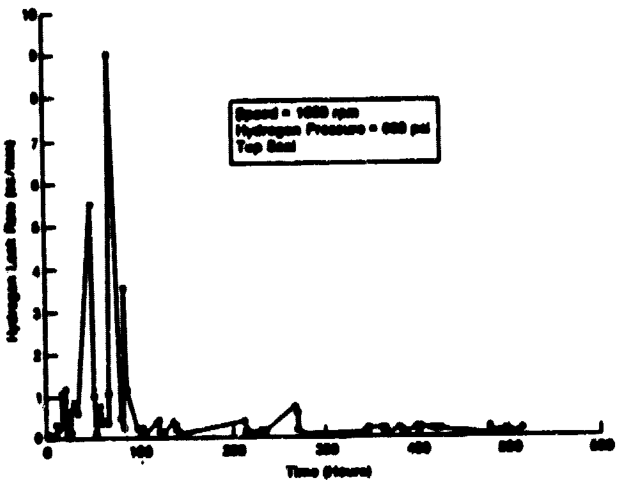

Fig. 3-50 Seal Set 23 Hydrogen Leakage Rate

Previous tests of double-angle seals had shown that leakage began after some period due to deformation of the seal, which essentially eliminated the lower inlet angle (β). Seal Set 22 (Figure 3-45) was an attempt to overcome this problem. The lower inlet angle was formed in an extension of the seal that was not in a loaded region. After 72 hours, an unacceptable oil leak was detected. The lower inlet angles were still present, but the seals had suffered gross extrusion, i.e., the extended sections of the seals were approximately twice their original length, and had circumferential cracks.

In an attempt to overcome the extrusion problem, the seal material was changed to Torlon 4301 (Seal Set 24), a polyamide-imide material that is much harder, and has a modulus of elasticity approximately four times that of filled PTFE* materials. To maintain an effective seal between the seal and brass seat, a 1/16-inch o-ring was provided and located in a groove in the lower conical surface of the seal (shown in Figure 3-46). With the higher modulus material, the interference fit between the rod and seal was reduced. At start-up under cold conditins, the seals gave very high friction. Seal temperatures increased rapidly to 100°C, and then the friction and seal temperatures decreased until they reached steady-state values very similar to those of PTFE seals. The lower seal had slightly more interference than the upper seal, and ran at a slightly higher temperature, but it gave consistently low hydrogen leakage (Figure 3-51). The upper seal gave higher gas leakage initially, which progressively decreased until, after about nine hours, it was at the same level as the lower seal (Figure 3-51). Testing was terminated after 24 hours when oil leakage was detected. Examination showed that most of the oil had come from the upper seal and had entered through the seal/seat interface. The introduction of the o-ring groove in the seal had severely reduced the area of the contact surface between the seal and the seat. With the high starting friction, there was a strong possibility that the seals had not remained in contact with the seats, and their motion had generated a squeeze film effect which pumped the oil past the o-rings.

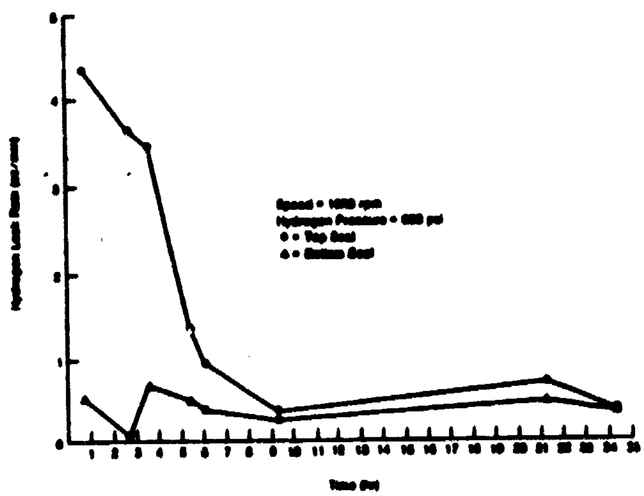

Fig. 3-51 Seal Set 24 Hydrogen Leakage Rate

Seal sets 28, 29, 30, and 32 are modified designs of Torlon, double-angle seals. The main design change was the use of a much smaller o-ring (0.5 mm) (Figure 3-47); thus, the reduction in contact area was minimal. Seal set 28, with a smaller interference, gave higher gas leakages than those experienced with Seal Set 24 (Figure 3-52); however, leakage

*polytetraflouroethylene

from both seals decreased during testing, and was less than 33 cc/min when the test was terminated after 181 hours when slight oil leakage was detected. Examination showed that the small o-ring, combined with the slightly higher preload, had effectively prevented oil passing between the seal and seat. The oil leakage that had occurred between the seals and the rod was minimal, and only sufficient to produce a thin film on the components close to the seals.

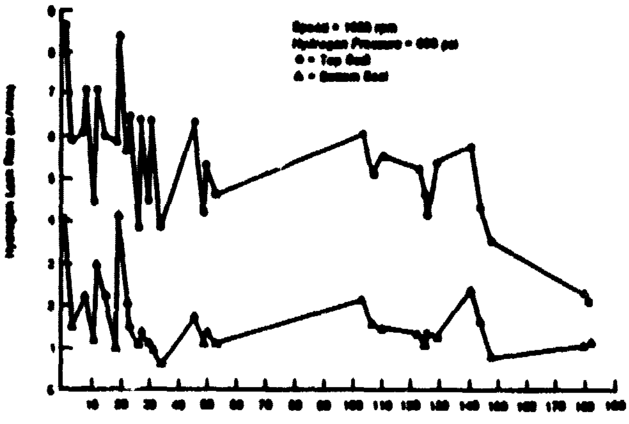

Fig. 3-52 Seal Set 28 Hydrogen Leakage Rate

The seals in Set 29 were of the same design as Set 28, but were selected to have a slightly greater interference fit on the rod and, to compensate for the expected increase in friction, the preload was increased. The increased interference resulted in higher friction, but gave lower gas leakage (mainly less than 1 cc/min). Testing was terminated after 73 hours when oil leakage was detected. Subsequent examination showed minimum leakage of oil, which had entered between the seals and the rod. At the end of the test, the seals were still an interference fit on the rod, but their bores had increased slightly.

Seal Set 32 was nominally a repeat of Seal Set 28. The seals had a comparable interference fit, and the preload was 100 lbs. Gas leakage was low, but testing was terminated after five hours when significant oil leakage occurred.

Reasons for the variability in performance of the Torlon seals are still being investigated, but it would appear that small differences in roundness of the seal bores may be a significant factor.

An alternative seal design aimed at minimizing the extrusion of the seal while utilizing conformable-filled PTFE materials is shown in Figure 3-48. The seal body is bonded to a brass seat and machined as a unit. Seals of this type were manufactured for test using Rulon J for the seal body. The dimensions were chosen to give a nominal radial clearance between the brass base and the rod surface when assembled. Initial tests had very high gas leakages which were traced to leak paths through the bonded interfaces. This problem was overcome by applying a bead of silicone sealer to the outer edge of the Rulon J/brass joint. Subsequent tests (Seal Sets 25, 26, and 27) with this type of seal gave very poor results, i.e., large oil leakages occurred after very short periods of time.

During the tests, friction forces and seal temperatures were very similar to those experienced with PL seals, and gas leakage was very low. At the end of the tests, the seals were not worn, nor had they deformed. The seals and other components were thoroughly wet with oil. It was not possible to determine the path by which the oil had entered the seal cavity; however, there is a distinct possibility that it passed through the bonded area. High pressures generated in the inlet section on the upstroke would be sufficient to pump oil through any potential leak path in the bond against the opposing gas pressure. While there was no direct evidence that this had occurred, it is quite feasible that this mechanism could produce oil leakage without incurring any significant or excessive gas leakage.

Another variation on the double-angle seal concept is shown in Figure 3-49. In this arrangement, the seal and seat are separate components, but the lower inlet angle is formed in the seat so that the two components combine to function as a double-angle seal. Seals of this type were manufactured using Rulon J for the seal body and Torlon 4203 for the seat.

Seal Set 31 had a large inteference between the seal body and rod, and the seat had a small interference. These seals exhibited high friction, and oil leakage was detected after one hour. There was reason to believe that the high friction was caused by the interference fit between the seat and rod, possibly causing the seats to lift and produce the oil leakage. A second set of seals was manufactured with a smaller diametral clearance between the seat and rod (Seal Set 33). Under test, these seals gave low friction and gas leakage (less than 0.5 cc/min), but oil leakage was detected after 11 hours. A third set of seals (Seal Set 34) with an even smaller diametral clearance between the seat and rod were tested with a higher preload of 120 lbs. They exhibited low friction and gas leakage, but oil leakage was detected after 8 hours. In Seal Sets 33 and 34, the friction forces were much smaller than the applied preload, so there was no reason to suspect that the seals might be unseated; however, it is possible that the hydrodynamic pressures generated in the rod/seal interface were sufficient to pump oil between the seals and seats without incurring high gas leakage.

The double-angle seal is a proven concept that can provide consistent, controlled lubrication to minimize friction and gas leakage. The major problem is to maintain the geometry of the seal, particularly that of the inlet region on the crankcase side. During the second half of 1983, rig testing will concentrate on evaluating double-angle seal designs aimed at achieving this objective.

PISTON RINGS

Piston ring development during the first half of 1983 concentrated on the pressure-balanced H-ring concept. Tests have been carried out in the Mod I motored engine, and are described in the Engine Drive System Development Section of this report.

Engine Drive System (EDS) Development

The primary goals of this task are to develop a Reduced Friction Drive (RFD), and evaluate new seal concepts under motored engine test conditions. Activity began in the latter half of 1981 with the installation of a motored Mod I drive system with a dummy heater head.

Efforts completed during this report period include continued testing and evaluation of seal hardware, a complete remapping of EDS motoring friction for the baseline journal bearing unit with split/solid piston rings, retrofit of the RFD with Lightweight Reduced Friction Drive (LRFD) crankshafts, connecting rods, and piston rods, and a complete remapping of the motoring friction for the LRFD with split/solid piston rings.

During the latter half of 1983, work will continue on testing and evaluating various seal configurations, including motored EDS performance with capseals removed, split/solid rings with P_{min} cylinder liner vent, and single H-rings.

SEAL HARDWARE TESTS

Table 3-12 lists the various H-ring tests that have been or are going to be conducted as part of the seal evaluation/development effort.

Table 3-12

H-Ring Tests

Test	Ring Type	Seal Type	Expander Type
1	Basic H-Ring (Split)	1/16" O-Ring	Flat Ring
2	L H-Ring (Split)	1/16" O-Ring	Wave Spring
3	L H-Ring (Solid)	1/16" O-Ring	Wave Spring
4	L H-Ring (Split)	Quad-Ring	Wave Spring
5	L H-Ring (Split)	1/16" O-Ring	Double Quad-Ring
6	L H-Ring (Solid)	1/16" O-Ring	Double Quad-Ring
7	L H-Ring (Solid)	3/32" O-Ring	Double Quad-Ring
8	L H-Ring (Split)	3/32" O-Ring	Double Quad-Ring
9	Single H-Ring	1/16" O-Ring	Double Quad-Ring

The first configuration tested, the basic H-ring (shown in Figure 3-53), showed evidence of o-ring fretting after four hours of operation. The fretting was thought to be caused either by cyclic expansion/contraction of the o-ring in response to the pressure wave or by the o-ring contacting the cylinder wall during a pressure transient.

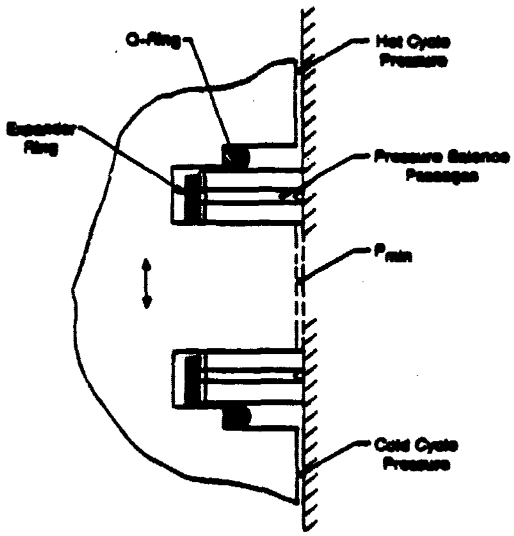

Fig. 3-53 Basic H-Ring System - Two Rings/Piston

The second configuration tested, the split, L-shaped H-ring shown in Figure 3-54, was adopted to provide o-ring retention. This would assure that the o-ring did not contact the cylinder wall. Cycle-cycle pressure balance was poor with this configuration, and o-ring fretting was once again observed after four hours of operation as a result of this test. Cylinder wall contact was eliminated as a cause of the o-ring fretting.

The third configuration tested was identical to the second except that the H-ring was solid. O-ring fretting was also observed after four hours of operation. A degradation of cycle-cycle pressure balance occurred as the o-ring abrasion progressed.

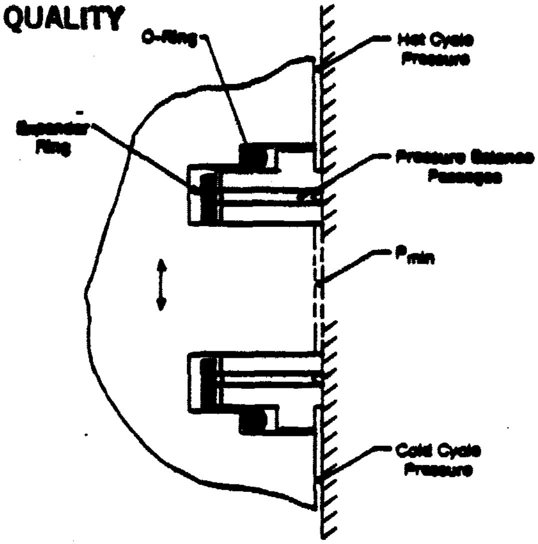

Fig. 3-54 L-Shaped H-Rings (O-Ring Retention) Two Rings/Piston

The fourth configuration tested was the split, L-shaped H-ring with the o-rings replaced with quad-rings, as shown in Figure 3-55. Cycle-cycle pressure balance was still unsatisfactory. It was discovered on disassembly that the quad-rings had twisted during assembly and had not provided an effective seal.

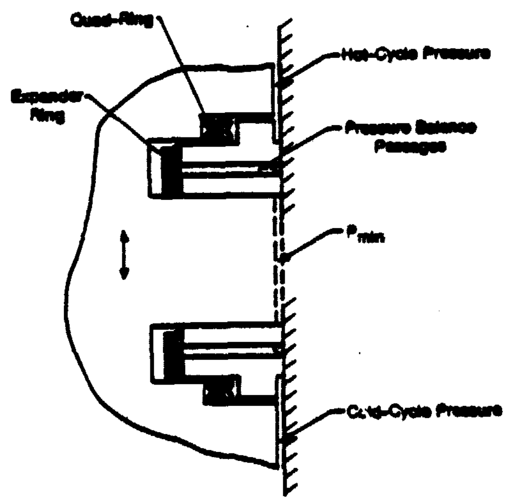

Fig. 3-55 L-Shaped H-Rings (Quad-Ring Retention) - Two Rings/Piston

The fifth configuration tested was the split, L-shaped H-ring (Figure 3-56) with double quad-rings used as expander rings. Excessive cycle-cycle pressure imbalance was noted, and o-ring fretting was observed on disassembly.

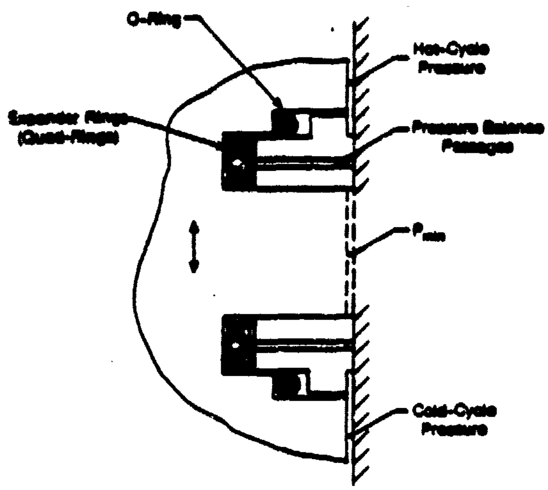

Fig. 3-56 L-Shaped H-Rings (O-Ring Retention - Two Rings/Piston

The sixth configuration tested was identical to the fifth except that the H-ring was solid. Performance was not measurably improved over the prior test. It was concluded that insufficient axial squeeze of the o-ring was allowing gas to bypass the ring at higher operating pressures (9 MPa and above), thus producing the cyclic expansion/contraction of the o-ring and the resultant ring abrasion.

