DOE/NASA/0032-15
NASA CR-167907-2
MTI 82ASE278SA2

AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

SEMIANNUAL TECHNICAL PROGRESS REPORT FOR PERIOD: JANUARY 1 – JUNE 30, 1982

Mechanical Technology Incorporated

(NASA-CR-167907) AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM Semiannual Technical Progress Report, 1 Jan. - 30 Jun. 1982 (Mechanical Technology, Inc.) 65 p HC A04/MF A01 CSCL 13F G3/85
N83-15177
Unclas
02279

October 1982

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

for
U.S. DEPARTMENT OF ENERGY
Conservation and Solar Applications
Office of Vehicle R&D

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MTI Report No. 82ASE278SA2

Semiannual Technical Progress Report

Period Covered:
January 1 - June 30, 1982

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MECHANICAL TECHNOLOGY INCORPORATED
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Approved By: N. P. Nightingale

Approved/Ratifed
By NASA: W. K. Tabata

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Mod I Engine Installed In Transient Test Bed

TABLE OF CONTENTS

TABLE OF CONTENTS CONTINUED

LIST OF FIGURES

LIST OF FIGURES CONTINUED

LIST OF TABLES

INTRODUCTION

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 consists 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 vehicle systems development and engine/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 make 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), defined as the best-engine design generated at any given time within the program that will provide the highest possible fuel economy and meet or exceed all other program objectives, utilizes all new technologies that are reasonably expected to be developed by 1984, and that 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 development improvements) is an engine design directed toward 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 program.

Since the Mod II could not be designed, it was reasoned that the Mod I could be used to develop and demonstrate RESD technology through a series of design upgrades, i.e., it was more cost-effective to use existing Mod I hardware than to design and build an entirely new set of Mod II engine hardware; hence, the "Proof-of-Concept" logic evolved.

Mod I engine hardware is to be used to "prove" the designs and technologies embodied in the RESD. In order to prove the RESD concepts, the necessity of conducting two upgraded designs of the Mod I was recognized. These two upgraded versions were identified as the Mod I-A and Mod I-B. Inherent limitations were also recognized in the proof-of-concept since Mod I hardware was larger than RESD and Mod II hardware.

During the past six months, the ASE Program directed its resources primarily toward the assembly and test of Mod I engines, and design initiation of the Mod I-A engine system and supporting component development. Effort has also been directed toward reducing the manufacturing cost of the RESD.

The four Mod I engines currently in the ASE Program have accumulated a total of more than 955 test hours. One engine has been installed in an AMC Lerma vehicle for transient performance testing.

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, covering the period of January 1 to June 30, 1982, is the second Semiannual Technical Progress Report issued under the contract, and includes technical progress only. Although the program has been modified to a proof-of-concept program, the objectives described below still apply to the RESD. The upgraded versions of the Mod I engines are 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: $NO_x = 0.4$ 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:

Task 1 - Reference Engine - 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.

Task 2 - Component/Subsystem Development - 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/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

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

advancing engine technology in terms of durability/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.

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. Two upgraded versions of the Mod I (Mod I-A and Mod I-B) will be conducted.

Task 5 - Deleted from the program.

Task 6 - Deleted from the program.

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.

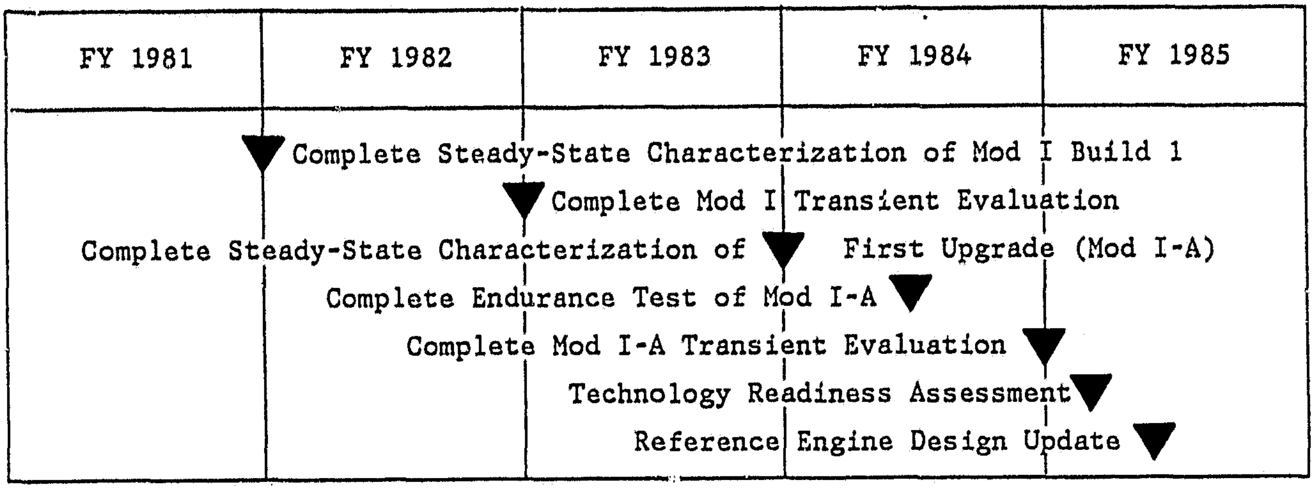
Task 9 - Program Management - Work under this task will provide total program control, administration, and management, including reports, schedules, financial activities, test plans, meetings, reviews, and technology transfer.

Program Schedule

A current schedule of the major milestones for the ASE Program is presented below:

STIRLING PROOF-OF-CONCEPT PROGRAM

MILESTONES



Program Status and Plans

A brief summary of the accomplishments made in the ASE Program during the last six months, and plans for the next semi-annual reporting period, are presented below:

MAJOR ACCOMPLISHMENTS

Mod I engine No. 1 completed an acceptance test with its digital control, and was shipped from USAB to AMG late in February, where it was installed in the AMC Lerma vehicle. The purpose of this effort was to develop systems using the Lerma vehicle as a transient test bed. Installation was completed by the end of May, and initial CVS testing was conducted at a 3750-pound inertia weight setting and a 11.1-hp road load. The results are tabulated below for hot starts:

Cycle	MPG	Emissions (g/mi) HC	CO	NO_x
Urban	20.6	.110	3.50	.348
Highway	29.2	.240	5.26	.240
Combined	23.8			
Total Particulates:			0.089	

Mod I engine No. 2 was assembled and tested for the first time on February 29, 1982 as a Basic Stirling Engine (BSE). A BSE provides the ability to evaluate the performance of the basic engine void of any losses associated with control systems and auxiliaries, which are required for an SES. The following list identifies those components installed in an SES, but not mounted on a basic engine:

- combustion air blower;
- start-up motor;
- air-atomizer compressor;
- oil-servo hydraulic system;
- electronic control;
- alternator;
- starter; and,
- variator.

Mod I engine No. 3 completed its acceptance test at MTI in June, 1982, and was disassembled, inspected, and shipped back to USAB for endurance testing.

A fourth ASE Program Mod I engine is the U.S.A.-built engine. As outlined in previous MTI reports, a majority of the engine's major parts were manufactured in the United States. The purpose of this activity was to establish American vendors and demonstrate the transfer of

Stirling engine technology in the area of manufacturing. This engine has been designated Mod I engine No. 10 to distinguish it from the other three engines, which were manufactured in Europe. During this report period, engine No. 10 was assembled up to the Hot Engine System.

Figure 1 shows the total test hours (955) accumulated in the Mod I engine program through June 30, 1982, and the breakdown per engine.

Mod I engine development has not been without its setbacks. The major concern focused around the high scrappage rate in cylinder and regenerator housing castings. The program has been significantly delayed because of the shortage.

During CGR*-combustor testing, the engines experienced large variations in circumferential temperature profile which, in turn, created nonuniform temperature patterns on heater head materials. Further, thermal stress failures caused by the rapid expansion of the combustor liner, which was confined within the preheater assembly, were experienced by the CGR combustor during cold starts. Further development of the CGR system within the component development area of the program was indicated; consequently, Mod I engines No. 1 and 3 have been retrofit with EGR** combustors.

Fuel nozzles have shown a tendency to coke during testing; this problem is also being addressed in the component development area.

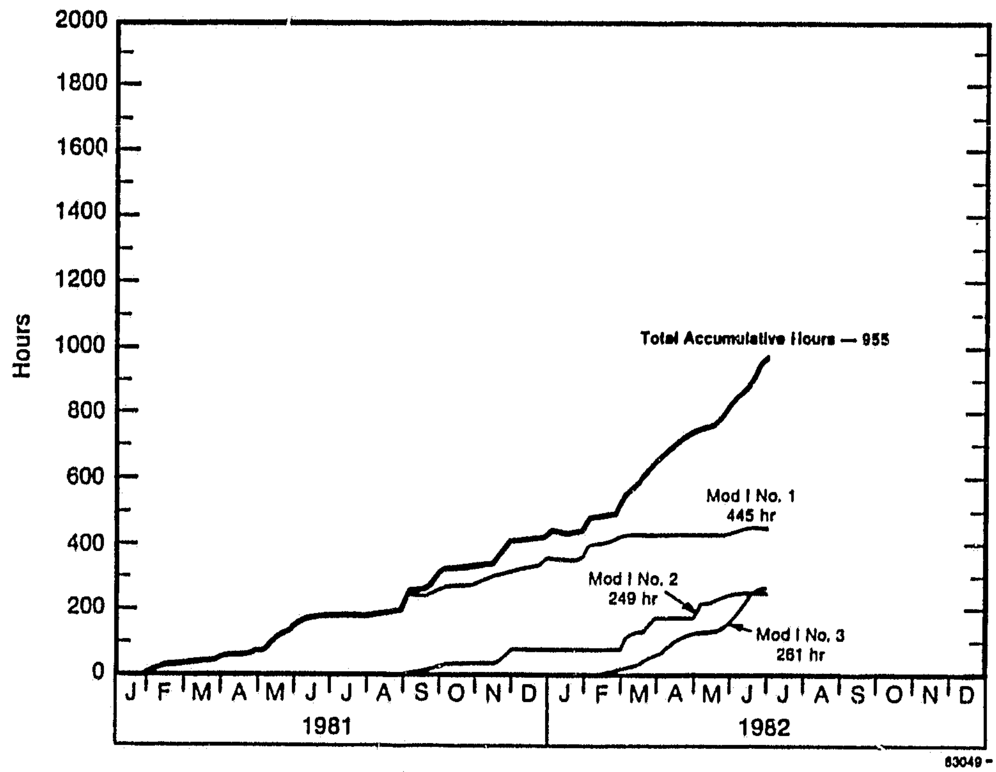


Fig. 1 Total Mod I Engine Test Hours

*combustion gas recirculation
**exhaust gas recirculation

Finally, use of the Electronic Digital Control System during transient test bed testing of Mod I engine No. 1 greatly increased the engine's cold-start transient start-up time from the slower start-up time experienced with the Analog Control System. As a result, the rear-row tubes and fins on the heater head thermally expanded at a more rapid rate than the inherently cooler manifold, causing areas of high stress in the braze between the tubes and manifold. To relieve this stress and provide flexibility in the tube arrangement, the fins were cut between each tube row from halfway above the manifold base.

The design effort to upgrade the Mod I engine was accelerated during this report period. Engine designers from USAB and MTI worked to an integrated schedule to complete designs on the various systems. All activities were directed at meeting design goals presented in the last semi-annual report (MTI Report No. 82ASE248SA1). A priority system for the design was established based on criticality to attainment of these goals.