The o-ring cross-section diameter was increased by 50% (from 1/16" to 3/32") for the seventh test, and a solid, L-shaped H-ring was used. The identical hardware, with the H-ring split, was used for the eighth test. No o-ring abrasion occurred in either test; however, the cycle-cycle pressure balance, which was excellent with the solid ring, was seriously degraded by the addition of the small leak path through the split in the ring. Figure 3-57 shows the cycle pressures and torque for the solid H-ring tested at 7 MPa mean pressure and 2000 rpm, and Figure 3-58 shows the cycle pressures and torque for the split H-ring tested at identical conditions. The motoring torque was higher for both tests than was anticipated for a properly designed H-ring system. The direct substitution of a 50% larger cross-section o-ring effectively eliminated leakage; however, the piston groove depth was insufficient to provide adequate radial clearance for the larger o-ring. This made the ring radial loading dependent on pressure, which is contrary to the H-ring design philosophy. Ring grooves will be modified to provide adequate o-ring radial clearance prior to the resumption of testing.

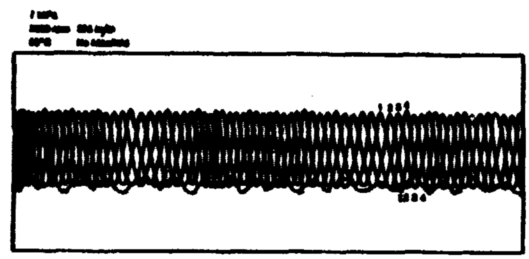

Fig. 3-57 Cycle-to-Cycle Pressure/Torque Variations for Double H-Ring Tests - Solid Rings

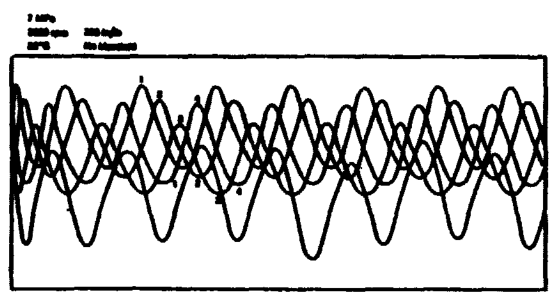

Fig. 3-58 Cycle-to-Cycle Pressure/Torque Variations for Double H-Ring Tests - Split Rings

The single H-ring (shown in Figure 3-59) will be tested after completion of the ongoing baseline tests which have a two-fold objective: 1) to establish the friction reduction for rolling-element versus journal bearings with back-to-back tests on the same test rig; and, 2) to provide a baseline for performance comparison of seal hardware planned for future tests.

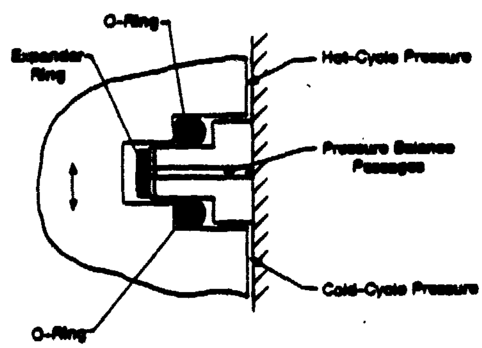

Fig. 3-59 Single H-Ring

The Mod I journal bearing baseline test with split/solid rings has been completed. Data comparing MTI and USAB Motoring Rig results for H_2 and N_2 are shown in Figures 3-60 and 3-61. Note that the friction power is higher for the MTI data at the high power points. This may be due to the differences in pressure ratio for the MTI and USAB tests (shown in Figure 3-62). Figure 3-63 shows the comparable motoring friction results for helium in the Motoring Rig.

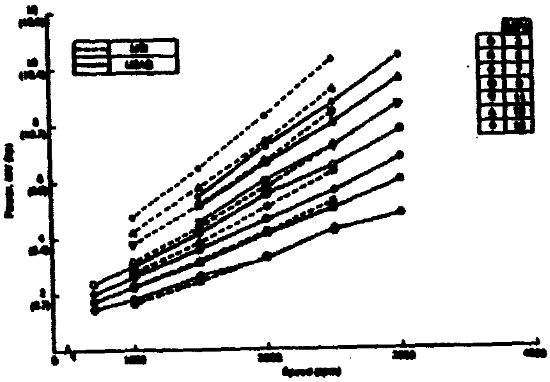

Fig. 3-60 Comparison of Baseline Journal Bearing Test (Mod I) and USAB #4 Data (H_2)

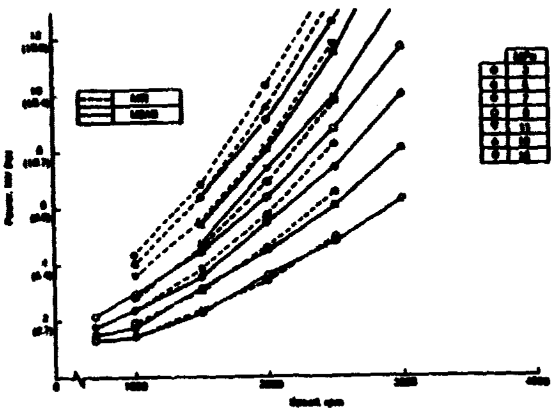

Fig. 3-61 Comparison of Baseline Journal Bearing Test (Mod I) and USAB #4 Data (N_2)

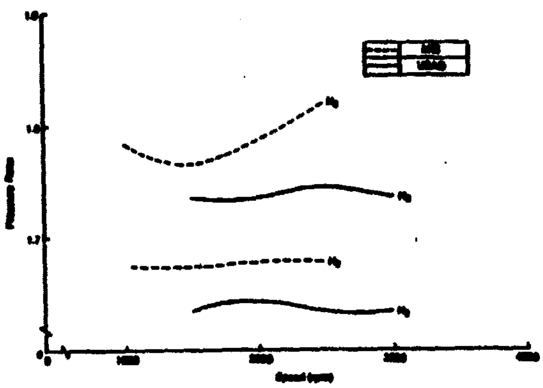

Fig. 3-62 Comparison of Baseline Journal Bearing - Split/Solid Ring and USAB #4 Data (15 MPa)

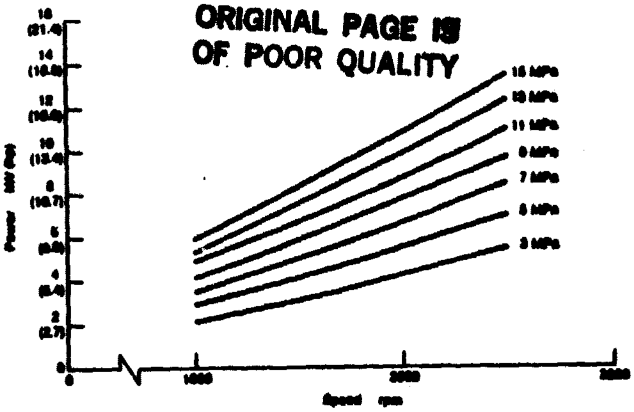

Fig. 3-63 Baseline Journal Bearing Test - Mod I He (June 25, 1983)

Control System/Auxiliaries Development

The major goals of this task include the development of engine control and auxiliaries systems. Specific control systems goals include the development of a simplified mean pressure control, a reliable microprocessor-based electronic control, and a highly flexible electronic Air/Fuel Control (AFC) with low pressure drop and low minimum fuel flow. A goal for the auxiliaries task includes the development of a positive-displacement combustion air blower. These hardware designs must be compatible with the extremes of an automotive operating environment.

Development during this report period focused on the combustion control system during transient operating conditions, the Digital Engine Control (DEC) software and reliability improvements, the conceptual design and analysis of a suitable combustion air blower, and the finalization of the Upgraded Mod I servo-oil system.

Activities and goals for control systems development for the latter half of 1983 will emphasize the utilization of the AFC System's flexibility, i.e., the ability to vary the air-to-fuel ratio as a function of engine parameters, and continuation of the reliability improvements in DEC software and hardware. Auxiliaries efforts will center on development of a positive-displacement combustion air blower, review of the engine check valve design and reliability, and the conceptual design of a hydrogen compressor to provide improved pump-down capability.

COMBUSTION CONTROL

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The objective of the combustion control task is to develop an AFC that provides more system flexibility and better control of the air-to-fuel ratio during transients than the current Bosch K-Jetronic system.

In January, the electronic AFC was tested with a pressure-atomized, dual-orifice fuel nozzle on Mod I engine No. 1. The test, done with the engine mounted in the vehicle, was limited to ignition and steady-state idle operation. Based on the results of that testing, the system was run on a CVS cycle. Results of that testing showed that the combustion control and fuel nozzle as a system resulted in neither good emissions nor good mileage. The performance was below that of the K-Jetronic system with an air-atomized nozzle. In May, the electronic AFC was tested with an air-atomized fuel nozzle on the CVS cycle. The results (Table 3-13) agreed very well with the emissions and fuel economy obtained with the Bosch K-Jetronic system.

Table 3-13

Electronic Air/Fuel Control Testing Comparison

CVS Urban Cycle	%EGR	HC g/mi	CO g/mi	NOx g/mi	Fuel Economy
9/82 Bosch	23	0.25	3.30	0.90	19.3 mpg
5/83 Bosch	>20	0.64	1.25	0.52	19.7 mpg
5/83 AFC	8.4	0.53	3.08	0.82	20.6 mpg

CVS Highway Cycle	%EGR	HC g/mi	CO g/mi	NOx g/mi	Fuel Economy
9/82 Bosch	23	0.004	0.31	0.66	32.1 mpg
5/83 Bosch	>20	0.007	0.10	0.34	35.0 mpg
5/83 AFC	12.5	0.024	0.75	0.45	33.9 mpg

The influence of EGR on both CO and NOx emissions has been demonstrated. The higher the EGR flow, the lower the CO and NOx. Equivalent EGR values were not obtained because of the increase in air pressure drop across the Bosch K-Jetronic when fuel pressure was present (only when running fuel through the K-Jetronic). The effect of EGR was demonstrated by running highway cycles with and without EGR:

CVS Highway Cycle	%EGR	HC g/mi	CO g/mi	NOx g/mi	Fuel Economy mpg
5/83 Bosch	High	0.009	0.13	0.29	36.5
5/83 Bosch	Off	0.007	0.74	0.94	34.3

When the difference in EGR is factored into the data, the electronic AFC compares favorably with the Bosch system.

To date, there has been no effort to make use of two potentially large advantages of the electronic AFC system: 1) the ability to vary the air-to-fuel ratio (λ) as a function of engine parameters other than just airflow (i.e., ignition versus running); and, 2) the ability initiate changes in airflow and fuel flow with appropriate delays to coordinate the changes in airflow and fuel flow rates to occur simultaneously at the fuel nozzle.

The electronic AFC has been integrated with the DEC installed in the MTI engine test cell. The system has been the only combustion control system for Upgraded Mod I engine No. 10.

DIGITAL ENGINE CONTROL

The objectives of this task are the development of a detailed understanding of the engine controls, modification of the software for the Upgraded Mod I, fabrication of control systems for engines in the program, and support of the engine controls in service.

A printed circuit board version of the DEC was installed in the MTI test cell, along with the required software changes, to run Upgraded Mod I engine No. 10. Two significant software changes were needed, both related to the thermocouple (T/C) configuration. Instead of four front-row heater tube T/C's and four back-row heater tube T/C's (as in the Mod I), the Upgraded Mod I has eight rear-row T/C's, thus necessitating change to the start sequence to turn on the engine starter motor at a lower temperature. The Mod I system cranked the engine based on a front-row heater tube temperature of 600°C. A change was made to crank the engine at a rear-row heater tube temperature of 475°C, which corresponds to ~600°C on the front row.

The temperature-control algorithm was changed to make the temperature control less sensitive to bad T/C's. Instead of controlling to the arithmetic average of the four rear-row T/C temperatures, the Upgraded Mod I is controlled to the median T/C temperature of the eight rear-row T/C's in the following manner: the temperature-control algorithm ranks the input values highest to lowest, discarding open T/C's. If both T/C's in a heater head quadrant are open, the engine is shut down, and the median of the remaining values is then selected as the control. If there are an even number of T/C's, the higher of the two middle values is selected.

For protection, two evaluations are made, i.e.,if a maximum allowable temperature is exceeded by any T/C, the engine is shut down. The algorithm also examines the difference between the highest and lowest indicated temperatures. If the difference exceeds a value indicating bad temperature distribution, the engine is shut down. The permissible difference between the hottest and coldest temperatures is reduced if the difference between the second highest and second lowest T/C exceeds a certain value. The intent is to quickly identify a large-area temperature maldistribution.

In addition, a modification to the EGR logic was implemented. The original control logic kept the EGR valve closed until the heater tubes were warm, and the fuel valve had been open for a specified time period, then the EGR valve was opened until the engine was shut down. That strategy was modified to additionally close the EGR valve at 2.5 grams per second for increasing fuel flow, and open the EGR valve again at 2.0 grams per second on decreasing fuel flow. This change (in the test cell control, but not in the vehicle control) was made to increase engine performance at the higher fuel flows. Following training of the test cell operators, the DEC was used in routine operation of the test cell.

Cold-start testing conducted on the TTB brought attention to the fact that the DEC's were fabricated with commercial-grade digital electronics components. The operating temperature range of those devices is typically 0°C to 70°C. A decision was made to extend the operating temperature range of the control by using extended temperature range components. Those components were specified and, where necessary, circuit redesigns were made to eliminate components that did not have a corresponding extended temperature range design. The only area where operation to -30°C is not expected is the video portion of the engine monitor, i.e., all functions of the engine control will operate normally, but the parameter display may not be available to the operator. Long delivery times associated with some of the components mean that the extended range system will not be assembled for testing until October, 1983.

Other changes in the DEC include a redesigned watchdog timer circuit, the addition of a T/C simulation switch for calibration (will disable the fuel valve when temperature is being simulated), integration of the electronic AFC into the DEC software, and definition of an improved power-control valve alignment procedure.

SERVO-OIL SYSTEM

The objective of the servo-oil system work was to provide a system capable of supplying adequate servo-oil pressure for all operating conditions. The inability of the servo-oil system to maintain pressure in the Mod I was traced to two areas: 1) a regulator that continually bypassed flow; and, 2) a pump that was too small. A larger pump was mated with the redesigned pressure-regulating valve, resulting in a system that could develop and maintain adequate servo-oil system pressure. The system was run on the vehicle for more than two hours at idle (a condition that had previously demonstrated loss of system pressure within fifteen minutes) without any loss in performance. The system was then observed during three weeks of CVS testing, and a two-day trip of more than 600 miles (again without any loss of performance) was made. The only remaining tests will be cold-temperature testing when weather permits.

Incorporation of a pressure-regulating valve that unloads the pump when adequate pressure is available provides an estimated 75% reduction in average power consumption for the servo-oil system (based on the observed cycle of the unloader valve). No further development of the system* is anticipated.

COMBUSTION AIR BLOWER

The objective of the combustion air blower task is to develop a variable, positive-displacement air blower that will result in an engine system that does not require airflow measurements for combustion control, variator for blower speed variation, or an air throttle valve for airflow adjustment. The blower should be designed for capability of operation with either EGR or CGR combustion systems.

Analysis of a vane design revealed potential problems with the inlet/outlet porting and with the stroking mechanism. The end porting of the design resulted in large efficiency and performance losses. Changing the blower stroke (displaced volume) by rotating the center line of the vane section in an arc relative to the housing resulted in poor performance because of the change in the suction and compression portions of the cycle relative to the location of the inlet and discharge ports, which occurs during rotation through the arc.

The constant-velocity coupling required between the vanes and housing proved to have a twice-per-revolution acceleration deceleration. At speeds above 1500 rpm, the resulting vibration was unacceptable

In the process of reviewing the design modification required, a design was formulated that eliminated the problems identified for the original concept. This "lobe" design was mechanically simpler and aerodynamically cleaner, so design efforts have been switched to that concept. Analysis has shown it to require less power than the centrifugal blower when the variator and air throttle valve are included with the centrifugal blower (Figure 3-64).

The major benefits to be realized from a combustion air system utilizing a variable, positive-displacement blower are reductions in the number of system components, control complexity, power consumption, and cost.

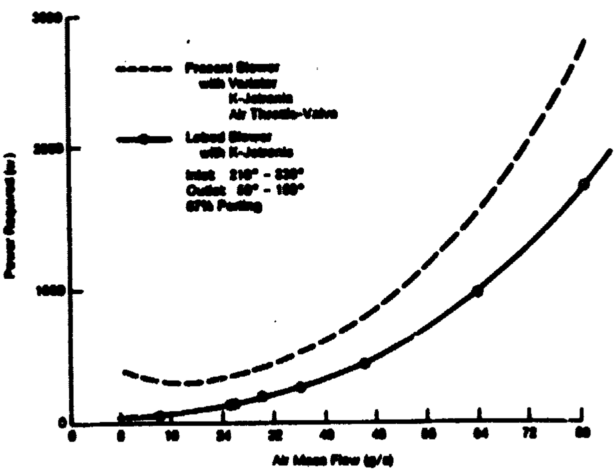

Fig. 3-64 Blower Power Versus Airflow - Preliminary Analysis

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*System drawings have been completed.

During the first half of 1983, four Mod I engines were under test: engines No. 2 and 3 at USAB, engine No. 1 in the Lerma TTB at MTI, and engine No. 10, which was removed from test at MTI during January and converted into an Upgraded Mod I engine configuration. A total of 2689 hours have been accumulated on the Mod I engines. Mod I engine testing during 1983 thus far has concentrated on development issues related to the Upgraded Mod I engine system.

Mod I Engine No. 1 (ASE 4-123-1)

Mod I engine No. 1 in the Lerma TTB has been extensively involved in transient Stirling-engine performance relating to emissions and fuel economy. The primary TTB effort during this report period was directed at improving emissions and mileage results established as a baseline in September, 1982 and reported in the last semiannual report*. Secondary efforts involved cold-start performance, cold-start penalty (CSP) determination, controls and fuel system development, and general hardware development.

In May the Lerma (Figure 4-1) established a new, improved performance baseline. Overall emissions levels for CO and NOx were reduced, while urban and highway fuel economy were improved. Table 4-1 and Figure 4-2 compare the May, 1983 emissions and mileage with the September, 1982 figures. Cold Urban Cycle mileage was up from September by 2.2%, and the Highway Fuel Economy test mileage was up 9.1%. These results increased the combined mileage by 4.4%. CO and NOx emissions levels were down significantly (nearly 50%), while HC levels (although still acceptably low) were up ~150% for the Urban Cycle. The basic reasons for the May improvements were changes in the TTB drive train, installation of a "high-stall" torque converter, and modification of the shift schedule for the transmission. Other minor contributions were a new hydraulic pump (unloads the system when the pressure is low), and shutdown of the combustion air blower during the hot-soak period between Phases 2 and 3 of the Urban Cycle.

Fig. 4-1 Mod I Lerma TTB

Table 4-1

Lerma TTB CVS* Cycle Emissions and Mileage**

	Urban/Highway	
	Sept., 1982	May, 1983
HC (g/mi)	0.254/0.004	0.640/0.007
CO (g/mi)	3.300/0.310	1.246/0.100
NOx (g/mi)	0.900/0.660	0.522/0.342
mpg	19.30/32.07	19.70/35.00
Combined mpg	23.50	24.50

*EGR Schedule - Fixed-Size Orifice; 23% EGR at 1.0 g/s fuel flow; EGR off during start-up

**3850-lbs. inertia weight, 11.1-hp road load, indolene fuel, EGR combustion system

*MTI Report No. 83ASE308SA3

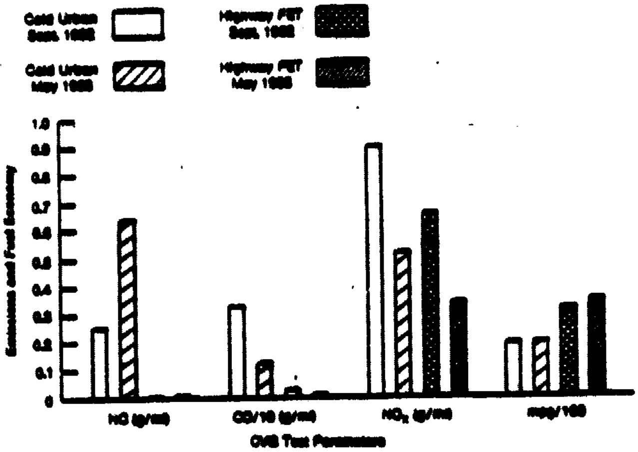

Fig. 4-2 Lerma TTB CVS Cycle Test Results Comparison - Emissions Levels and Mileage

One of the prime concerns of Stirling-engine fuel economy performance is the amount of fuel needed to raise the EHS and HES to operating temperature. To measure this parameter on the Lerma, special CVS Urban Cycle tests were conducted that involved running the TTB through a CVS Urban Cycle without starts, and comparing the results with a normal CVS Urban Cycle with a cold-start before Phase 1 and a hot start before Phase 3. The difference between the Phase 1 no-start and cold-start fuel consumption was 308.70 grams; the difference between the Phase 3 no-start and hot-start was 188.70 grams. The CSP calculated by summing 43% of the cold-start fuel consumption difference and 57% of the hot-start fuel consumption was 240.30 grams, which equates to a Mod I engine penalty of 7.0 mpg for the Urban Cycle. Special tests with distilled water and standard antifreeze used separately in the cooling system were also conducted. No difference in engine fuel economy was noted from one coolant to the other.

The Lerma was also used by MTI's Component Development Engineers to evaluate a nonair-atomized fuel nozzle and an electronic AFC System. Initial testing of the two in combination was disappointing. The nozzle, a Delavan dual-orifice, could not operate in a transient manner when combined with the electronic AFC - it was subjected to carbon buildup at low fuel flows. Later testing conducted with the BOM air-atomized nozzle and electronic AFC produced results equivalent to the current standard Bosch K-Jetronic AFC. Results of these tests are reported in more detail in Section III of this report.

Initial cold-start testing on the Lerma at low ambient temperatures (~30°F) was plagued with hardware problems. The servo-oil system portion of the Power-Control System was unstable due to cold hydraulic (automatic transmission) fluid. The USAB DEC System malfunctioned continually because "nonwinterized" components were used. These problems were addressed by investigating alternate hydraulic fluids, and establishing the use of winterized components in future electronic controls. Although unstable for the above-mentioned reasons, the engine did start continually at the lower ambient temperatures.

Future Lerma testing will involve testing similar to that described above. The fuel economy and emissions levels will be improved through controls modifications and hardware changes. The TTB will also be used to develop hardware for the Upgraded Mod I engine.

Mod I Engine No. 2 (ASE 4-123-2)

During this report period, engine No. 2's (located at USAB) primary function was performance development. From January until March, 1983, the engine was in a Mod I BSE configuration, and was used for Mod I/Upgraded Mod I performance development. In April the engine was converted into an Upgraded BSE configuration. The development items evaluated from January to March include:

- Mobil I lubricating oil instead of 20W30;
- reduced-capacity oil pump (11-mm versus 18-mm width);
- heater head and combustor temperature distribution; and,
- upgraded EHS development and performance.

The first testing was done to determine if the use of Mobil I lubricating oil offered any measurable performance gains over the standard 20W30 lubricating oil - no measurable performance gains were found with the Mobil I oil. The second test involved reducing the capacity of the lubricating oil pump by 39% and checking performance. The current standard 18-mm wide "gerotor"-type pump was replaced with an 11-mm wide pump - again,

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no measurable performance gains were found.

The next test was conducted with a heater head and combustor that were temperature-painted with two varieties of paint - the head was cleaned and painted with 0A128*; the combustor with 0A128* on its heat-exposed side and TP6* on its cool side. All painting and reading of the paint was done by Rolls Royce in England. The hardware was run at a mean pressure level of 7 MPa and a rear-tube temperature of 720°C for 10 minutes. Results of this testing for temperature ranges that are currently undergoing more analysis are tabulated below:

Heater Head	
Rear-Tube Row	610- 740°C
	740- 840°C
Front-Tube Row	740- 840°C
Cylinder/Regenerator Manifolds	610- 740°C
Cylinder/Regenerator Housings (Primary)	610- 740°C
Combustor (CGR)	
Inner Surface	780- 940°C
	940-1030°C
Outer Surface (exterior)	840- 910°C
	890- 910°C
	910°C +

The last testing conducted on engine No. 2 during this report period, prior to its conversion to a complete Upgraded Mod I BSE, was to evaluate the Upgraded Mod I EHS. Testing involved running at λ values of 1.25 and 1.15, while EGR was being varied from 0-60% (including 40 and 50%). This testing was started by USAB, but was cut short by the need for conversion of the engine into an upgraded configuration. Results of the testing are reported under the Performance Analysis and Combustion Development subsections.

The Upgraded Mod I EHS tests revealed a problem with the preheater, i.e., distribution of air-side airflow was not uniform. As a result, high air-side exit differential temperatures exist. This problem is currently being worked on by both MTI and USAB.

*paint temperature range designators

Mod I Engine No. 3 (ASE 4-123-3)

Engine No. 3, located at USAB, is being used as an endurance engine to accumulate hours on the Mod I SES. The endurance program calls for the following sequence of tests:

- 100 starts and stops - 25 with the engine at room temperature, and 75 with the engine warm;
- 1000 hours of endurance based on the test cycle in Figure 4-3;
- 100 starts and stops - same as above; and,
- 1000 hours of endurance - same as above.

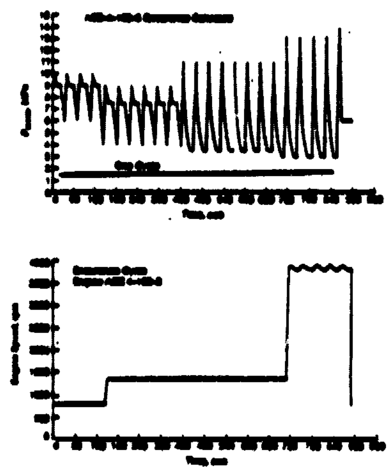

Fig. 4-3 Mod I Engine No. 3 Endurance Cycle - Mean Pressure and Speed Vs. Time

The test cycle based on data from the Metro/Highway Driving Cycle, was created to subject the SES to the accelerated conditions an engine would experience driving 55% of the Urban (metro) CVS cycle and 45% of Highway Fuel Economy Test. The engine will experience fatigue- and creep-related loading identical (as much as possible) to an actual engine during a vehicle driving cycle. Each cycle lasts 830 seconds and, in 2000 hours time, will be repeated a total of 8675 times. Engine teardowns and inspections are scheduled after each 100 starts and after each 500 hours of cyclic running. The engine's performance will be checked at the max-efficiency point every 100 hours. If the engine is not within 5% of its original performance level, it will be torn down for inspection. A record will be kept of all parts' failures and changes.

During the first half of this year, engine No. 3 had completed the first 100 starts and accumulated 400 hours of cyclic endurance. Numerous problems were encountered primarily with the engine seals and auxiliaries. Figure 4-4 presents the incident record for the 100-starts test. A total of 11 failures occurred. The main areas involved were the main seals and piston rings in the CES, and the Power-Control System portion of the auxiliaries.

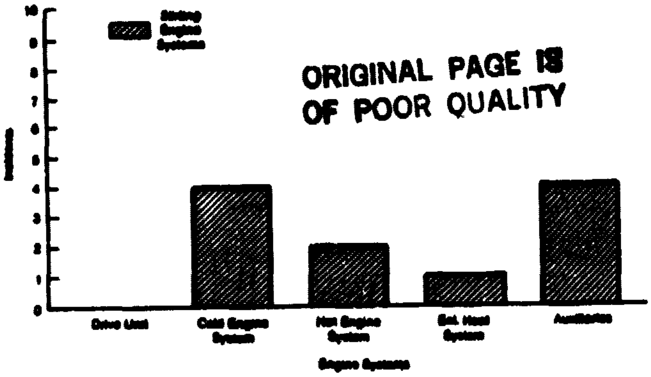

Fig. 4-4 Mod I Engine No. 3 Incidents Record First 100 Starts and Stops Total Test

Figure 4-5 presents the incident record for the first 110 hours of cyclic endurance running. A total of 21 failures occurred during the time period, with the major portion (15) occurring in the CES and auxiliaries. The main problem areas were the seals in the CES and the blower in the auxiliaries.

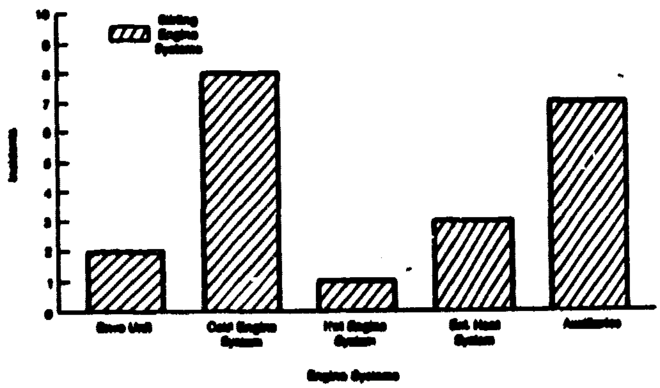

Fig. 4-5 Mod I Engine No. 3 Incidents - Cyclic Endurance Test (110 Hrs, 477 Cycles)

Figure 4-6 is a stacked bar chart that contains the incident record cumulatively and by test. The main seal and ring problems (present for some time) are the subject of special Component Development programs at MTI. The seals and rings used in engine No. 3 are the original Mod I main seals and split/solid piston ring designs. It should be noted that all but one of the piston ring replacements were caused by the main seal failures, and are considered secondary in nature.

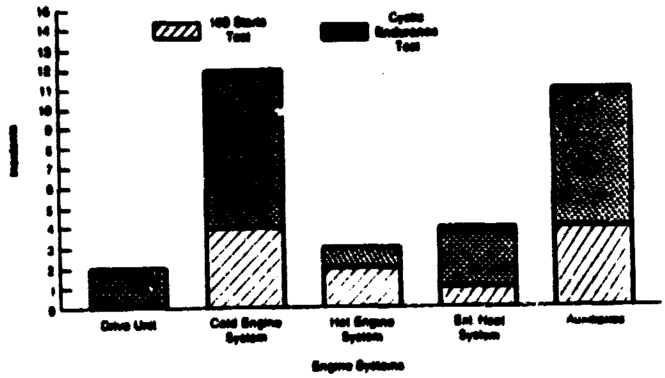

Fig. 4-6 Mod I Engine No. 3 Incidents - Cumulative Failures (100 Starts, 477 Cycles, 110 Hrs)

In the auxiliary area, five of the eleven failures occurred in the blower "free wheel" area. The blower has been redesigned in the drive area and, as soon as the new design is available, will be installed on the engine. Two other failures worth mentioning in more detail involved a heater head quadrant tube and two fuel nozzles. The tube failure occurred along a welded seam of Multimet tubing. This tube material has been replaced on newer heads by seamless Inconel 625 and, in the future, by CG-27 tubing. Both materials represent a cost savings and better creep-rupture resistance. The tubing used on the original Mod I heads (Multimet) was also of poor quality. The two fuel nozzle failures occurred because of clogging. The fuel nozzle is the subject of special studies and development.

Engine No. 3 will continue on its cyclic endurance test until a total of 1000 hours is reached, at which time an extensive teardown will be conducted.

Mod I Engine No. 10 (ASE 4-123-10)

Early in the year, Mod I engine No. 10 was removed from test and rebuilt into an Upgraded Mod I engine (ASE 4-123A-10). Reporting on the engine is covered in Section V of this report.