First priority was given to the External Heat System where weight and size were to be drastically reduced; the preheater matrix has been redesigned with fewer plates and a smaller diameter. Preheater plate thickness has been reduced from .15 to 0.1 mm to decrease cold-start penalty.

Second priority was assigned to the Hot Engine System where the goal was to reduce weight and size, and remove materials containing expensive strategic elements such as cobalt. Cylinder and regenerator housing casting materials were changed from HS-31 (as used in the Mod I) to XF-818, and tube material was changed from N155 to CG-27. The operating temperature was raised from 720°C (as used in the Mod I) to 820°C (as called for in the RESD). Further part-power optimization was factored into the design, resulting in a smaller regenerator diameter. Correspondingly, the coolers were reduced in diameter. The Mod I-A will continue to use the Mod I aluminum water jacket.

Third priority was given to the controls/auxiliary systems where simplicity and weight reduction were major goals. Control blocks were redesigned to a smaller, lightweight configuration.

The Cold Engine System and Drive System were assigned the fourth and fifth priorities, respectively. Lighter piston bases, a simplified seal housing assembly, and an integrated cold-connecting duct-plate/cylinder-liner assembly have been designed to remove o-rings and simplify the designs. Currently, the Mod I-A design is on target for meeting its 100-pound weight-reduction goal and specific weight goal of 7.5 lb/hp.

Cylinder and regenerator housing castings have been ordered from the Howmet Corporation, with a scheduled delivery date of mid-October. Preheater matrices have been stamped; delivery is scheduled for the end of October.

In support of the RESD technologies embodied in the Mod I-A design effort, a comprehensive component development program has been activated. The Combustion Performance Rig finished an operational checkout phase at MTI, and is now undergoing baseline testing. In support of this Rig, a Free-Burning Rig was used to evaluate alternative fuel nozzles. A dual-orifice Delavan nozzle, which does not require the use of an atomizing air compressor, thus significantly improving its efficiency, was found to perform adequately. This nozzle will be tested in the Performance Rig as a full system. Activity to find and develop low-cost regenerator matrix materials continues. Two promising candidate materials have been identified.

The search for more reliable/durable main seals continues. The Exploratory Rig has been modified to operate 24 hours a day, seven days a week. The affect of changes in various main seal parameters was evaluated during this report period, concluding with a design that shows a leakage level three-to-four times better than with the Pumping Leningrader seal currently installed in the Mod I engines.

Extensive work has been applied in developing an air/fuel control system that would eliminate the K-Jetronic system. This was a design goal of the RESD, as well as the Mod I-A, in that it would reduce airflow losses and provide the capability to vary Lambda values over the operating fuel flow range. Rig tests have been conducted on the J-Tec airflow meter and a Fluidyne fuel flow meter. Extensive work has also been applied in programming and developing a digital control based on the TI9995 using compatible U.S.A. circuits and connectors.

Endurance testing on the High-Temperature P-40 (HTP-40) engine continued during this semiannual report period. The purpose of this testing was to evaluate material durability and their reactions to the combustion environment of actual engines at an operating temperature of 820°C. The HTP-40 was configured with one of two heater heads manufactured from castings and tube materials being developed within the ASE Program as alternatives to the existing cobalt-based materials of the Mod I. Table 1 summarizes the casting/tube combinations of the four quadrants initially assembled in the engines. At the end of June, the HTP-40 engine had completed 1000 of its scheduled 2000 testing hours.

TABLE 1 HTP-40 HEATER HEAD MATERIALS

Quadrant No.	Casting Material	Tube Material
1	HS-31	Inconel 625
2	CRM-6D	Sanicro 32
3	XF-818	12RN72
4	SAF-11	CG-27

Work Planned for Next Reporting Period

The following activities have been planned for the next semiannual report period:

- reconfigure Mod I engine No. 1 to an EGR system, and characterize engine in the test cell;
- characterize transient behavior of Mod I engine No. 1 in the Lerma vehicle;
- begin endurance-test program on Mod I engine No. 3;
- complete steady-state characterization of Mod I engine No. 10;
- complete Mod I-A design schedule;
- begin assembling two Mod I-A engines;
- complete development of Delavan nozzle for EGR system;
- complete development of the H-ring piston system; and,
- initiate statistical engine testing of Mod I PL seals on three P-40 engines.

During the first half of 1982, Mod I automotive Stirling engine activity included test cell/Transient Test Bed (TTB) installation testing of Mod I engines No. 1, 2, and 3. A fourth engine (Mod I engine No. 10), manufactured in the United States, was nearing assembly completion and test cell preparation by the end of June. Total accumulated Mod I engine test hours for this semiannual report period are 955 (includes 89 miles and 22.6 hours of TTB operation).

A significant change in Mod I External Heat System (EHS) occured during this report period, i.e., early data from Mod I engine No. 1 (tested at USAB) indicated a large temperature difference between cycle working gas temperatures. This difference was directly related to a poor temperature distribution with the CGR combustor. The differences between the minimum and maximum measured working gas temperatures were on the order of 50-125°C over the engine's fuel input range with CGR. Testing with a non-CGR combustor over the engine's fuel range resulted in a 10-25°C working gas temperature spread.

A second problem encountered with the CGR combustion system was with thermal growth, i.e., during shortened engine start sequences (high heat input rates), the temperature differences between the combustor and preheater caused distortion of the mating sheet metal hardware between the two pieces.

Because of these two problem areas, a decision was made to return the CGR system to the component development stage and use an EGR combustion system that would improve the temperature distribution problem and, since it is less massive, solve the distortion problems with mating hardware.

Heater head delivery caused some delay in planned testing. Mod I engines No. 3 and 10 suffered delays in testing and assembly, respectively. The USAB casting vendor, Bulten Kanthal, had problems in manufacturing castings for the heads. Castings intended for Mod I engine No. 10 were sent to Sweden and used for the remaining ASE Program engines.

These delays caused the No. 3 engine to complete its acceptance test at the end of June, rather than at the end of May, as originally planned. Engine No. 10 will be forced to begin running with the spare program head rather than with its own U.S.A.-manufactured part.

Mod I Engine No. 1

In January, 1982, Mod I engine No. 1 completed transient testing with the Digital Control, and was shipped to AMG during February, where the engine was installed in the Lerma TTB with the USAB Digital Control System. The engine was started in the TTB in April and, following final TTB preparation, was tested during June at the Mercedes Benz Corporation of Ann Arbor, Michigan.

Engine No. 1 was received from USAB with a CGR combustor. TTB data obtained during CVS testing (at 3750-lb. inertia weight setting, and a 11.1 hp road load) is tabulated in Table 1-1. (All testing was conducted with hot engine starts.)

TABLE 1-1

CVS TESTING DATA OF TTB

	(g/mi)				
Cycle	HC	CO	CO_2	NO_x	MPG
Urban	.110	3.50	425.4	.348	20.567
Highway	.240	5.26	294.3	.240	29.187
Total Particulates - .089 g/mi					
Combined Fuel Economy - 23.75 mpg					

In order to meet the ASE Program milestone by the end of September, 1982, Mod I engine No. 1 is scheduled for removal from the TTB, conversion to an EGR configuration, characterization in the Test Cell, reinstallation, and baseline evaluation for emissions and transient behavior.

Mod I Engine No. 2

During March, Mod I engine No. 2 completed its acceptance test as a Basic Stirling Engine (BSE) (complete engine less auxiliaries) while at USAB. The engine was then converted to a Stirling Engine System (SES) (complete engine with auxiliaries) and used to map exhaust emissions levels with various percentages of EGR. EGR mapping was completed in April, and the data was used to develop an EGR schedule for an EGR-equipped Mod I engine. Following completion of EGR mapping, performance mapping of the engine as an SES began and is continuing. Figures 1-1 through 1-3 present the power, efficiency, and fuel consumption results of engine No. 2 as a BSE.

Mod I Engine No. 3

Mod I engine No. 3, shipped from USAB to MTI in February, was installed in MTI's Test Cell and started for the first time, representing the first Mod I engine to be run in the United States. Since engine No. 3 was shipped with a heater head not capable of more than 5 MPa mean working gas pressure, characterization testing had to be delayed until May.

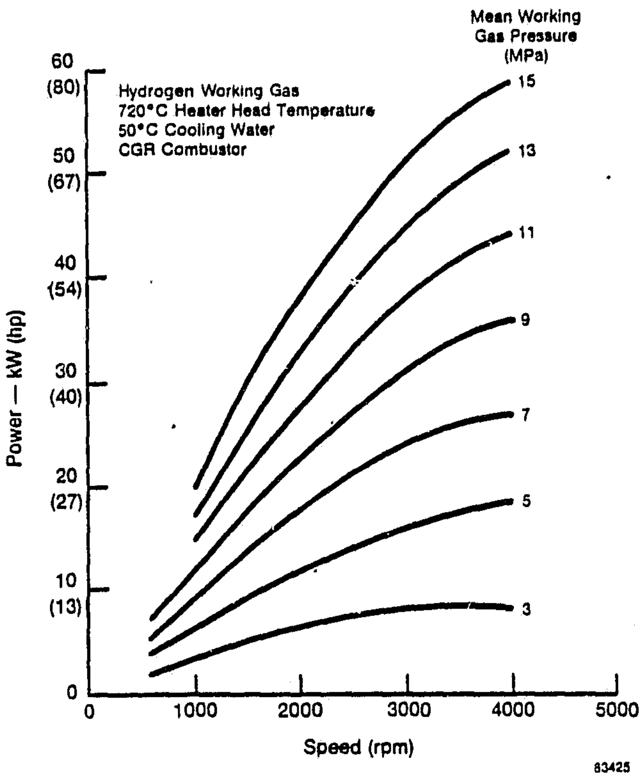


Fig. 1-1 Mod I Engine No. 2 BSE Power

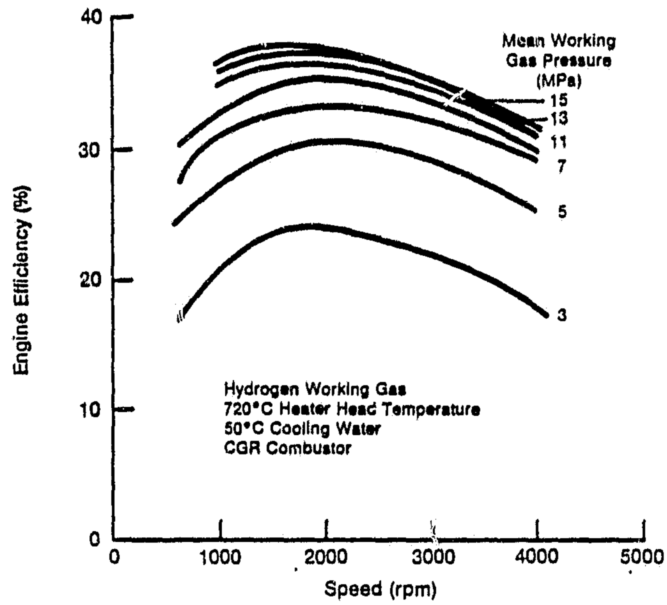


Fig. 1-2 Mod I Engine No. 2 BSE Efficiency

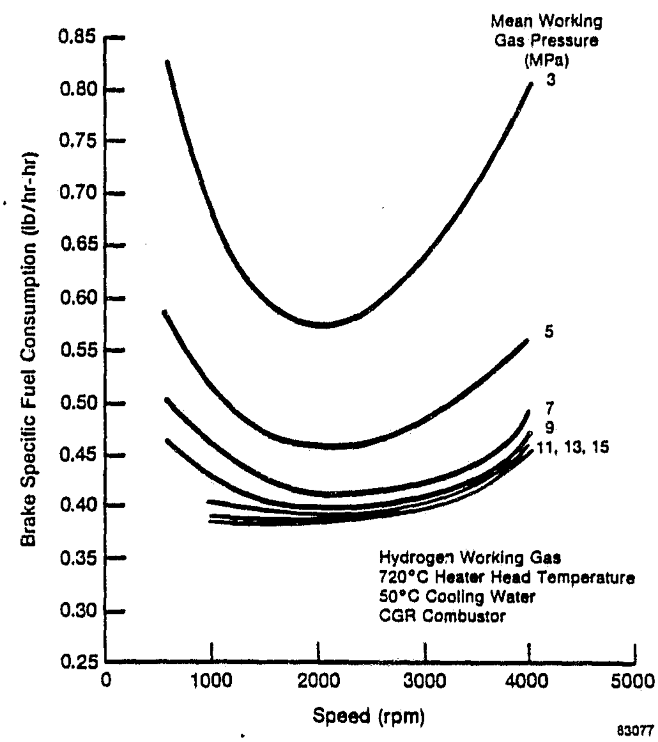


Fig. 1-3 Mod I Engine No. 2 BSE Fuel Consumption

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Initial full performance testing of the engine was used to measure the affects of EGR on engine performance and, with the aid of engine No. 2 mapping, establish an EGR schedule for all the Mod I engines. The established schedule, which had minimum affects on engine performance while maintaining acceptable NO_x levels, was a simple on/off orifice device scheduled as:

- no EGR during starts or until a stabilized combustion system was attained;
- orifice was sized to provide 20 to 25% EGR at a 1.0 g/s fuel flow; and,
- EGR was to be used over the complete fuel schedule.

Following the establishment of this schedule, engine No. 3 was set up to run its official acceptance test. The test was completed in June, and the engine was removed from the Test Cell. Figures 1-4 through 1-6 present the engine's power, efficiency, and fuel consumption characteristics during its acceptance test.

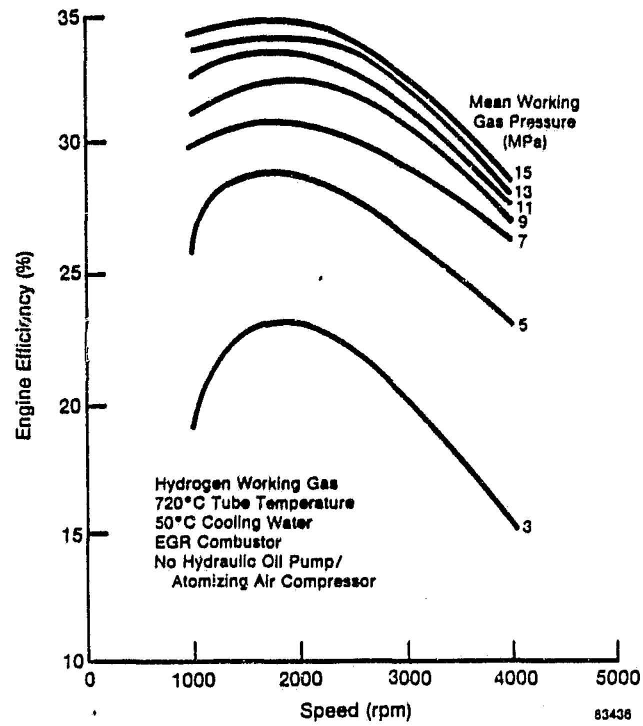

Fig. 1-5 Mod I Engine No. 3 SES Efficiency

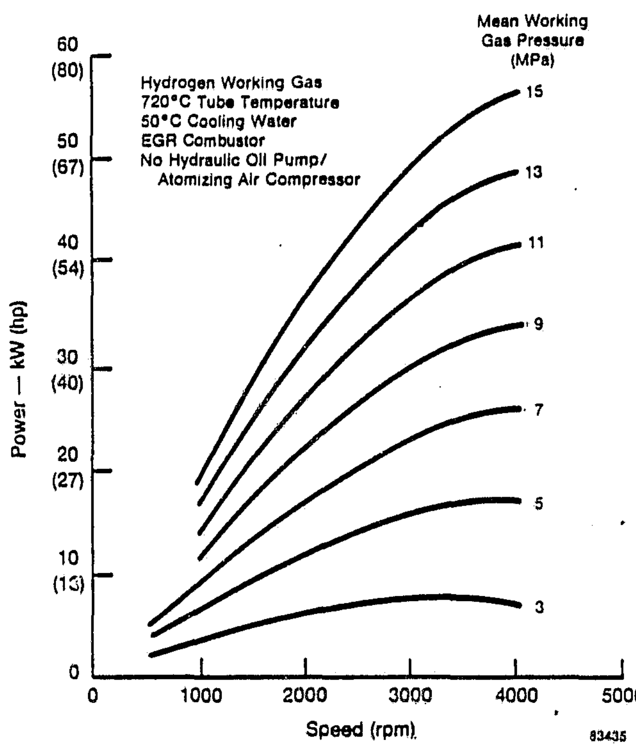

Fig. 1-4 Mod I Engine No. 3 SES Power

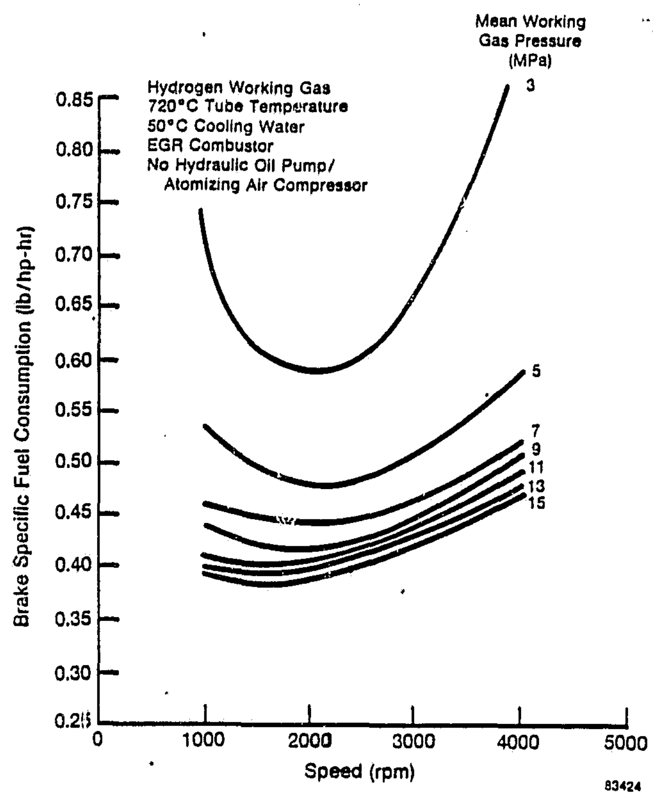

Fig. 1-6 Mod I Engine No. 3 SES Fuel Consumption

Mod I Engine No. 10

The assembly of U.S.A.-manufactured Mod I engine No. 10 began with the drive unit assembly and measurements during January, 1982. Assembly of the remaining parts was completed as the respective parts became available. The major item delaying the build was a heater head. Mod I engine No. 10 is currently assembled as a BSE, and will be installed in MTI's Test Cell for acceptance testing by the end of October.

Upgraded Mod I (Mod I-A)

An RESD is representative of technology that will be available in the mid-1980's. The program provides a logical development plan to reach that goal via a proof-of-concept approach. Upgrading of the Mod I engine will be accomplished during program years 1982 and 1983 to demonstrate the achievement of proof-of-concept in an engine environment. Goals for the Mod I-A were established consistent with a logical growth path toward RESD performance levels; these goals are presented in Table 1-2, and their relationship to the current Mod I and RESD are shown in Figures 1-7 through 1-11.

TABLE 1-2

MOD I-A GOALS

Power	kW (hp)	58	(78)
Weight	kg (lb)	263	(580)
Specific Weight	kg/kW (lb/hp)	4.50	(7.50)
Brake Net Efficiency (%)	- Maximum	39	
	- Part-Power	33	
Specific Fuel Consumption (lb/hp-hr)	- Minimum	0.355	
	- Part-Power	0.420	
Cold-Start Penalty (g)		140	