Design and Performance Summary

The design of the Upgraded Mod I engine was completed during this report period. The objective of the Upgraded Mod I was to provide a proof-of-concept design for advanced ASE technology, incorporating as much proposed RESD technology as possible into the Mod I configuration. A secondary goal of the design was to provide improved cost, weight, reliability, and durability relative to the Mod I engine. The resulting Upgraded Mod I design retains the basic U-4 canister arrangement of the Mod I engine, but incorporates major changes to the drive unit (incorporation of rolling-element bearings), HES (revised heater head designed for part-power optimization), EHS (reduced size preheater), and controls and auxiliaries. The performance projections for the Upgraded Mod I engine are compared to the Mod I in Table 5-1 for operating temperatures of 720°C and 820°C. As noted, improvements in power and efficiency are projected throughout the operating regime.

Table 5-1

Upgraded Mod I Performance Projections

Operating Point	Quantity		Mod I	Upgraded Mod I	
			720°C	720°C	820°C
15 MPa 4000 rpm	Power	kW	55.9	58.1	69.0
		hp	75.0	77.9	92.5
	η	%	28.1	29.7	33.0
Maximum Effic.	Power	kW	26.2	33.7	35.9
		hp	35.1	45.1	48.1
	η	%	35.3	37.5	40.1
5 MPa 2000 rpm	Power	kW	10.4	11.9	13.4
		hp	13.9	15.8	18.0
	η	%	26.9	31.6	33.7
14 kW* 1500 rpm	Pressure	MPa	7.8	7.4	6.6
	η	%	31.5	34.3	35.4
12 kW 2000 rpm	Pressure	MPa	5.6	5.1	4.6
	η	%	28.2	31.7	32.7

*Average Operating Point

A description of the design changes incorporated in the Upgraded Mod I are presented in the following paragraphs.

DESIGN DESCRIPTION

External Heat System (EHS)

EHS design emphasis was placed on size/weight reduction, which was accomplished by reducing the number and thickness of preheater plates, and utilizing a more efficient insulation. The preheater/inlet/exhaust cover assembly and installation has been redesigned into a two-piece unit - the cover is one piece; the inlet/exhaust/preheater matrix form a second piece. Two clamps hold the preheater to the engine and the cover to the preheater. The EHS assembly is shown in Figure 5-1, which shows both the EGR (left side) and CGR (right side) combustors. The Upgraded Mod I combustor is an EGR-type. Efforts are underway to develop a CGR system for eventual incorporation into the Upgraded Mod I.

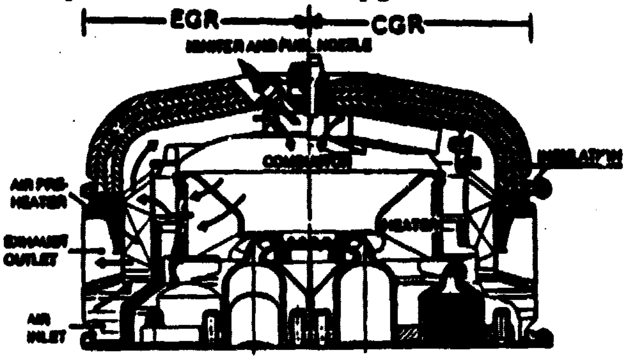

Fig. 5-1 Upgraded Mod I External Heat System

Hot Engine System (HES)

Major changes were made in the heater portion of the engine during this report period. Operating history on the Mod I and improved stress analysis capability were used to completely redesign the cylinder and regenerator castings. The redesign includes thinner walls, modified manifolding, changes to the manifold-to-casting entry geometry, and the use of an iron-based, nonstrategic material (XF-818) for the castings. A weight savings of 4.5 kg was achieved, with a corresponding reduction in cold-start penalty. The system was also optimized for improved part-power performance, which resulted in a smaller regenerator and cooler. The regenerator shape and manifold-to-casting entry was also changed to improve flow distribution. The Upgraded Mod I features a dome-shaped regenerator with a single entry manifold. The Upgraded Mod I HES assembly is shown in Figure 5-2.

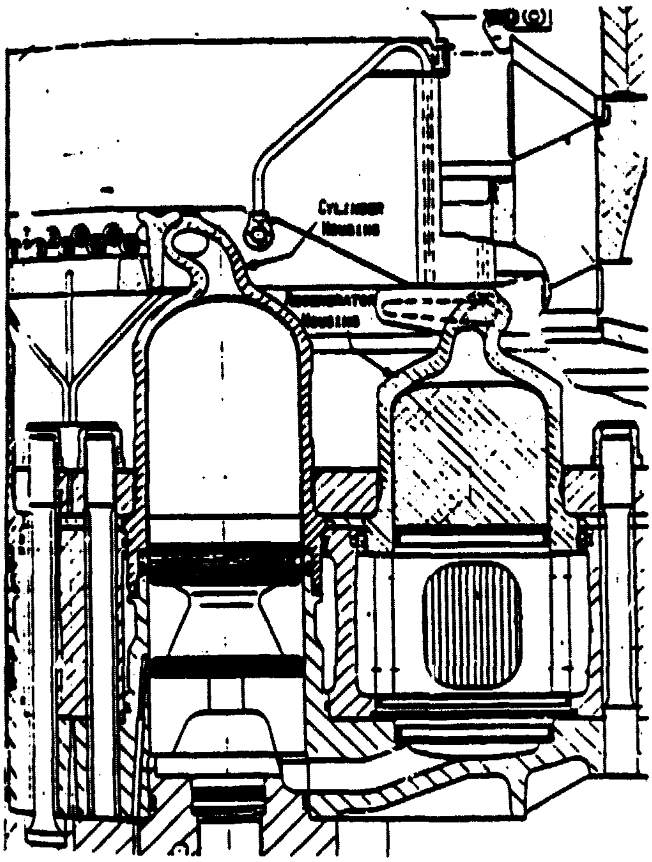

Fig. 5-2 Upgraded Mod I Hot Engine System

Cold Engine System (CES)

Emphasis in the CES was placed on simplification. The cylinder liner and cold connecting duct were integrated into a single unit, providing better engine alignment, as well as decreasing the number of parts.

Drive Unit - The drive unit was redesigned to incorporate rolling-element bearings, resulting in an estimated improvement of two points in maximum engine efficiency. To provide interchangeability with the Mod I, no changes in cylinder or bearing spacing were made. The drive units utilize Mod I crankcase and bedplate assemblies modified only for the rolling-element bearings. A lower capacity oil pump was also used, made permissable by the reduced oil flow required by the rolling-element bearings. The drive unit is shown in Figure 5-3. A reduction in friction power requirement of ~1.5 kW (~2 hp) at maximum power is estimated for the Upgraded Mod I drive unit relative to the Mod I.

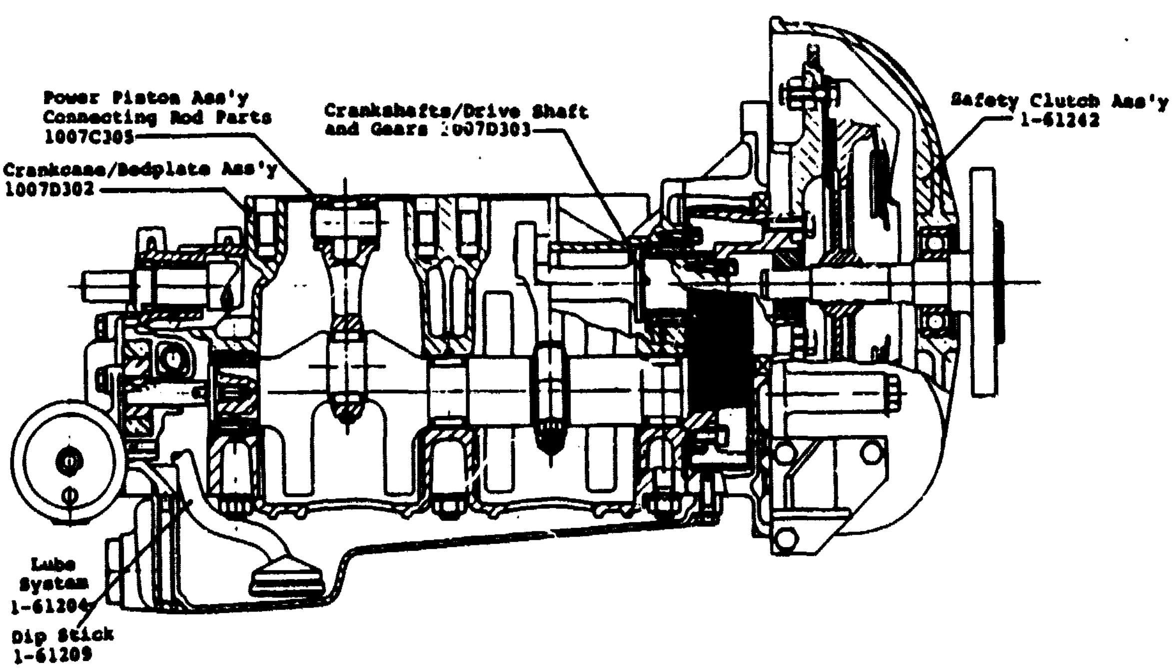

Fig. 5-3 Upgraded Mod I Lightweight Reduced Friction Drive

Auxiliaries and Controls

Emphasis in the auxiliaries and control systems was placed on weight reduction to permit easier assembly on the engine, and address the problems encountered during the Mod I program. The changes are discussed below:

Blower - The input shaft bearing and freewheel arrangement were revised to correct problems encountered in Mod I operation.

Servo-Oil Pump/Atomizer Air Compressor - These units were an integrated auxiliary in the Mod I engine; however, separate units were designed for the Upgraded Mod I to eliminate the servo-oil system overheating problems encountered in the Mod I program. The drive belt wrap angle was also increased to clear up a problem experienced with the Mod I system.

Bracketry - The auxiliaries, accessory, blower motor, and alternator brackets were redesigned to either reduce weight or because redesign was required due to necessary changes to the Mod I mounting locations. In addition, the auxiliary bracket was redesigned so that all auxiliaries can be mounted and aligned on the bracket prior to installation on the engine.

Power-Control System - The power-control blocks were redesigned to reduce weight (a weight savings of 7 kg is projected with this system). The hydrogen compressor body was also redesigned for lighter weight. In the working gas system, an improved filter was incorporated in the seal house vent line to enhance oil removal. A filter was also added to the servo-oil system, upstream of the Moog valve actuator, to prevent contamination and eventual malfunction of the valve. In the Air/Fuel Control System, the K-Jetronic unit of the Mod I has been replaced with a Digital Electronic Combustion Control System that senses airflow and, using a predetermined λ, calculates and meters the fuel flow. The system is easily set up for any schedule, and can be integrated with the engine power control.

Engine Performance Predictions

Performance projections were developed for both 720°C and 820°C heater head operating temperatures. The component maps used to develop those projections are presented in Figures 5-4 to 5-12, and the performance maps for the full SES are shown in Figures 5-13 and 5-14 for 720°C and 820°C, respectively.

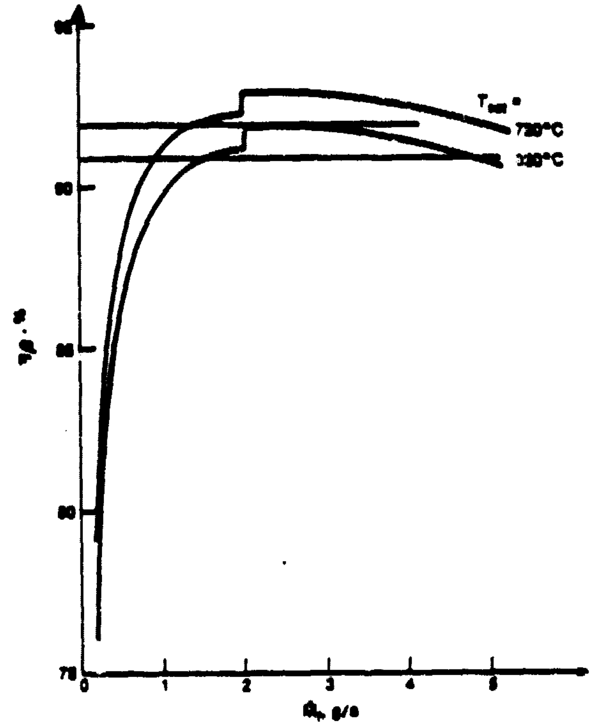

Fig. 5-4 Upgraded Mod I EHS Efficiency (T_{set} = 720°C and 820°C)

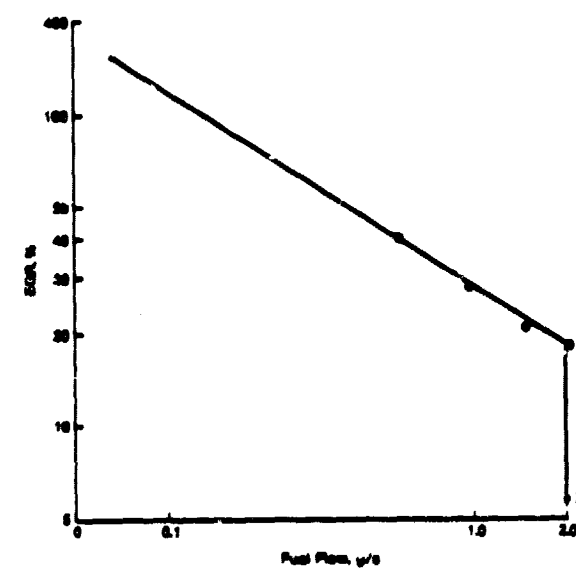

Fig. 5-5 Upgraded Mod I EGR Schedule

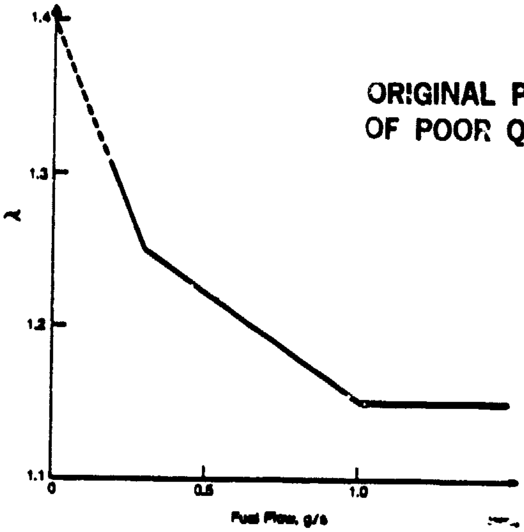

Fig. 5-6 Upgraded Mod I Excess Air (λ) Schedule

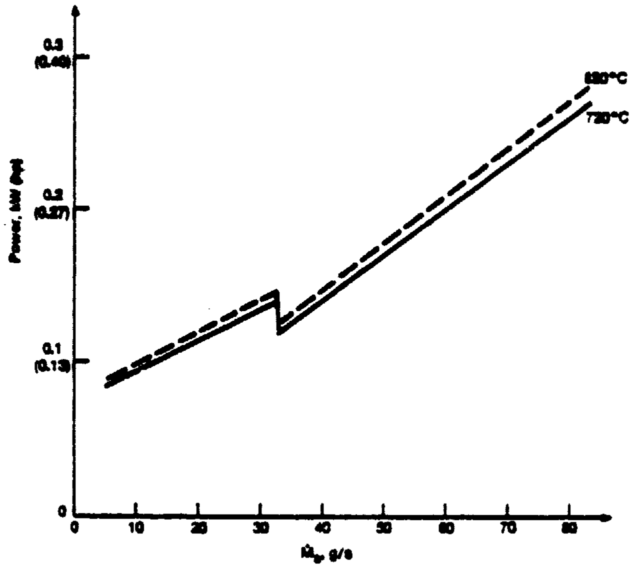

Fig. 5-9 Upgraded Mod I Servo-Oil Pump Power

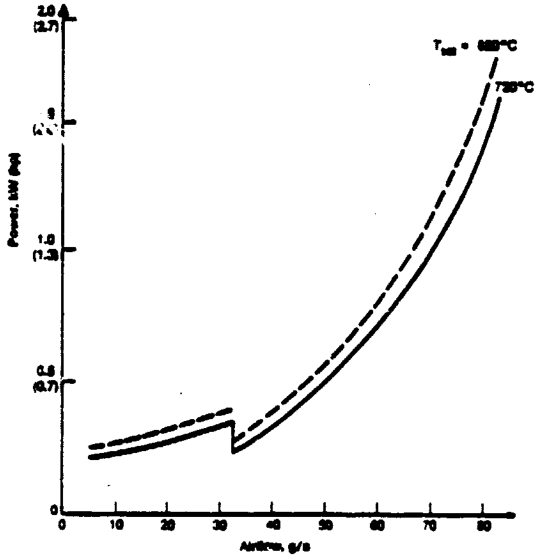

Fig. 5-7 Upgraded Mod I Combustion Air Blower Power

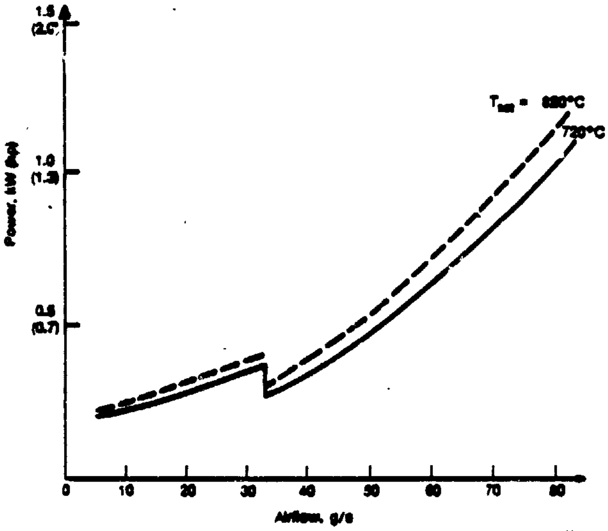

Fig. 5-8 Upgraded Mod I Atomizer Air Compressor Power

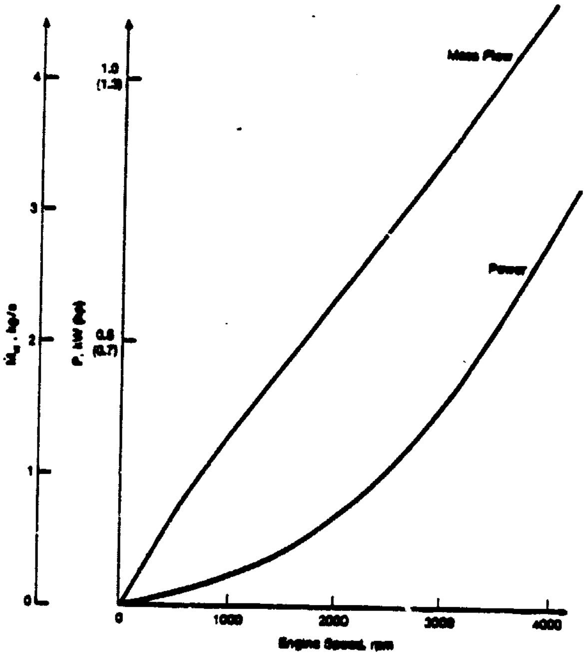

Fig. 5-10 Upgraded Mod I Water Pump Power and Mass Flow

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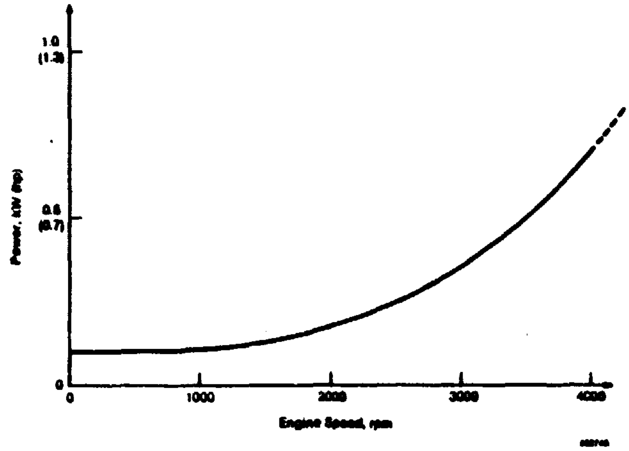

Fig. 5-11 Upgraded Mod I
H_2 Compressor Power

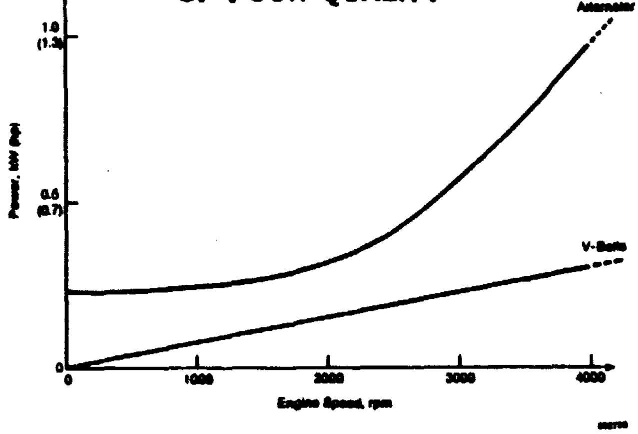

Fig. 5-12 Upgraded Mod I Alternator
and V-Belt Power

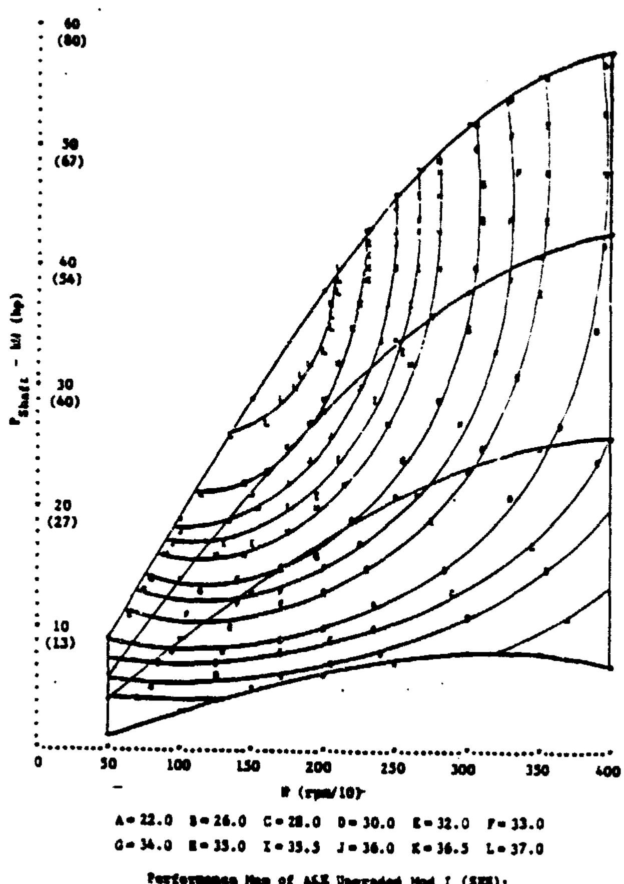

Fig. 5-13 Upgraded Mod I Engine
Performance (T_{set} = 720°C)

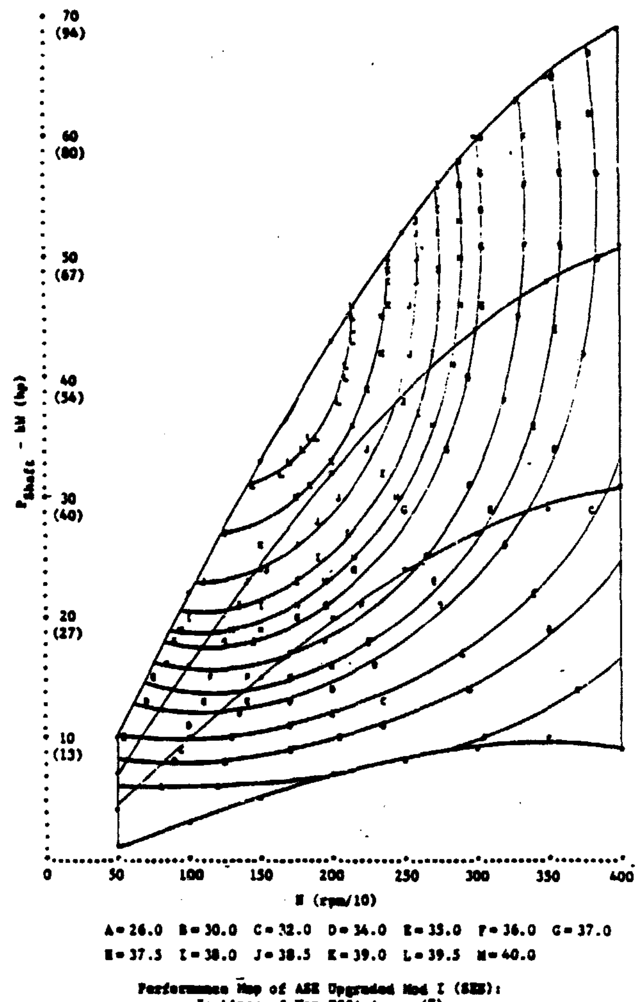

Fig. 5-14 Upgraded Mod I Engine
Performance (T_{set} = 820°C)

As noted previously, power and efficiency improvements are anticipated throughout the operating range. Preliminary vehicle fuel economy estimates (Table 5-2) have been prepared based on the engine map at 820°C. The mileage decrease with cold-start penalty was estimated based on the ratio of stored heat between the Upgraded Mod I and Mod I engines. This projected performance can be expressed in terms of improvement relative to the Mod I engine by analytically adjusting both Mod I and Upgraded Mod I vehicle performance to the same weight and power/weight ratios. If both are adjusted to a test weight of 3400 lbs., and a power/weight ratio of .0192 hp/lb., the fuel economy with unleaded fuel would be:

Engine	Combined mpg	% Improvement
Mod I	27.8	Base
Upgraded Mod I	32.7	+17.5

Significant improvement is therefore anticipated with the Upgraded Mod I engine.

Table 5-2

Preliminary Upgraded Mod I Vehicle Fuel Economy Estimates*

Urban w/o Cold-Start Penalty	30.1
Urban with Cold-Start Penalty	26.6
Highway	41.9
Combined with Cold-Start Penalty	31.8

Initial Engine Test Results and Analysis

Initial testing of an Upgraded Mod I engine began in April with Upgraded Mod I engine No. 10, and continued throughout the remainder of this report period with additional data generated on Upgraded Mod I engine No. 2 incorporating only the Upgraded Mod I EHS and HES. The first full-power testing of Upgraded engine No. 10 produced poor power and efficiency levels**. To understand the reasons for the poor performance level, further analysis was made. Figure 5-15 is a comparison of measured EHS efficiency to predictions. The lower efficiency of ~1-1/2% does not account for very much of the measured overall engine efficiency (~1/2 percentage points versus 8 percentage points measured). Figure 5-16 shows the measured power deficiency as a function of both speed and charge pressure. (Note that the difference is indeed a function of both speed and charge pressure.) The possible causes for this difference were analyzed to be:

- ring leakage;
- friction;
- pumping;
- regenerator ineffectiveness; and/or,
- cycle temperature.

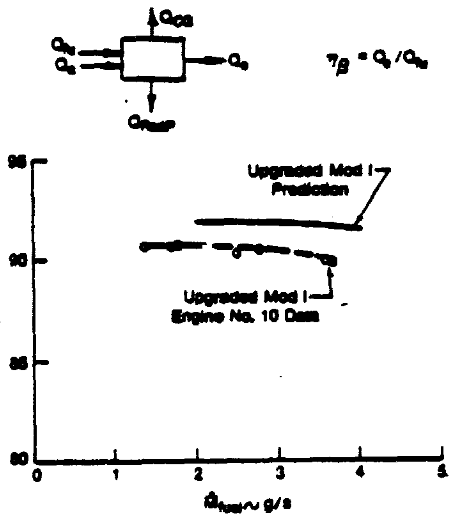

Fig. 5-15 Measured Upgraded Mod I EHS Efficiency

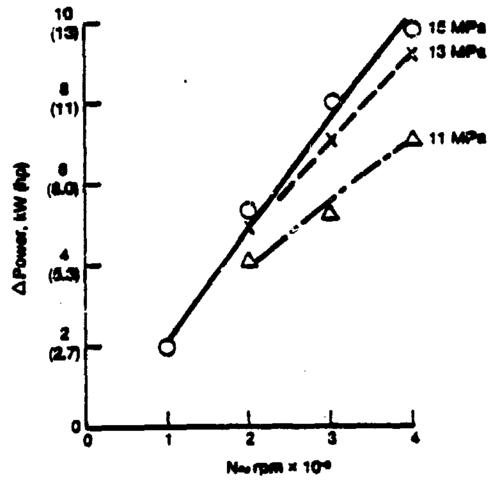

Fig. 5-16 Upgraded Mod I Power Deficiency Compared to Predictions

*unleaded fuel
**see Figures 5-23 and 5-24

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As a singular cause, pumping and leakage can be eliminated due to the different shaped characteristics associated with these effects (Figure 5-17). Cycle temperature, friction, and regenerator ineffectiveness have, in general, the same shape as shown by the engine data; however, a large increase in drive unit friction would be evidenced by a high lubricating oil heat-rejection rate. The oil temperature rise across the engine was very low, eliminating friction as the cause of the problem.

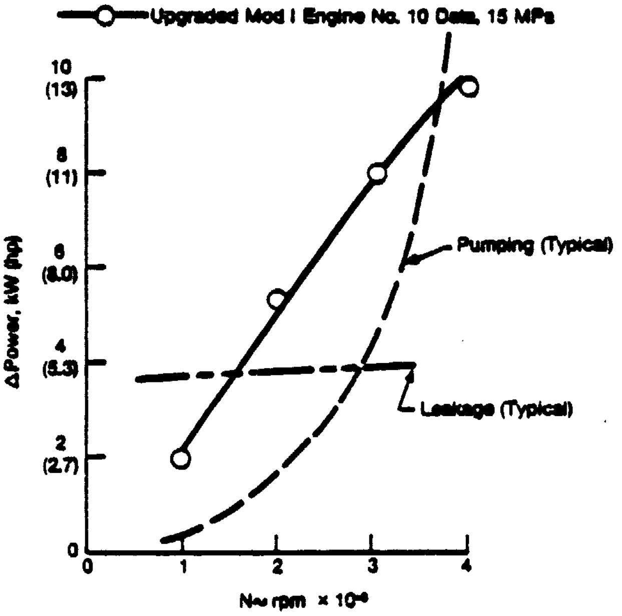


Fig. 5-17 Upgraded Mod I Power Loss Comparison

In reviewing the design, there are several items that may contribute to producing a decrease in regenerator effectiveness; namely: 1) the regenerator fill factor is reduced substantially from the Mod I engine (32.5% versus 43%); 2) the regenerator flow distribution is a single-center entry, as opposed to the split manifold/separate tube entries of the Mod I; and, 3) regenerator fit in the housing is different for the Upgraded Mod I in several areas, specifically in the contact (line-on-line) with the housing at the top, in the stackup above the cooler retainer flange, and in the side clearance fit at operating temperature.

In theory, the side clearance provides the seal to prevent regenerator bypass flow. The difference in material properties for the Upgraded Mod I housing results in an increased side clearance (.199 versus .096 diameter) relative to the Mod I; however, the regenerator sealing tests showed that the external bypass was not a problem, and that regenerator ineffectiveness was due to porosity, poor manufacturing techniques, or poor flow distribution within the regenerators. Tests are planned to address each of the above items during the latter half of 1983.

Figure 5-18 is a comparison of cooling water heat rejection for a typical Mod I engine and Upgraded Mod I engine No. 10. The heat rejection of Upgraded Mod I engine No. 10 is higher than that of the Mod I engine, confirming the measured power deficiency levels.