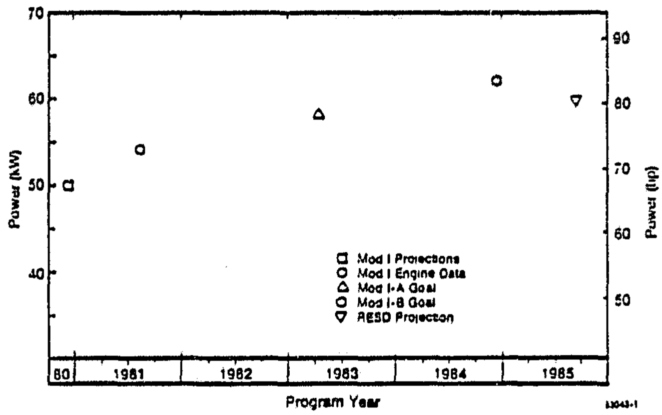

Fig. 1-7 Engine Power

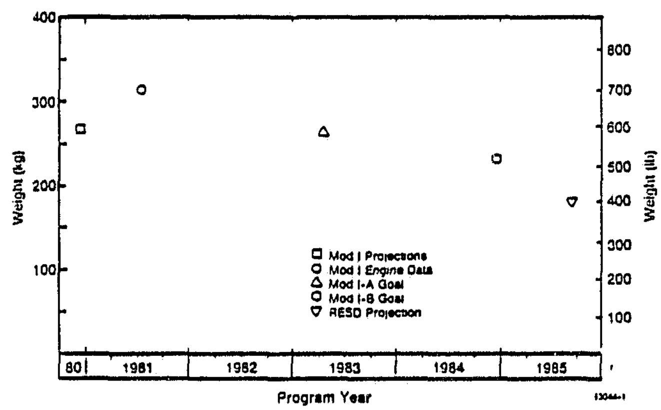

Fig. 1-8 Engine Weight

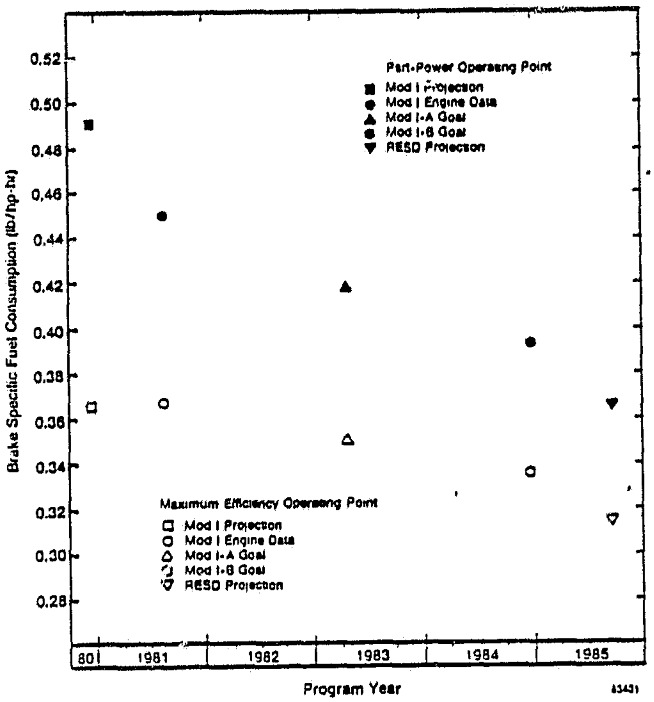

Fig. 1-9 Specific Fuel Consumption

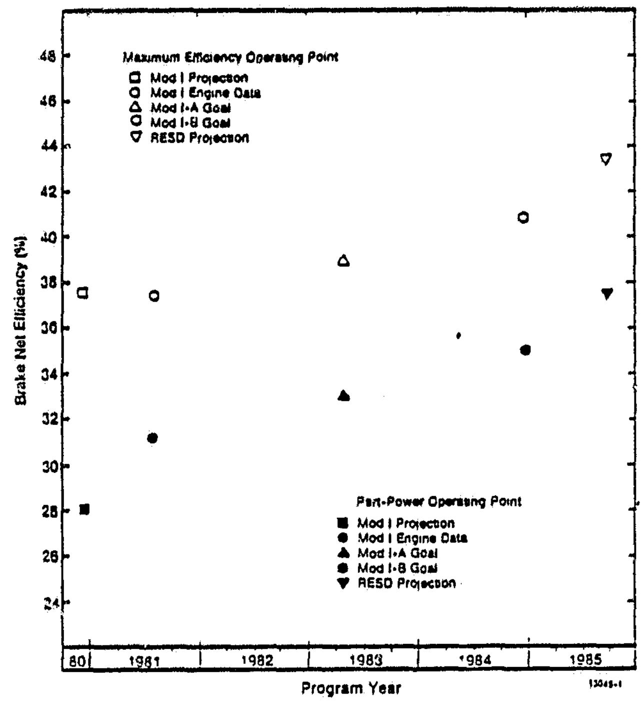

Fig. 1-10 Brake Net Efficiency

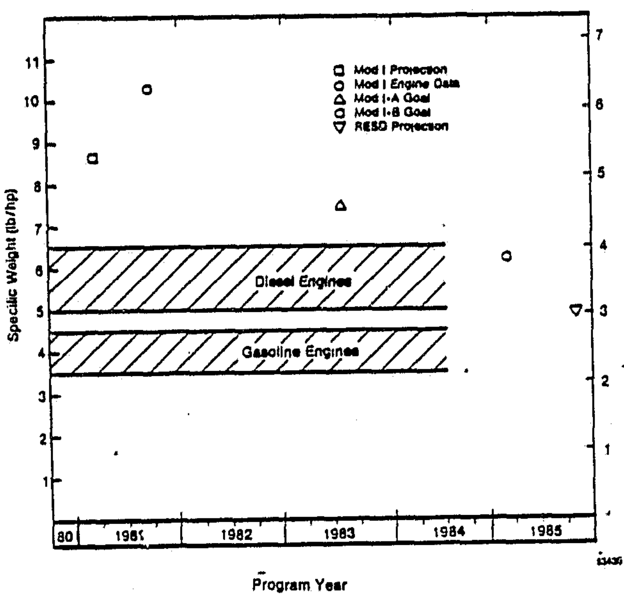

Fig. 1-11 Engine Specific Weight

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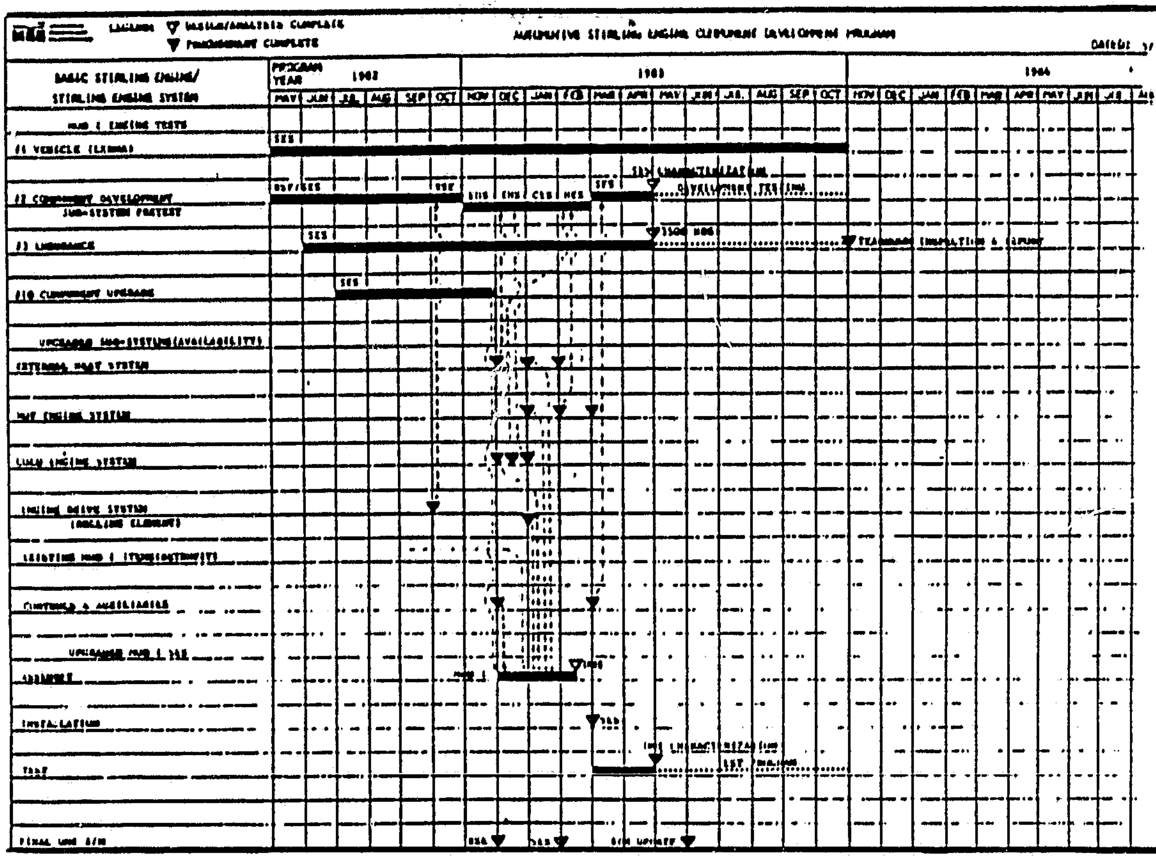

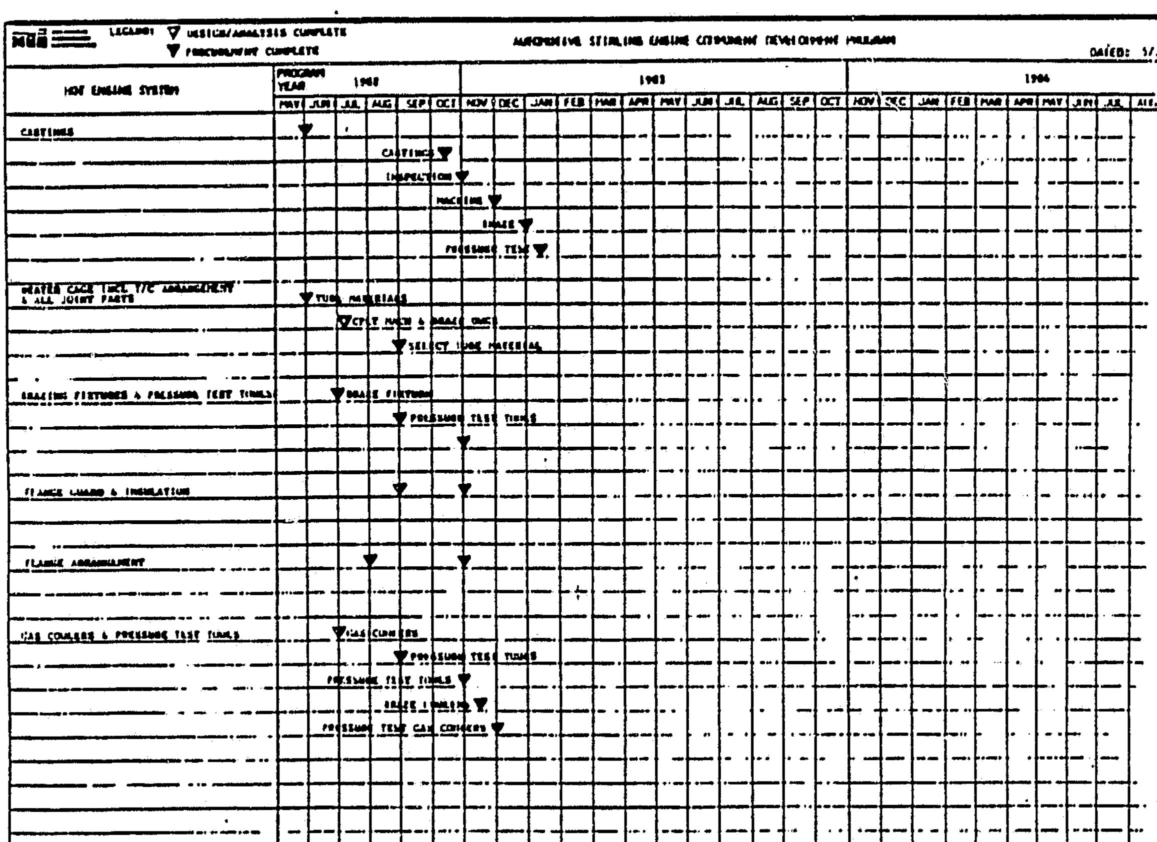

Fig. 1-12 Mod I-A Design Schedule

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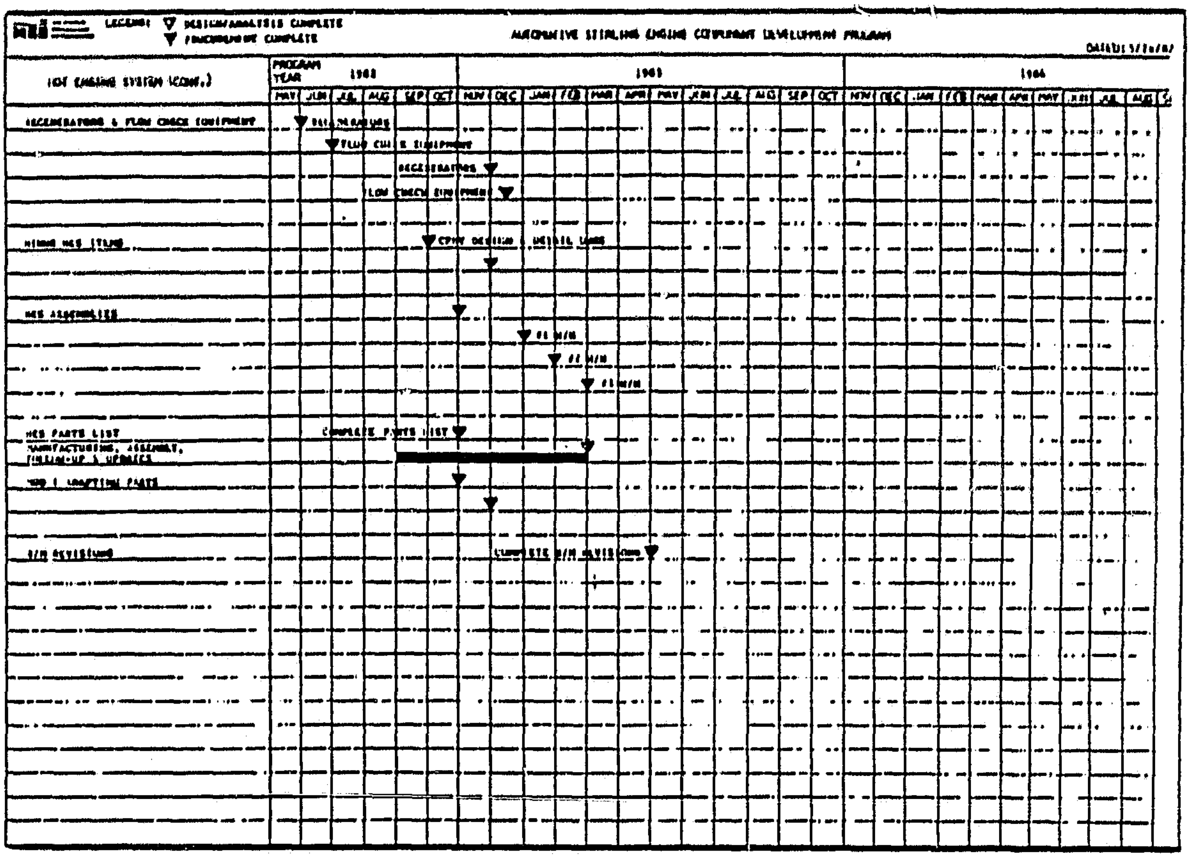