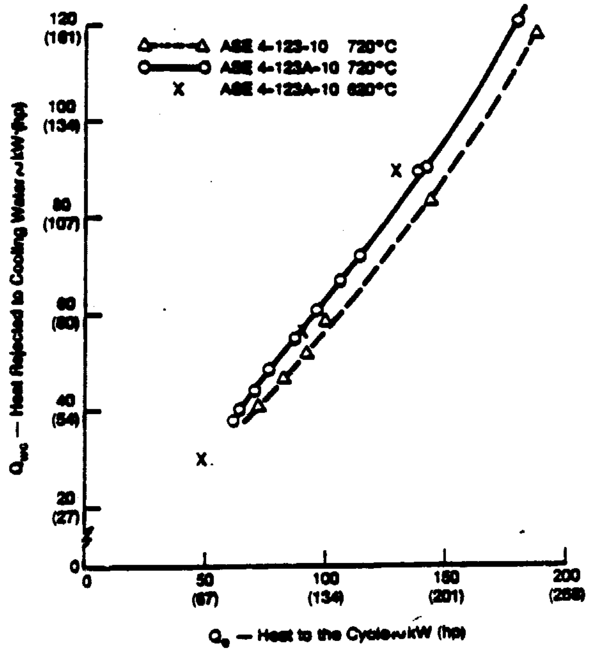


Fig. 5-18 Upgraded Mod I Cooling Water Heat Rejection Comparison

Upgraded Mod I engine No. 2 ran baseline testing in the BSE configuration during this report period. Engine power and efficiency levels compared to predictions are shown in Figure 5-19, where No. 2's performance levels have been analytically adjusted to SES levels. Included on the curve is data from Upgraded Mod I engine No. 10. Note that the performance levels of Upgraded Mod I engine No. 2 are considerably better than that of Upgraded Mod I engine No. 10. This measured difference is confirmed by the difference in cooling water heat rejection (Figure 5-20).

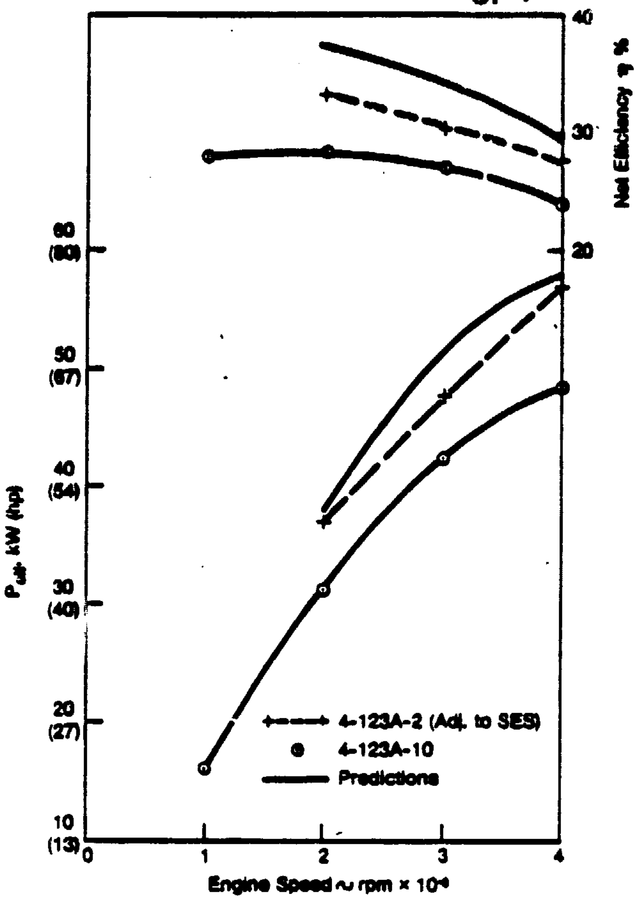


Fig. 5-19 Upgraded Mod I Performance Data Analytically Adjusted to SES Configuration

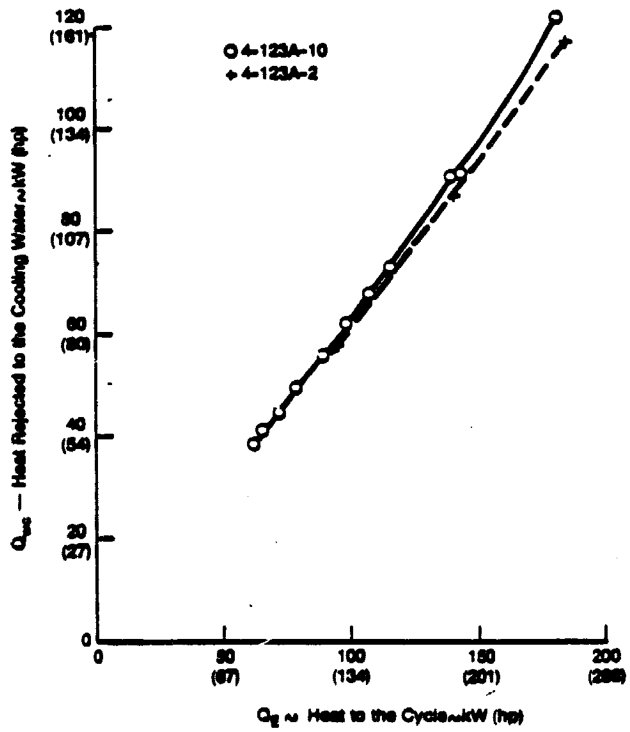


Fig. 5-20 Upgraded Mod I Engines No. 2 and 10 Cooling Water Heat Rejection

Testing during this report period on Upgraded Mod I engine No. 10 with improved sealing on the heater head combustion-gas side, and with repaired Pmean snubbers improved performance to a level close to that of Upgraded Mod I engine No. 2 (shown in Figure 5-21). It can be expected that the remaining performance difference is due to regenerator and appendix gap loss differences. This will be further defined by running both heater heads on Upgraded Mod I engine No. 10 during the latter half of 1983.

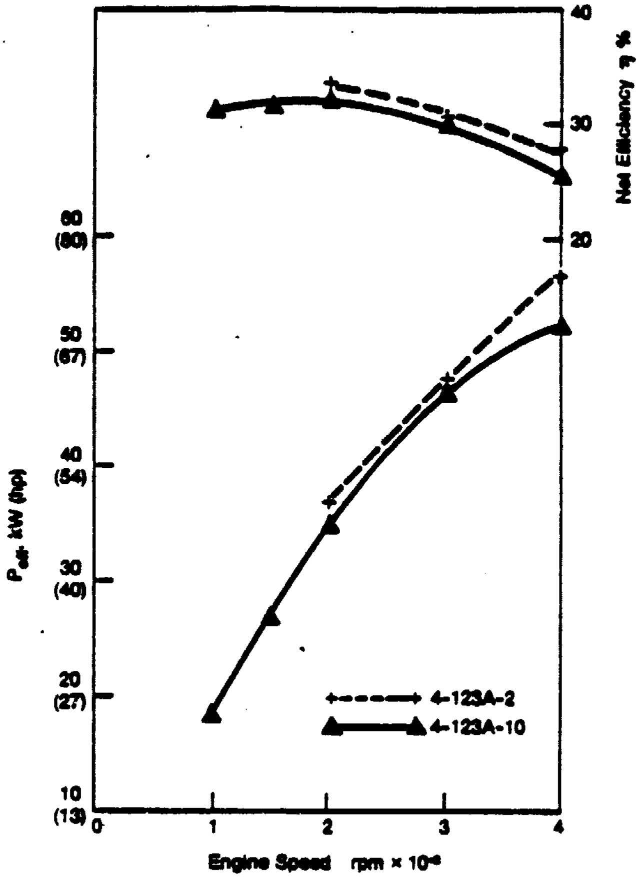


Fig. 5-21 Upgraded Mod I Performance Data Analytically Adjusted to SES Configuration

Highlights

Two ASE Program engines are presently operating in an Upgraded Mod I engine configuration. Engine No. 2 (at USAB) was converted to an upgraded configuration in April, and is in a BSE configuration. To date, Upgraded Mod I engine No. 2 has accumulated 56 hours of operating time.

Engine No. 10 (at MTI) was converted to an upgraded configuration in January and February, and ran in an SES configuration on February 28th for the first time. To date, Upgraded Mod I engine No. 10 has accumulated 110 hours of operating time.

Upgraded Mod I Engine No. 10 (ASE 4-123A-10)

Following a short piston ring breakin of ~20 hours at 600°C rear-tube temperatures, a performance check of the engine (at 720°C rear-tube row) revealed it to be below performance predictions for this operating temperature. Figure 5-22 is a plot of Upgraded Mod I engine No. 10's measured and predicted power levels, and Figure 5-23 is a plot of measured and predicted engine efficiency.

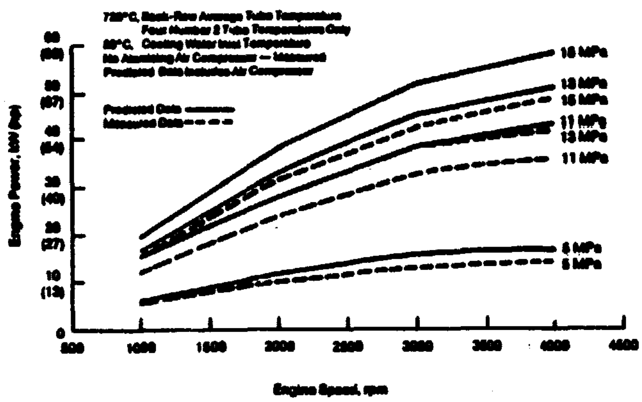


Fig. 5-22 ASE 4-123A-10 Power Vs. Speed

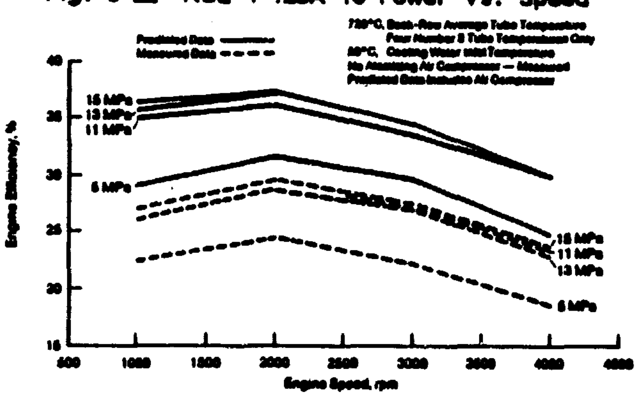


Fig. 5-23 ASE 4-123A-10 Efficiency Vs. Speed

The engine's power level was down 9.4 kW (12.6 hp) at the maximum power point, and efficiency was down ~8.5% at its maximum efficiency point. Analysis of the deficiency suggested that performance was down primarily due to the cycle proper rather than the EHS. Engine performance improved with later running (results are plotted in Figures 5-24 and 25).

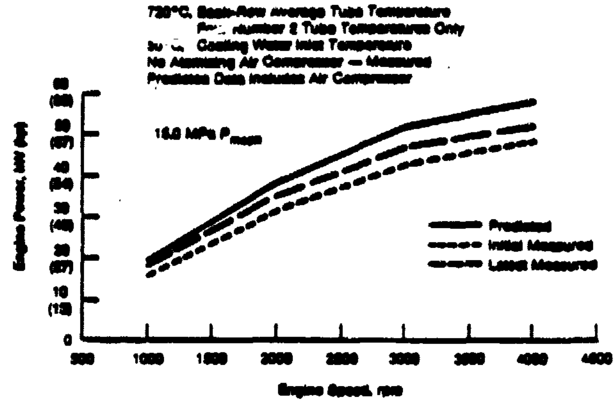


Fig. 5-24 ASE 4-123A-10 Power Vs. Speed - 15.0 MPa P_{mean}

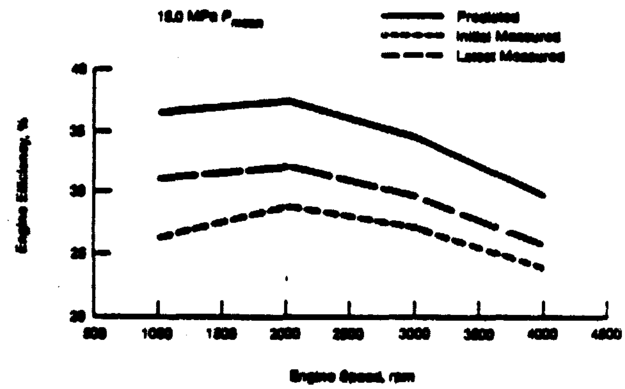


Fig. 5-25 ASE 4-123A-10 Efficiency Vs. Speed - 15.0 MPa P_{mean}

The problem was discovered to be the P_{mean} instrumentation taps. Orifices ("snubbers") are installed in the seal housings of each cylinder. These orifices are small-sized holes that are exposed to the sinusoidal pressure wave of the cycles on one side, and lines and pressure measurement transducers on the other side. The orifices dampen out the cycle pressure wave, and should produce a mean pressure on the transducer side. The orifices were found to be missing o-rings and were loose in their fittings, resulting in an equivalent orifice much too large to dampen out the cycle pressure wave. This created a larger cold-space dead volume in the engine, and recorded mean pressure levels in the engine that were too high. Resultant engine power and efficiency levels were up from the original performance run following repair of the orifices. Engine power increased from 48.6 kW (65.2 hp) to 52.2 kW (70 hp) at the maximum power point, and maximum engine efficiency increased from 28.8% to 32%. The engine is still low in performance compared to the predicted values of power and efficiency (58 kW (78 hp) and 37.4%, respectively).

Several areas have been identified as possible sources for the performance deficiency. Some of these areas have been examined in more detail by conducting specialized short test programs. The two key areas under investigation are regenerator performance and gas flow distribution into the cylinder housing from the manifold.

Analysis of engine test data has pointed towards the regenerator as a possible cause of the engine's poor performance. The regenerator does not appear to be absorbing as much heat as anticipated. Possible explanations for this are poor flow distribution into the regenerator's hot side, bypass leakage around the regenerator, and poor regenerator manufacturing techniques. To date, the bypass leakage flow around the regenerator has been examined because of evidence that the regenerators were loose and rotating in their housings. Viton seals have been placed between the gas coolers and regenerators to block leakage and prevent rotation. Tests have shown no improvement in performance.

Regenerators that were ruined because of rotation have been cross-sectioned. Evidence of poor manufacturing technique to produce the regenerators was apparent. To create the domed shape, a cylinder of compressed and sintered screens was compressed more on the outside diameter instead of the progressively smaller diameter screens being used.

This method of manufacture creates a non-uniform-porosity screen volume, and could affect internal flow patterns. Poor brazing of the screen element to the canister was also found. This could result in internal leakage around the screen proper. New regenerators are being manufactured with the dome shape formed by EDM of a sintered screen cylinder. The use of EDM will eliminate the O.D. compression and improve the braze surface to the canister.

Testing on the engine has also resulted in the melting of Hytrel backup rings on the top of the cylinder liners. The melting of the rings is oriented in an area 180-270° around the liner away from the engine's vertical centerline. A heater head quadrant cylinder housing was instrumented (shown in Figure 5-26). The data obtained (plotted in Figures 5-27, 28, 29) agrees with the location of the melted area on the backup rings. A gradient of ~225°C exists around the housing, with the section of housing running towards the vertical centerline of the engine running the coolest. The melted area of the backup rings matches the hot areas of level 3 plotted in Figure 5-29.

The gradient confirms original speculation on the cause of the melted rings, i.e., as the working gas leaves the neck area of the cylinder housing and enters the cylinder proper, it is directed towards the outboard wall section; causing overheating of the wall area of both the cylinder housing and liner underneath as the gas flows between them and the piston dome. The effect of this flow on appendix and shuttle losses is unknown; however, it is expected to cause an efficiency reduction. Action to correct this problem is being taken at both USAB and MTI. Existing hardware will be modified in the neck area of the cylinder housing to direct the flow towards the center of the dome by welding.

Figure 5-30 is a cross section of the CES/HES mating area. The items mentioned previously are identified in the cross section.

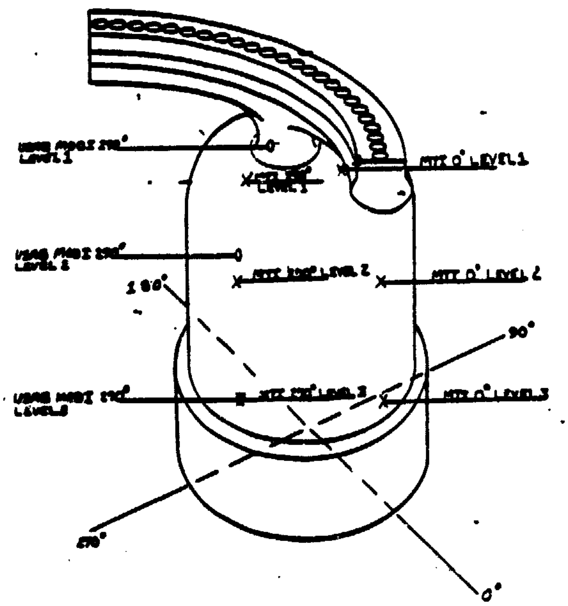
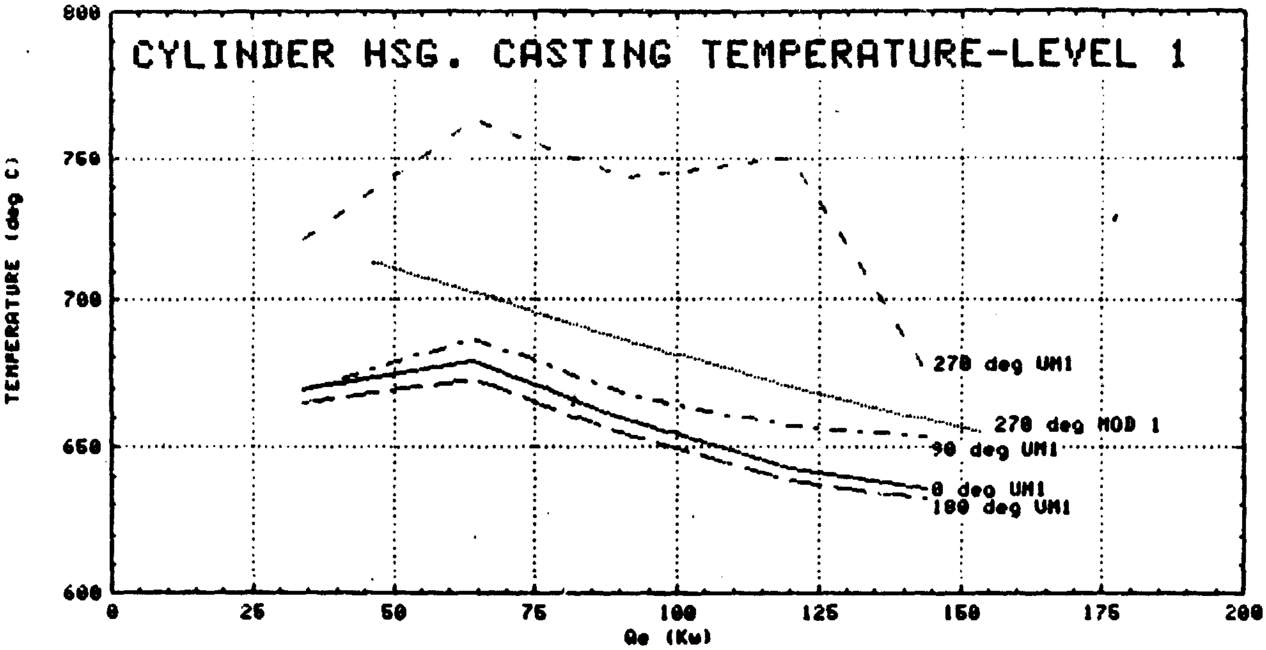
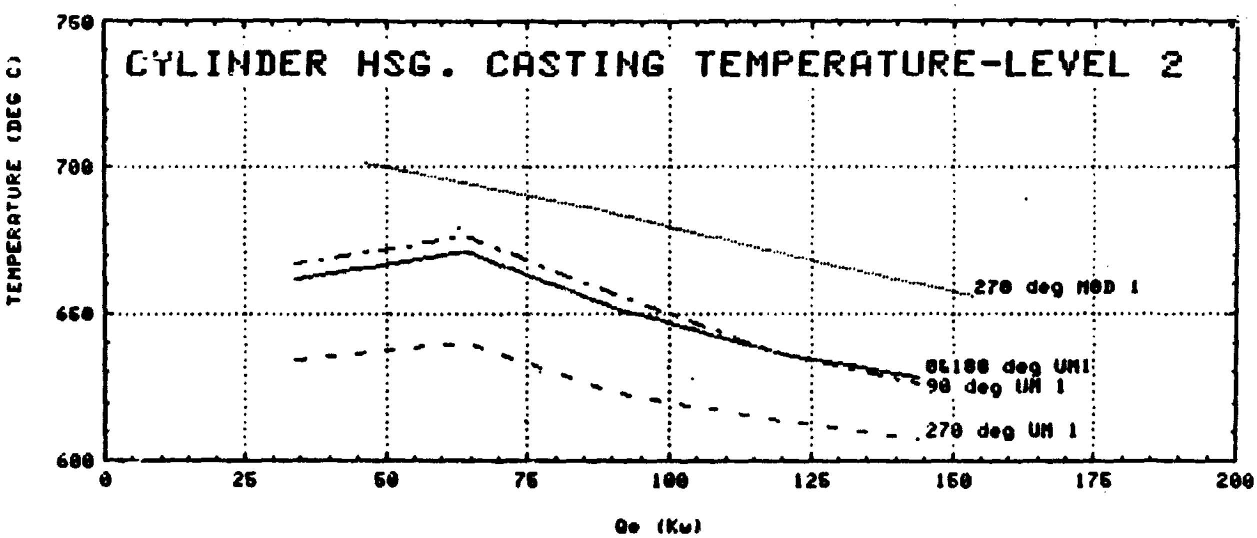


Fig. 5-26 Upgraded Mod I Cylinder Casting Temperature Measurement Locations



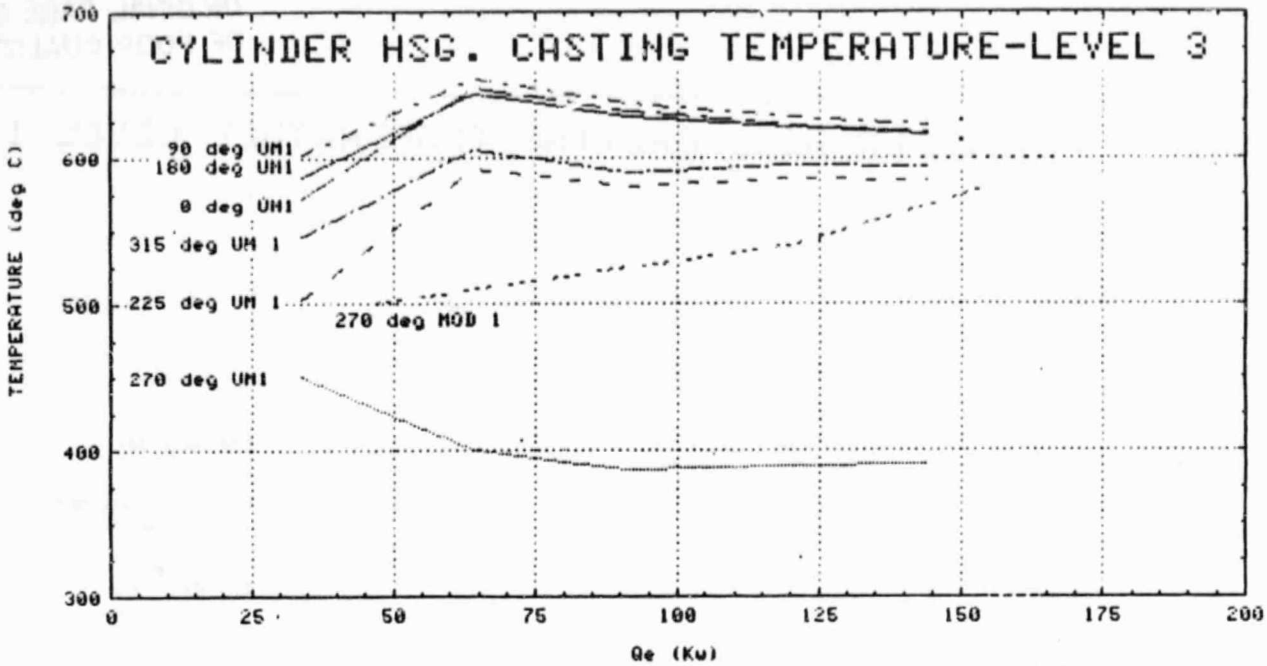
HTI ENGINE Qe	LEVEL 1 0 Deg	LEVEL 1 90 Deg	LEVEL 1 180 Deg	LEVEL 1 270 Deg	USAB ENGINE Qe	USAB LEVEL 1 270 Deg
33.89	670	669	665	721	46.28	713
27.13	679	687	673	763	84.4	690
91.17	659	668	654	743	121.12	670
119.90	643	657	639	750	153.74	655
144.10	636	653	632	675		

Fig. 5-27 Upgraded Mod I Cylinder Housing Casting Temperature - Level 1



HTI ENGINE Qe	LEVEL 2 0 Deg	LEVEL 2 90 Deg	LEVEL 2 180 Deg	LEVEL 2 270 Deg	USAB ENGINE Qe	USAB LEVEL 2 270 Deg
33.89	662	667	662	634	46.28	701
64.13	671	676	671	640	84.4	687
91.17	651	656	651	623	121.12	670
119.90	636	636	636	614	153.74	656
144.10	628	626	628	607		

Fig. 5-28 Upgraded Mod I Cylinder Housing Casting Temperature - Level 2



MTI ENGINE Qe	LEVEL 3 0 Deg	LEVEL 3 90 Deg	LEVEL 3 180 Deg	LEVEL 3 225 Deg	LEVEL 3 270 Deg	LEVEL 3 315 Deg	USAB ENGINE Qe	USAB LEVEL 3 270 Deg
33.89	587	602	572	503	450	546	46.28	500
64.13	644	654	648	592	400	605	84.4	521
91.17	627	637	631	580	387	589	121.12	541
119.90	621	627	623	585	389	595	153.74	579

Fig. 5-29 Upgraded Mod I Cylinder Housing Casting Temperature - Level 3

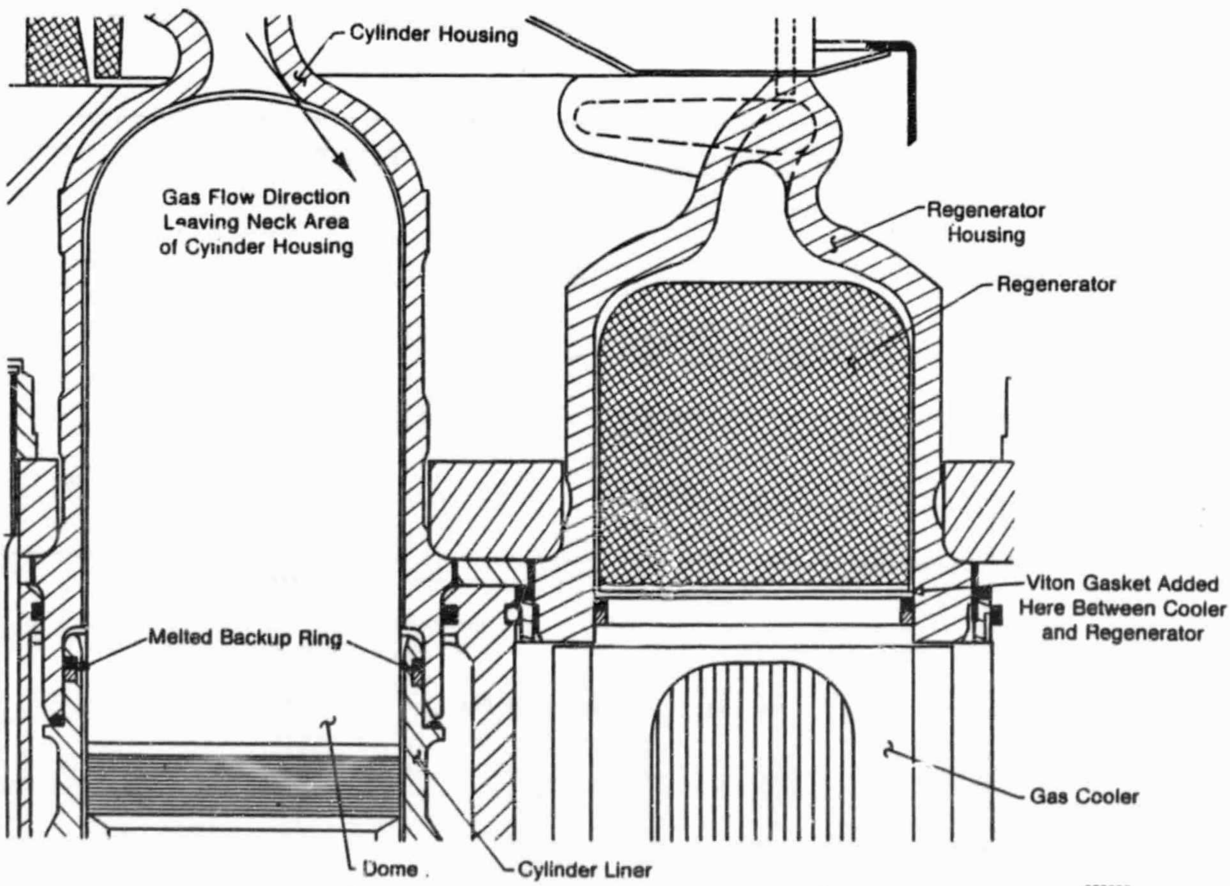


Fig. 5-30 Upgraded Mod I Cold/Hot Engine System

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Other problem areas encountered on Upgraded Mod I engine No. 10 include high combustion-gas temperatures after the second row of heat exchanger tubes, and an air-side preheater distribution problem. The high combustion-gas temperatures were brought down by sealing the combustion-gas leakage areas in the heater head quadrants. The gaps between quadrant-to-quadrant, second-row heat exchanger fins, and between the bottom fins and the regenerator housing were sealed with shim stock. This lowered the combustion-gas temperature entering the preheater by 100°C. The preheater air-side flow distribution is currently being examined. Current preheated air gradients are in the 100-150°C range at high flow rates.

Upgraded Mod I Engine No. 2 (ASE 4-123A-2)

The USAB engine, No. 2, has been in an Upgraded Mod I configuration since April, 1983. Test results on this engine have been similar to those obtained on engine No. 10 during its latest run. The problem areas are basically the same, i.e., Upgraded Mod I engine No. 2 is also low in performance, and has been used to explore these performance problems. Since the engine has no auxiliaries, it has slightly better power and efficiency levels than Upgraded Mod I engine No. 10.

VI. INDUSTRIAL TEST AND EVALUATION PROGRAM (ITEP)

The objectives of the Industrial Test and Evaluation Program (ITEP) are to:

- enhance the ultimate success of the ASE Program through the testing of Upgraded Mod I engines by automotive/engine-manufacturing companies outside the program;

- provide an independent evaluation of the technology level of the ASE Program, expand the transfer of Stirling-engine technology beyond the program, and provide an opportunity for automotive/engine-manufacturing inputs for improving the design and manufacturability of Stirling engines; and,

- provide a larger engine test base for assessing engine performance and durability.

During this program MTI will:

1) procure, manufacture, and fabricate sufficient hardware for delivery of two Upgraded Mod I engines to industry, and support of these engines with spares;

2) provide a 3,000-lb. AMC Spirit with one of the above engines installed and completely checked out;

3) establish baseline parameters for the Spirit prior to its delivery to the industry teams;

4) assemble and fully test two Upgraded Mod I engines (one installed in the Spirit) for delivery to the industry teams;

5) provide training for the industry teams' engineers and technicians for engine/system operational characteristics, engine assembly and disassembly, engine installation and testing, diagnostic and troubleshooting; and,

6) provide test site support at the industry locations as required for up to twelve months after delivery of the engines.

NASA will conduct negotiations directly with U.S. automotive/engine-manufacturing firms to determine contract extent, test program to be conducted, and the deliverables. To date, NASA has reached signed agreements with two industry companies - John Deere and Cummins Engine. A third signed agreement with General Motors, who will receive the Spirit for testing and evaluation, is anticipated in early July.

Activity under this task started on March 1, 1983. Initial task emphasis was devoted to the processing of purchase orders for hardware and component procurement for the two complete engines and their associated parts.