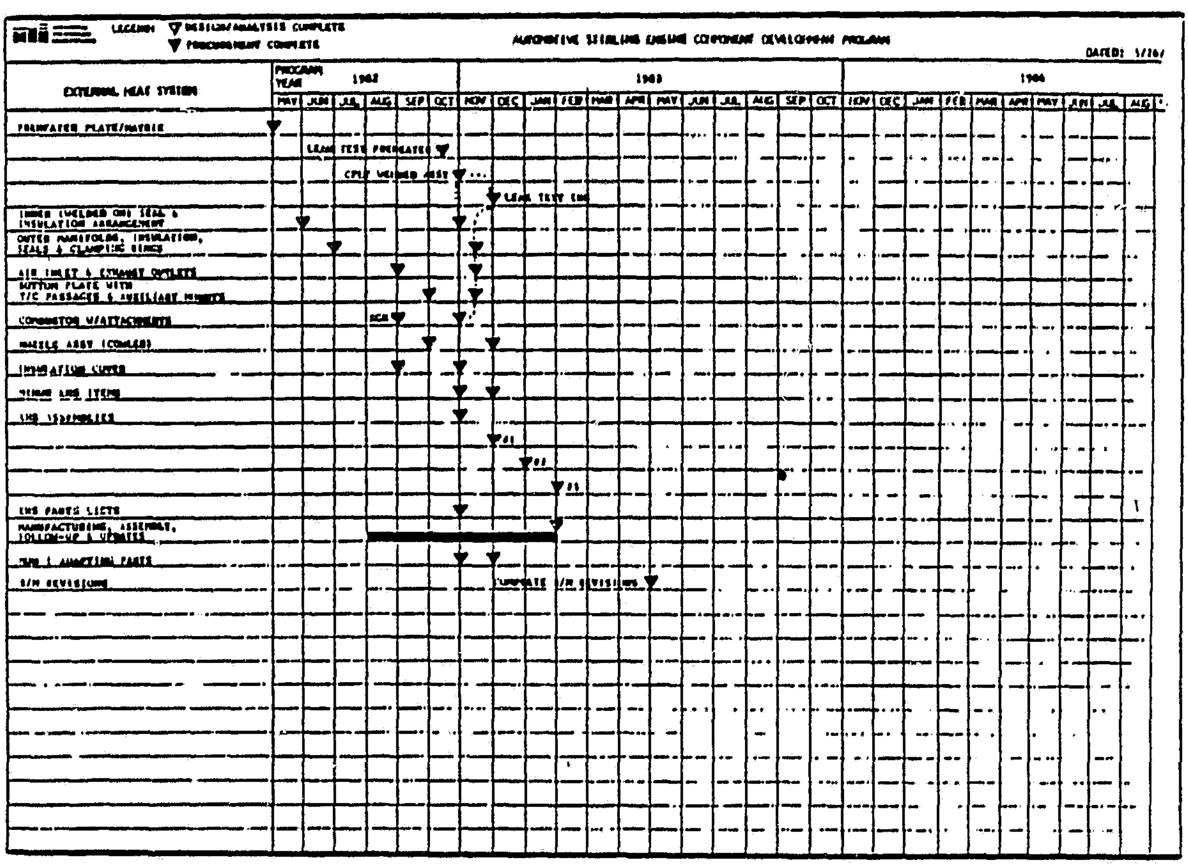

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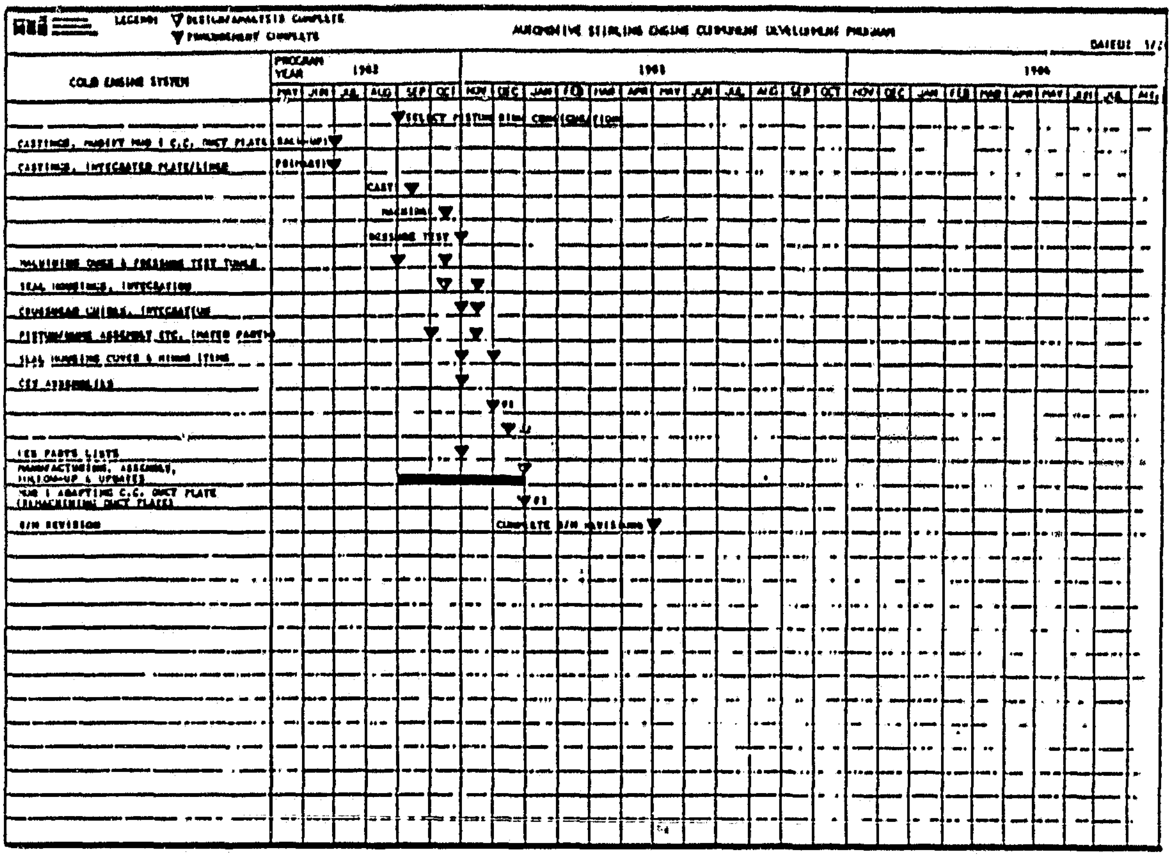
Fig. 1-12 Mod I-A Design Schedule (Continued)



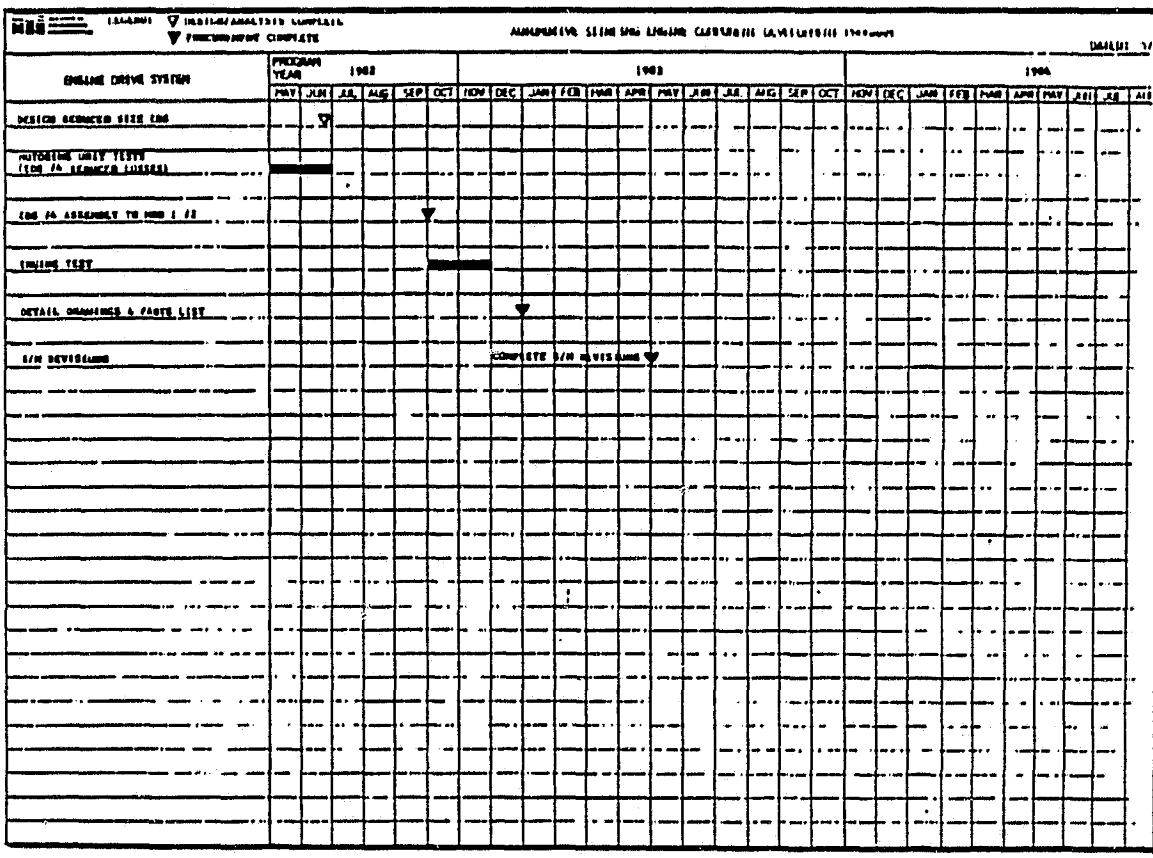


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Fig. 1-12 Mod I-A Design Schedule (Continued)

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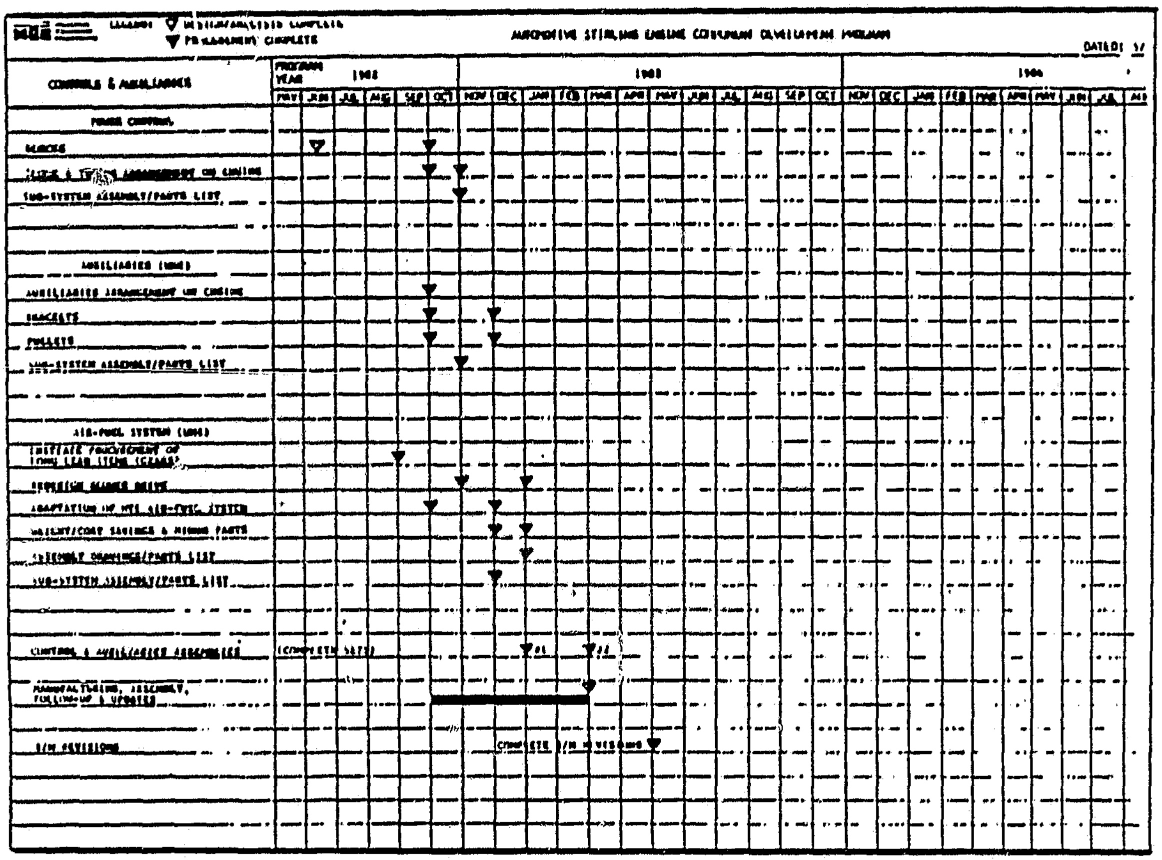

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Fig. 1-12 Mod I-A Design Schedule (Continued)

DESIGN FEATURES

Several system changes will be addressed in the Mod I-A design. The major thrusts, in addition to meeting the goals noted above, are to:

- reduce engine size (this will be addressed by reducing the diameter of the External Heat System);
- increase reliability (the number of sealing surfaces will be reduced as much as possible; dynamic seal systems (piston rings/main seals) will be improved); and,
- decrease strategic material useage (material selection will address the use of alloys with low strategic material content.

Implementation of these changes will be pursued by the following approach for each major subsystem:

- External Heat System - Reduce the number and thickness of the preheater plates to enable reduction in preheater diameter. Select a material that is lower in chromium content, and improve combustion system to eliminate need for atomizer air.
- Hot Engine System - Redesign the heater head castings for improved castability, utilizing an alternate material with no cobalt content. Optimize performance for part-power operation, and identify a reduced cost regenerator material.
- Cold Engine System - Incorporate a one-piece cold-connecting-duct cylinder liner casting to reduce the number of o-ring seals. Redesign piston, piston rings, and main seal area to provide reduced weight, lower friction, and improved sealing.
- Drive Unit - Design Reduced Friction Drive incorporating rolling elements bearings.
- Auxiliaries and Controls - Modify auxiliaries to correct deficiencies noted on Mod I. Configure new air/fuel control system to provide improved Lambda control.

The specific design approaches were selected for these systems, and a master schedule (Figure 1-12) was established for design, procurement, and assembly to meet a test date of April, 1983. A layout of the selected Mod I-A design is shown as Figure 1-13.

During this semiannual report period, long-lead-time designs associated with the External Heat and Hot Engine Systems were completed, as was the selection of key materials.

HEATER HEAD CASTINGS

The current Mod I engine heater head is made from Haynes Stellite 31, a high-cobalt alloy. A materials search was conducted to identify an alternative material with the capability of matching ASE requirements in terms of reduced strategic material content, acceptable strength, good casting properties, weldability, and reduced cost. Several alloys were reviewed using a weighted value system whereby a rated value was established for each criteria item and alloy (relative to the best alloy in the group for each criteria), and a "total" value was obtained. The final selection matrix is shown in Table 1-3.

SAF-11 was eliminated due to cracking problems in the investment casting pieces. As noted in Table 1-3, XF-818 was the selected material. The heater design itself concentrated on regenerator housing modification to attain improved castability/reduced weight. A domed shape with a raised manifold was incorporated, making the housing more like a pressure vessel, and providing the potential for thinner walls/reduced weight. The raised manifold provides improved castability with a stronger ceramic core, and improved regenerator flow distribution with the central manifold inlet. Cylinder and regenerator final designs are shown in Figures 1-14 and 1-15.

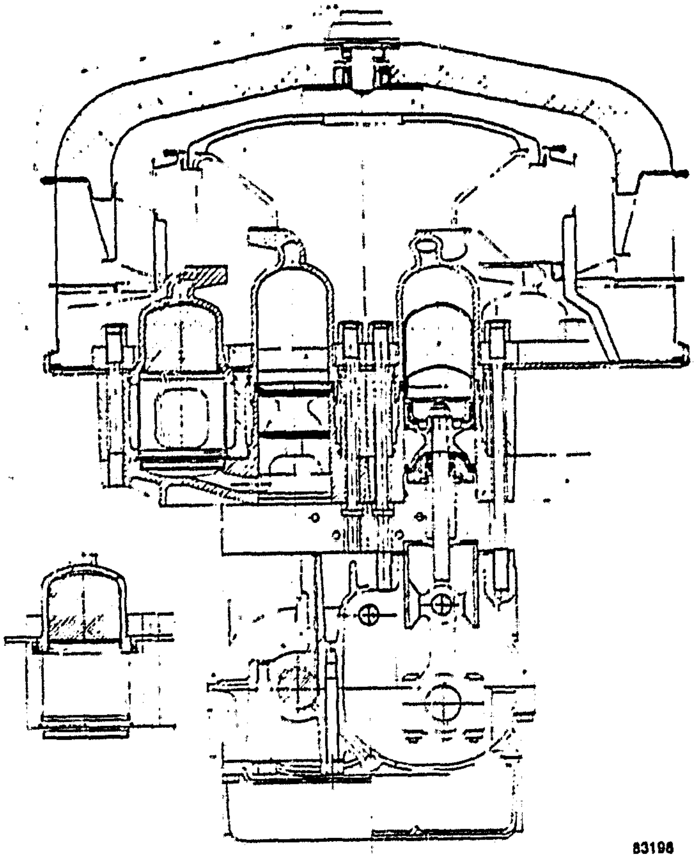

Fig. 1-13 Mod I-A Layout

TABLE 1-3

SELECTION MATRIX
ALTERNATE CASTING MATERIAL

	Weighted Value	Rated Value			Weighted Value x Rated Value: Highest Number is Selection		
	(Multiplier)	XF-818	CRM-6D	SAF-11	SAF-11	CRM-6D	SAF-11
Strategic Materials	40	10.0	9.56	9.30	400	382.4	(372.0)
Weldability	20	10.0	1.904	3.33	200	38.0	(66.0)
Fatigue — Low Cycle	10	10.0	10.0	10.0	54	106.0	(100.0)
Fatigue — High Cycle	10	7.9	7.9	10.0	82	89.0	(100.0)
Castability	--	-	9.6	-	-	--	---
Creep	10	10.0	10.0	9.9	100	96.0	(99.0)
Cost and Density	10	4.7	-	2.9	47	100.0	(29.0)
U.S.A. Availability	--	-		-	-	--	---
Total					834	795.4	(766.0)

( ) Rated for reference only.

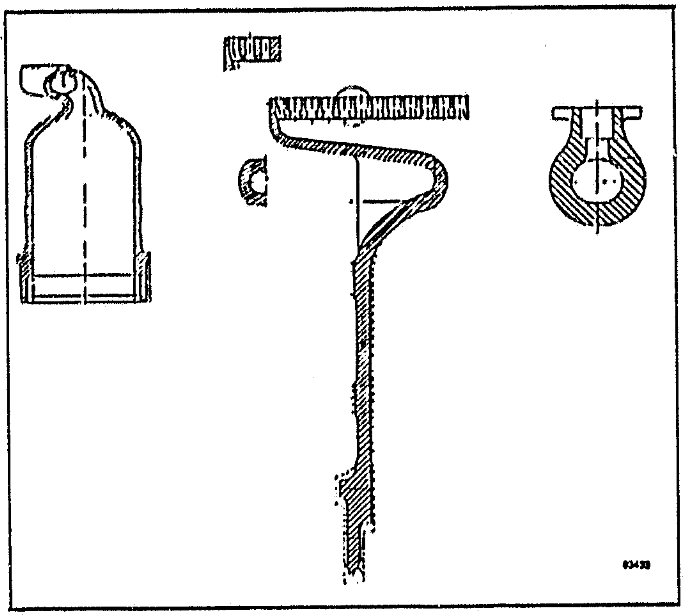

Fig. 1-14 Mod I-A Cylinder Housing

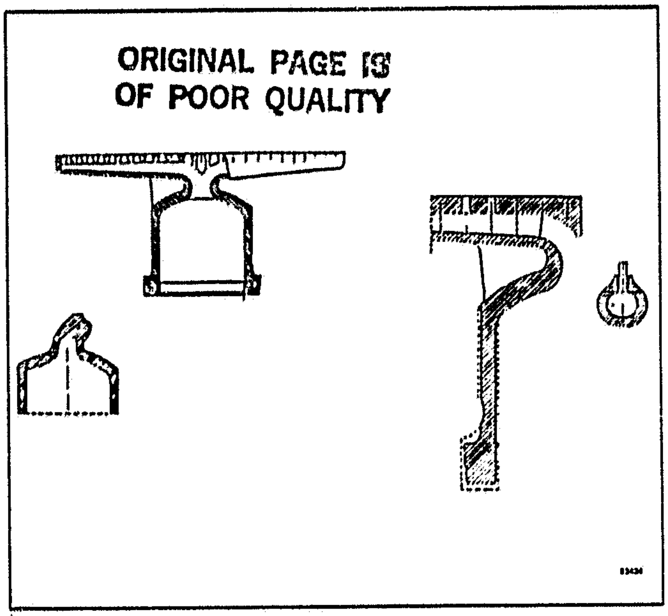

Fig. 1-15 Mod I-A Regenerator Housing

REGENERATOR

Regenerator size and material were selected during this report period, and the regenerator was optimized for highest efficiency at part-power operation. Analytical projections indicate no penalty in max power level. Metex knitted wire cloth was selected as the desired regenerator matrix material due to the substantial cost-reduction benefit achieved with minimum loss in performance.