Figure 6-1 is a schedule reflecting the time frames for the various work elements associated with ITEP. Procurement activities are progressing along in a timely fashion in accordance with the schedule. Preparation of the Spirit for one of the ITEP engines is in progress; it will be ready to accept the engine in accordance with the schedule. Currently, meetings with the industry team members are being planned to determine their specific training requirements.

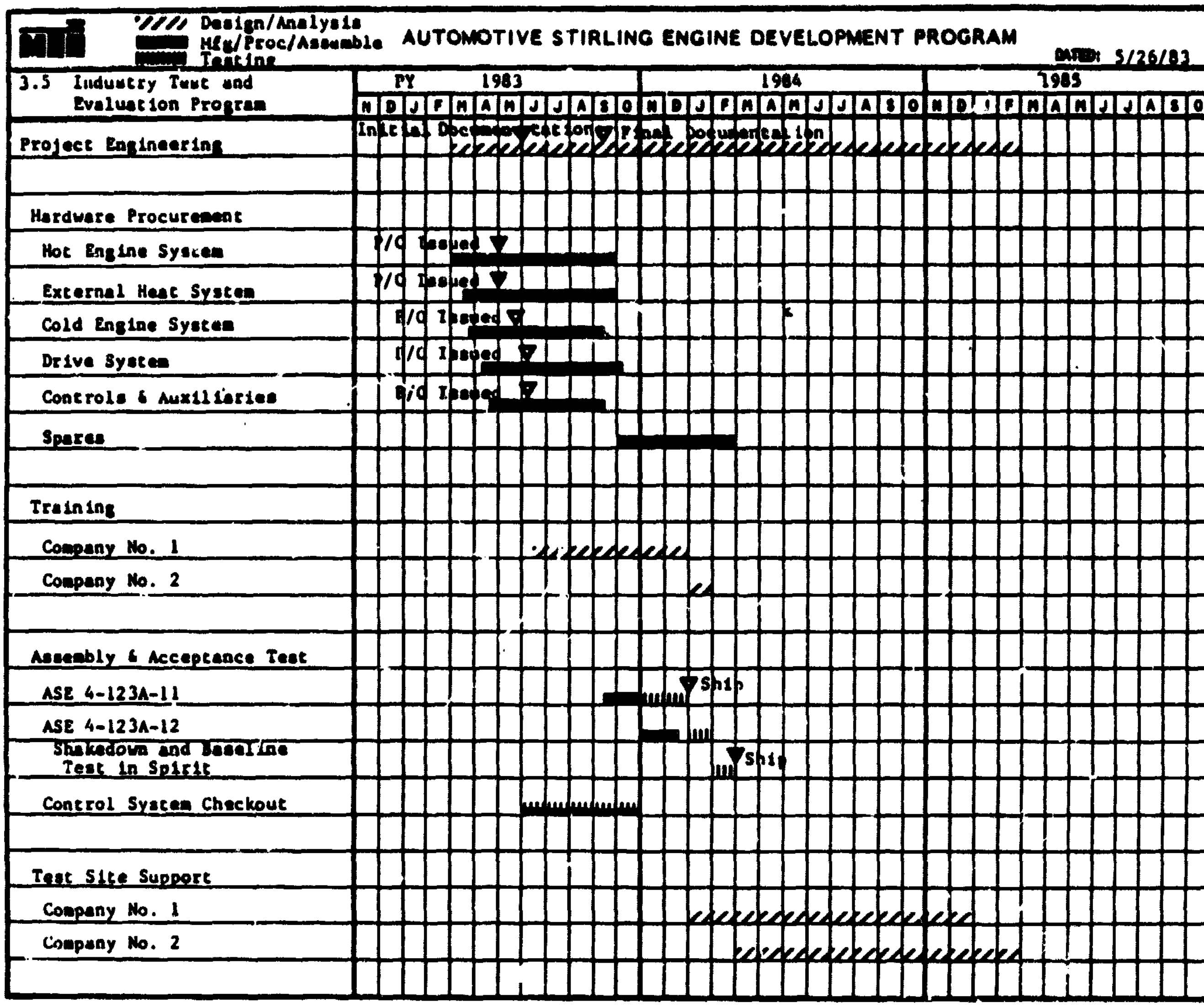

Fig. 6-1 Industry Test and Evaluation Program Schedule

Quality Assurance Overview

Below is the status of the Quality Assurance Reports (QAR's) as of the end of this report period:

Open QAR's	38
Closed QAR's (Total to Date)	487
P-40 QAR's	129
Mod I QAR's	366
Upgraded Mod I QAR's	30
Total QAR's in System	525

MOD I QAR EXPERIENCE

A summary of problems documented via the QAR system is presented in Figures 7-1 through 7-4. Problems are defined as items that: 1) cause an engine to stop running; 2) prevent an engine from being started; and, 3) cause degradation in engine performance. Problems that fall into these categories must be minimized to provide acceptable engine performance and mean time between failures. Major problems identified for individual units or assemblies are:

- Moog valve;
- heater head;
- check valves;
- combustion air blower;
- fuel nozzle
- igniter
- preheater
- atomizing air compressor and servo-oil pumps;
- combustor; and,
- flame shield.

Table 7-1 is a summary of operating times versus failures for all ASE Program engines.

Table 7-1

Summary of Operating Times Versus Failures for all ASE Program Engines

Engine	Operation Time	Mean Operating Time to Failure
P-40-4	10,084.0	142
4-123-1	822.0	52.7
4-123-2*	660.0	100.6
4-123-3*	803.0	160.6
4-123A-10*	110.3	-
4-123A-2*	56.0	-

*All classes of failures since initial start of the engine are included in the calculation of mean time to failure.

Remaining engines' operation times are based on time and failures occurring after acceptance test.

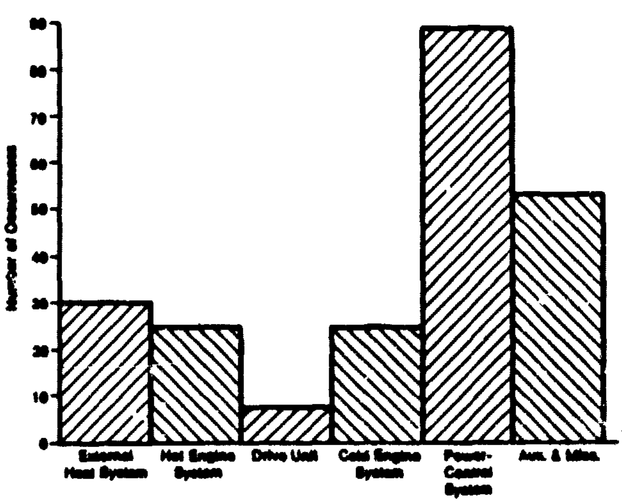

Fig. 7-1 Mod I Major Systems' Failures and Discrepancies

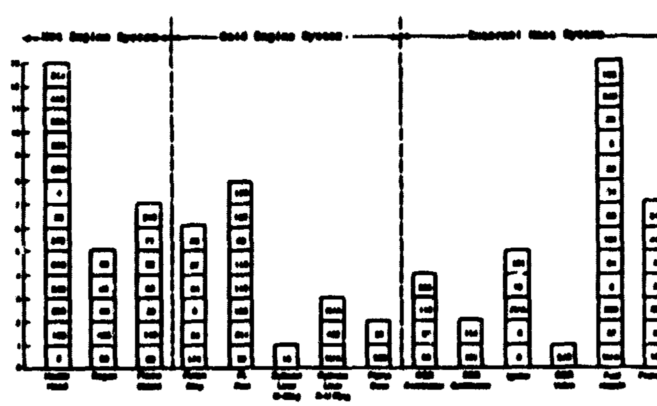

Fig. 7-2 Hot Engine, Cold Engine, External Heat Systems' Failures and Discrepancies

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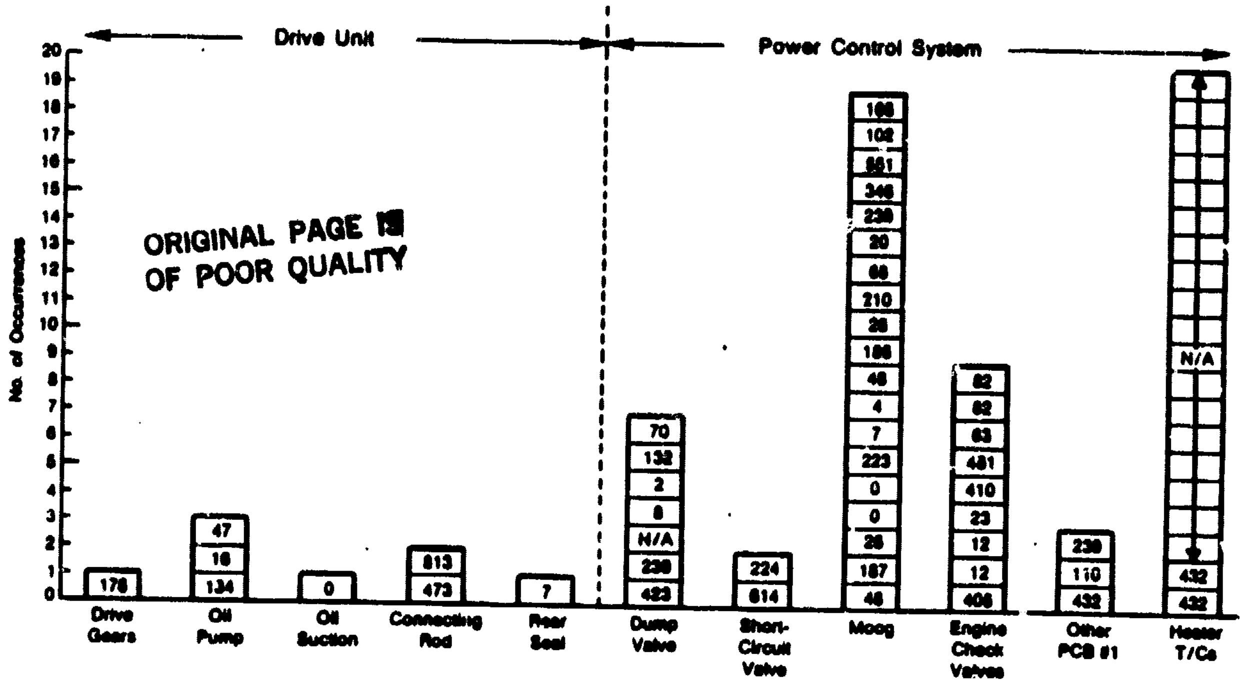

Fig. 7-3 Drive Unit and Power-Control System Failures and Discrepancies

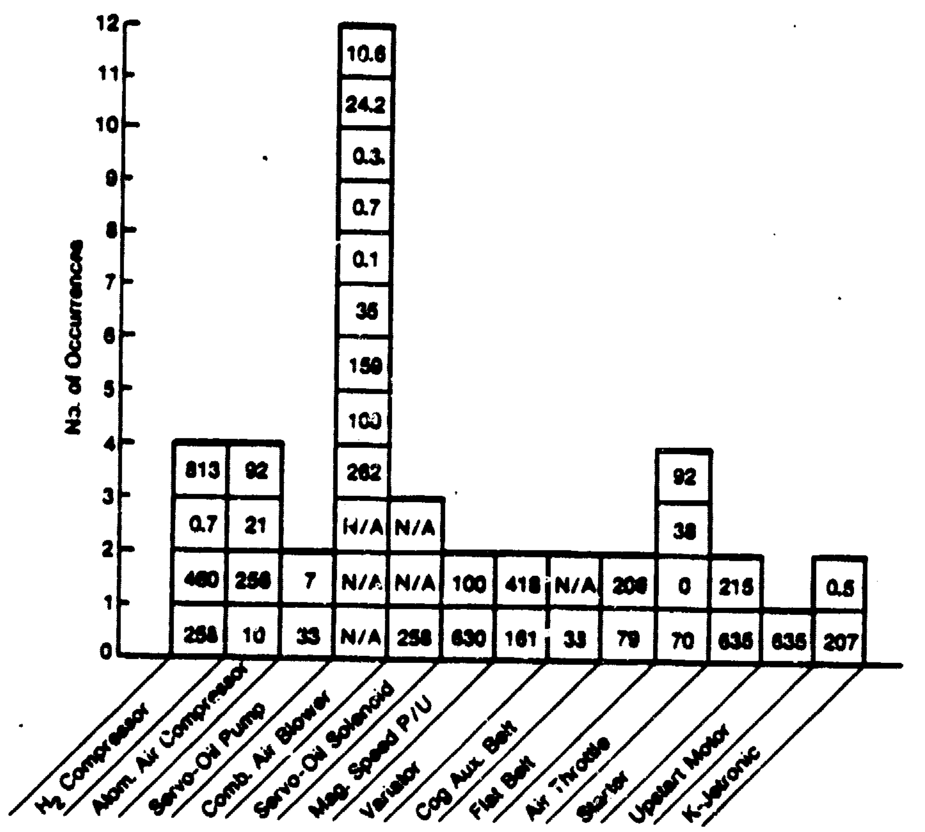

Fig. 7-4 Auxiliaries and Miscellaneous Items' Failures and Discrepancies

VIII. FACILITIES

Below is a brief summary of the facilities activities performed during the first half of 1983:

- Minor maintenance was performed on all ASE Program test cells.
- All test cells are currently operational.
- A preliminary design study was begun on the possible construction of a second engine test cell at MTI.
- A Mine Safety Equipment Gas Detection System is currently being installed throughout the ASE facility.

APPENDIX - TERMS AND DEFINITIONS

Term	Definition
AFC	Air/Fuel Control
Al	Aluminum
AMG	AM General
AOP	Average Operating Point
ASE	Automotive Stirling Engine
B	Boron
BOM	Bill-of-Materials
BSE	Basic Stirling Engine
C	carbon
°C	degrees Celcius
Cb	columbium
CES	Cold Engine System
CGR	combustion gas recirculation
Co	cobalt
CO	carbon monoxide
CO_2	carbon dioxide
Cr	chromium
CRU	Communications Register Unit
CSP	Cold-Start Penalty
CVS	constant volume sample
DAFC	Digital Air/Fuel Control
DAS	Data Acquisition System
DC	direct current
DCC	Digital Combustion Control
DEC	Digital Engine Control
DOE	Department of Energy
EDS	Engine Drive System
EGR	Exhaust Gas Recirculation
EHS	External Heat System
EHSTR	External Heat System Transient Response Code
°F	degrees Fahrenheit
Fe	iron
ft	foot
ft^2	square foot
ft^3	cubic foot
FY	fiscal year
g/mi	grams per mile
g/s	grams per second
HC	hydrocarbon
HES	Hot Engine System
hp	horsepower
HTP-40	High Temperature P-40 Engine
Hz	hertz
in	inch
kg	kilogram
ksi	thousand pounds per square inch
kW	kilowatt
lbs	pounds
LRFD	Lightweight Reduced Friction Drive

Term	Definition
m	meter
mi	mile
mm	millimeter
Mn	manganese
Mo	molybdenum
Mod I	first-generation automotive Stirling engine
MTI	Mechanical Technology Incorporated
MPa	megapascals
MQ	material quote
N	nitrogen
NASA	National Aeronautics and Spac Administration
Ni	nickel
NO_x	oxides of nitrogen
NTU	number of transfer units
PL Seal	Pumping Leningrader Seal
P_{max}	
P_{mean}	working gas pressures
P_{min}	
psi	pounds per square inch
psig	pounds per square inch gauge
PTFE	polytetraflouroethylene
P-V	pressure-volume
QAR	Quality Assurance Report
RESD	Reference Engine System Desig
RFD	Reduced Friction Drive
rpm	revolutions per minute
s	second
SES	Stirling Engine System
Si	silicon
S/N	serial number
SS	Stainless Steel
Ta	tantalum
T/C	thermocouple
T_{cgat}	combustion gas temperature after tubes
TTB	Transient Test Bed
USAB	United Stirling of Sweden AB
VE	Value Engineering
W	tungsten
λ	$\frac{air/fuel}{air/fuel_{stoichiometric}}$
η	efficiency
Δ_η	efficiency difference
ΔWt.	weight difference
ΔPower	power difference
ℓ/min	liters per minute