PREHEATER

The preheater design is complete, and a material selection has been made. The preheater matrix will have a metallic composition similar to the Mod I; however, matrix plate thickness has been reduced from .15 to .1 mm, and the number of plates from 1200 to 1100, resulting in a 60-mm (2.37-in.) reduction in preheater diameter. The selected material, Sandvik 253 MA (instead of 310 SS), gives a decrease in chromium content from 25 to 21%. Tests are being conducted on two other materials, Armco 12SR and 18SR (12 and 18% chromium, respectively), thus providing further potential for cost and chromium-content reductions.

DRIVE UNIT

The design of a Reduced Friction Drive Unit featuring the use of rolling element bearings was completed during this report period. In addition to reduced friction losses during steady-state operation, cold-start losses can be reduced with the use of a thinner oil allowed by these rolling element bearings.

Overall Mod I-A Status

WEIGHT AND PERFORMANCE STATUS

Estimates have been made of the Mod I-A design in terms of weight, power, and efficiency, relative to the Mod I and Mod I-A goals, which are summarized in Table 1-4 and shown in more detail in Table 1-5. At this point in time, all goals are expected to be met.

TABLE 1-4

CURRENT MOD I-A STATUS
RELATIVE TO GOALS

Item		Goal	Current Mod I-A
Weight	(kg)	-47.00	-47.6
	(lbs)	-103.60	-105.0
Max Power	(kW)	+ 4.22	+14.9
	(hp)	+ 5.65	+19.9
Efficiency	(%)	+ 9.00	+10.8

TABLE 1-5

MOD I-A SCORECARD

Weight Summary	
Item	ΔWt. (kg)
Preheater/Combustor	- 9.2
Air Inlet Manifold	- 5.5
Heater Head	- 6.0
Regenerator	- 1.4
CCD/Liner	- 2.6
Piston/Dome	- 1.6
Drive Unit	- 6.0
Controls/Auxiliaries	-15.0
Total	-47.6
Goal	-47.0

Max Power/Max η Summary		
Item	ΔPower (kW)	Δη(%)
H-Rings	+ 3.0	+ 3.5
Air-Atomizing Compressor	+ 0.4	0.8
Reduced Oil Pump Capacity	+ 0.5	0.3
Part-Power Regenerator	--	1.0
Drive Unit	+ 1.0	1.0
Set Temp (820°C)	+10.0	+ 4.2
Total	14.9	10.8

Cold-Start Penalty	
Reduced Size P/H	- 4.5%
Heater Head/Regenerator	-14.9%
Rolling Element Drive (Mobil 1)	- ? %
Total	-19.4%

COST ESTIMATE

A cost estimate prepared for the Mod I-A design is compared in Table 1-6 to existing Mod I hardware, and to potential further improvements achievable within the Mod I-A program. Substantial cost reductions achieved with the Mod I-A design are further detailed in Table 1-7.

TABLE 1-6

MOD I-A MANUFACTURING COST
ESTIMATES PER DRAWING/DATA OF
FEBRUARY, 1982

Hardware	Class 1[1] ($)	Class 2[2] ($)	Class 3[3] ($)
Cylinder Housing	410.00	106.24	106.24
Regenerator Housing	387.00	100.40	100.40
Heater Tubes	166.00	166.00	131.80
Regenerator Matrix	362.28	91.00	91.00
Gas Cooler Assembly	458.72	458.72	157.72
Preheater Matrix	251.58	169.32	141.05
Cap Seals	1.00	0	0
Water Jacket	16.98	16.98	16.98
Duct Plate	26.16	0	0
Cylinder Liner	109.77	0	0
Integrated Duct Plate/ Cyl. Liner	0	45.82	45.82
Crosshead Guides	57.27	57.27	57.27
Crankcase	52.37	52.37	52.37
Oil Sump	19.16	19.16	1.95
Gears	45.75	45.75	19.34
Total	2,364.04	1,329.03	921.94

[1]Existing Mod I Hardware
[2]Mod I-A Design (2/1982)
[3]Value-Engineering Proposals - Mod I-A

TABLE 1-7

MOD I-A MAJOR PARTS'/
SUBASSEMBLIES' COST COMPARISON

Component	Class 1 Existing Hardware	Class 2 Upgraded Mod I	Class 3 V. E. Prop.	Remarks Material/Design/Specification/Weight Changes
Hot Engine System Cylinder Housing (1-17036)	$ 410.00	$ 106.24	$ —	C1 - Material HS-31: cost $365; labor $10.72; mfg. burdens $30.24; scrap $3.92. C2 - Material XF-818: cost $75.40; labor $8.24; mfg. burdens $21.64; scrap $.96.
Regenerator Housing (1-17038)	$ 387.00	$ 100.00	$ —	C1 - Material HS-31: cost $341.64; labor $9.80; mfg. burdens $31.68; scrap $3.68 C2 - Material XF-818: cost $68.92; labor $7.80; mfg. burdens $22.80; scrap $.88.
Heater Tubes (3-17042 (92); 3-17120 (4); 3-17119 (4))	$ 166.00	$ —	$131.80	C1 - Material IN 625: cost $134.88; labor $11.64; mfg. bur. $17.56. No change in C2. C3 - Alt. material CG27: cost $101.16; labor $11.64; mfg. burdens $17.56. Other alt. mat.: Sanicro/Sanicro 31H; 12RN72
Regenerator Matrix (3-17041C)	$ 362.28	$ 91.00	$ —	C1 - Material Pheonix 304 SS wire cloth (.0021" dia.; .0029" gap, 200 mesh): cost $351.68; labor $5.64; mfg. burdens $4.64. C2 - Mat. 304 SS knit. (.0035" dia. Metex wire).
Gas Cooler Assem. (3-17021)	$ 458.72	$ —	$157.72	C1 - Material SS 321: cost $.1883 x 1796 qty. = $338.19; labor $17.96; mfg. bur. $35.92; other parts/scrap $66.65. C3 - Material Carbon Steel Tubes Galvanized: cost $.02 x 1796 qty. = $35.92; labor $17.95; mfg. burdens $35.92; other parts/scrap $67.92.
External Heat System Preheater Matrix (1-17054)	$ 251.58	$ 169.32	$141.05 $175.00	C1 - Material 1200 310SS Plates (.15mm thick): cost $185.16 (plates)/$32.16 (other); labor $8.57; mfg. burdens $25.12; weight 20.22 lbs. (plates)/4.57 lbs. (other). C2 - Material 1100 Sandvik 253 MA Plates (.10mm thick, 12.35 lbs.): cost $107.43; labor $7.55; mfg. burdens $22.13; others cost $31.88; weight 4.53 lbs. C3 - Alt. Mat. ARMCO 12SR/18SR: cost $79.15 (C2 design); labor $7.55; mfg. burdens $22.13; (C3) Ceramic preheater when fully devel. Est. cost will reduce. An active program in C3.
Combustion Chamber Assy. (1-17064)	$ 47.58	$ —	$ —	C1 - Material Nim. 75 ($21.52)/329 SS ($15.17): cost $36.69; labor $2.30; mfg. bur. $8.30. C2 - Sufficient information not yet available.
(Turbulator Mod I) (4-17218) (included for information only)	($4.80)	$ —	$ —	C1 - Material 329 SS Investment Casting: cost $2.75; labor $.24; mfg. burdens $1.77.
Cold Engine System Piston/Piston Rod Seal Assembly* (2-17332 (2); 2-17333 (2))	$ 146.80	$ 145.80	$ —	C1 - Material IN 718/Nitroloy/8625 HR: cost $89.96; labor (Rulon LD) $15.80; mfg. burdens $42.08. C2 - Current information does not justify cost reduction; material, weight, mfg. remain almost unchanged; only capseals are eliminated (cost $.25 each).
Water Jacket (1-17045)	$ 16.98	$ 16.98	$ —	C1 - Material AL Sand Casting (reest. w/o duct plates): cost $13.78; labor $.67; mfg. burdens $2.36. No change for C2.
Duct Plate (1-17025)	$ 26.16	$ —	$ —	C1 - Material Nodular Iron Cast.: cost $17.97; labor $1.65; mfg. burdens $6.32. C2 - Integrated with cylinder liner.

TABLE 1-7 CONTINUED

Cylinder Liner (3-17231)	$ 109.77	$ —	$ —	C1 - Material 410 SS Tubing: cost $102.25; labor $2.20; mfg. bur. $5.20. C2 - Integrated with duct plate.
Duct Plate/Cylin. Liner Integrated (no drawing)	$ —	$ 45.82	$ —	C2 - Material Single Integrated Ductile Iron Casting: cost $30.46; labor $4.00; mfg. burdens $11.00. No changes in C1.
Crosshead Guides (1-17315)	$ 57.27	$ 57.27	$ —	C1 - Material AL Die Casting & Centrifugal Cast Grey Iron Tube Liner: cost $23; labor $8.74; mfg. burdens $25.03. C2 same as C1.
Drive System Crankcase (1-61116)	$ 52.37	$ 52.37	$ —	C1 - Material AL Die Casting (reest. w/o cross-head guides): cost $24.75; labor $4.92; mfg. burdens $22.25. C2 same as C1.
Bedplate (1-61117)	$ 47.45	$ —	$ —	C1 - Material AL Die Casting: cost $20.88; labor $4.55; mfg. burdens $21.64. C2 same as C1.
Intercasing (1-61121)	$ 18.26	$ —	$ —	C1 - Material AL Casting: cost $8.70; labor $1.06; mfg. burdens $8.36.
Oil Sump (1-61119)	$ 19.16	$ 19.16	$ 1.95	C1 - Material AL Casting (reest. cost): cost $10.15; labor $1; mfg. burdens $7.88. C2 same as C1. C3 - Stamp execution.
Crankshaft (Cyl. 1 & 2) (1-61112)	$ 42.91	$?	$ —	C1 - Material Cast Ductile Iron (Alloyed): cost $12.01; labor $7.01; mfg. burdens $23.57.
Crankshaft (Cyl. 3 & 4)	$ 50.91	$?	$ —	C1 - Material Cast Ductile Iron (Alloyed): cost $17.01; labor $7.41; mfg. burdens $26.11.
Main Drive Shaft (1-6144)	$ 21.71	$?	$ —	C1 - Material AlSl-1020 Forging: cost $4.62; labor $3.64; mfg. burdens $13.30.
Helical Gear (1) - Drive Shaft (2-61168)	$ 18.77	$ —	$ 8.33	C1 - Material Steel BS970PT3: cost $3.20; labor $2.59; mfg. bur. $12.85. No change in C2. C3 - Material Rolled Helical Gear Induc. Hard.: cost $3.20; labor $1.02; mfg. bur. $4.06.
Helical Gear (2) Crankshaft (2-61169)	$ 26.98	$ —	$ 11.01	C1 - Material Steel BS970PT3: cost $3.08; labor $3.64; mfg. bur. $20.08. No change in C2. C3 - Mat. Rolled Helical Gear Induc. Hard., No Grinding, Teeth Burnished: cost $3.08; labor $1.54; mfg. burdens $6.33.
Main Bearings (12) (3-61109)	$ 9.60	$?	$ —	C1 - Material Vandervell Drwg. L10219/2 Proprietary Bearings: Purchased finished at $.80 each. C2 - Decision on using roller bearing to be made after getting quotes from vendor.

Notes: Class 1 - Cost/weight estimates of actual existing Mod I hardware.
Class 2 - Approved for design modification/build; not complete. Estimates are guide values.
Class 3 - New value-engineering concepts not yet approved. Estimates are guide values.

*

Major Part 4 Pieces Each	Material	Material Cost	Labor	Mfg Burdens	Total Cost
Upper Dome (Deep Drawn)	Inconel 718	$39.08	$2.72	$ 6.76	$49.00
Dome L. Part (Invest. Cast)	8625 HR	14.04	0.60	1.88	16.68
Piston Rod (Forging)	Nitroloy SIS 2940-03	3.08	5.04	16.48	24.79
Piston Ring	Rulon LD	7.52 (purchased)	—	—	7.52

Total cost has been included in C1 and C2 cost of piston, piston rod, and seal assembly.

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The semiannual report for the latter half of 1981 indicated that a manufacturing cost analysis of the RESD had been conducted by an outside Detroit-based consulting firm. The results of that analysis showed a need for a program to reduce and eliminate costly concepts in the RESD. Simultaneous to this conclusion, it was recognized that current and projected trends indicated that vehicle sizes were decreasing in weight class well below that of the X-body reference vehicle. As indicated, a study was initiated to investigate the affects on the RESD in downsizing the design to meet small-size vehicle requirements. (The goals for this study have been outlined previously in this report). In order to minimize expenses, these two studies were combined, i.e., manufacturing and downsizing affects. The progress and status of these affects are described below.

Baseline RESD Design Effort

The downsized RESD effort to establish a baseline design has been completed. In summary, the design goals established in the previous semiannual report called for a 45-kW (60-hp) engine capable of installation in a front-wheel-drive, subcompact vehicle. Several concepts studied to meet this goal were:

- a scaled version of the 60-kW (80-hp) U-4 RESD;
- a V-4 double-acting engine;
- a Z-crank-driven, double-acting, four-cylinder engine;
- a three-cylinder in-line engine (α-I and α-II); and,
- a four-cylinder air engine.

Layouts for these engine concepts are shown in Figures 2-1 through 2.6. Early in the engine-layout stage of the design effort, the Z-4 variable-stroke hardware was found to not scale well to the 45-kW (60-hp) application; no further effort was expended on this design. Cost and performance information developed for the remaining concepts are presented in Table 2-1.

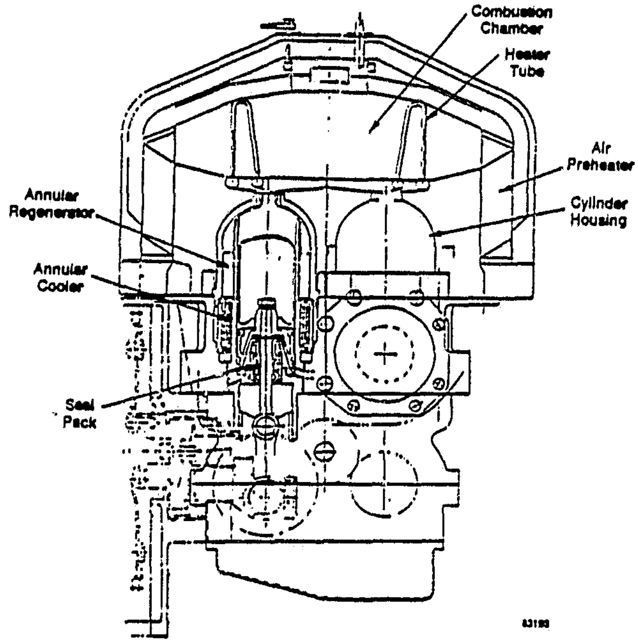

Fig. 2-1 U-4 Downsized RESD

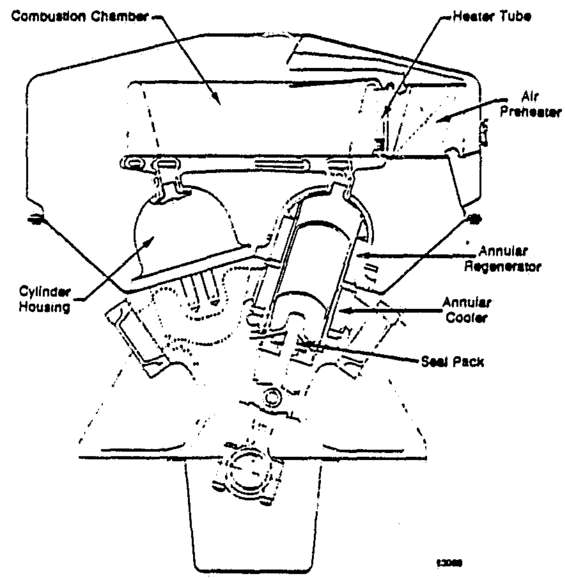

Fig. 2-2 V-4 Downsized RESD

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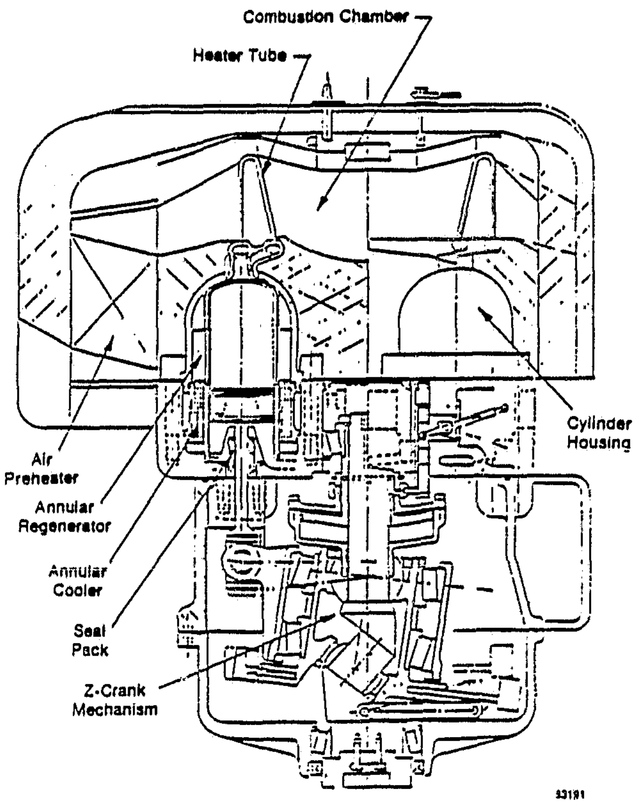

Fig. 2-3 Z-4 Downsized RESD

TABLE 2-1

DOWNSIZED RESD PERFORMANCE AND COST COMPARISON

Parameter	Downsized RESD Concepts			
	U-4	V-4	α-II	Air
Max Power (kW @ rpm)	45 @ 4000	45 @ 4000	45 @ 4000	45 @ 3600
Total Engine Weight (kg)	134	127	146*	143*
Specific Weight (kg/kW)	3.0	2.8	3.2*	3.2*
Peak Effic. Pt. Pressure (MPa)	15	15	15 (15)	(5)
RPM	1500	1500	1500 (1500)	(1200)
Indicated (%)	52.3	52.4	51.3 (53.0)	(55.6)
Brake (%)	42.8	43.5	40.9 (44.3)	(48.1)
Indicated Power (kW)	--	--	(32.8)	(28.9)
Brake Power (kW)	--	--	(27.4)	(25.0)
Cost (Mass-Production Estimate)				
Engine	$ 711	667	578	567
Controls and Auxiliaries	480	480	525	400
Total	$1191	1147	1103	967

*not optimized for minimum weight

Note: Parenthesized performance numbers are from MTI performance code; others are from USAB code; used to cross-verify prediction.

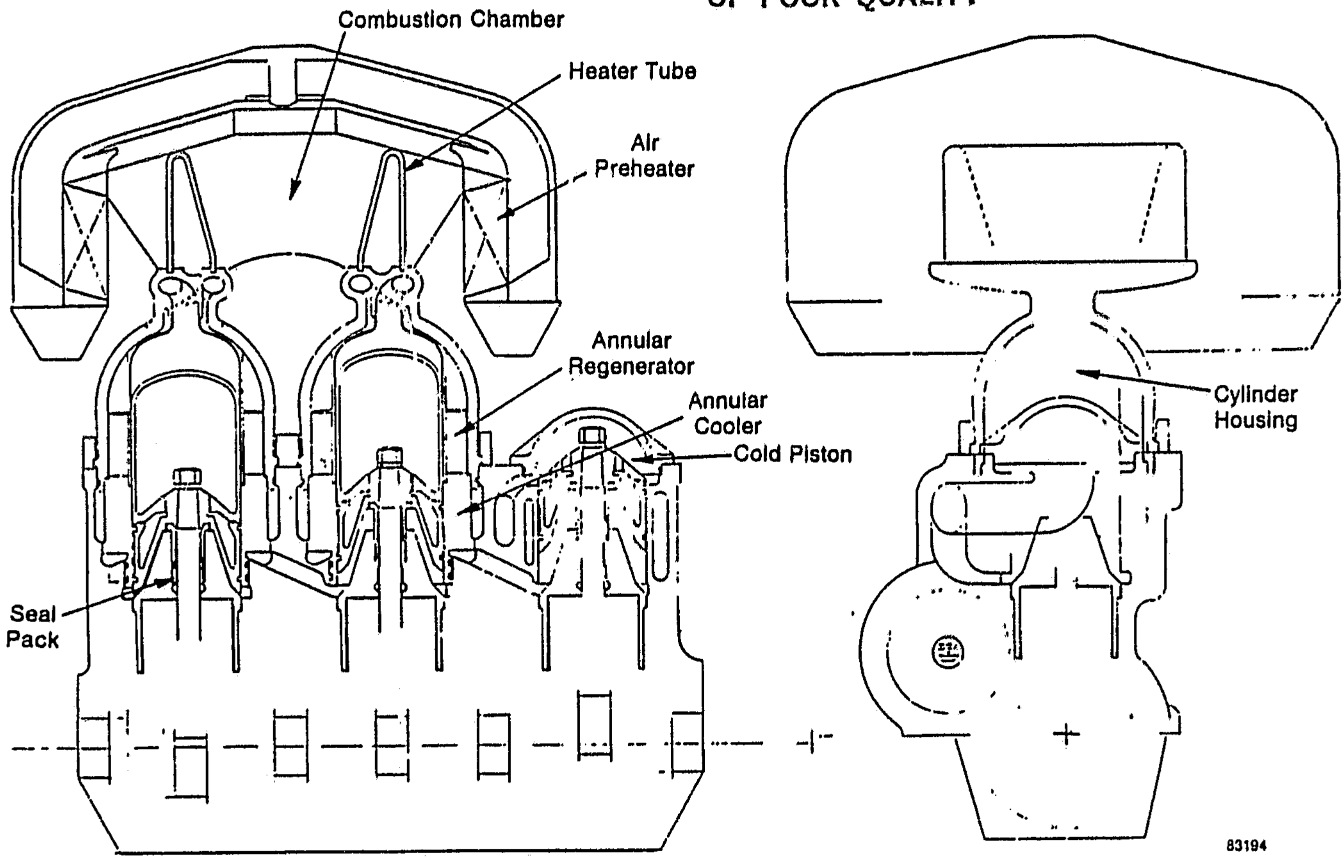

Fig. 2-4 α-I Downsized RESD

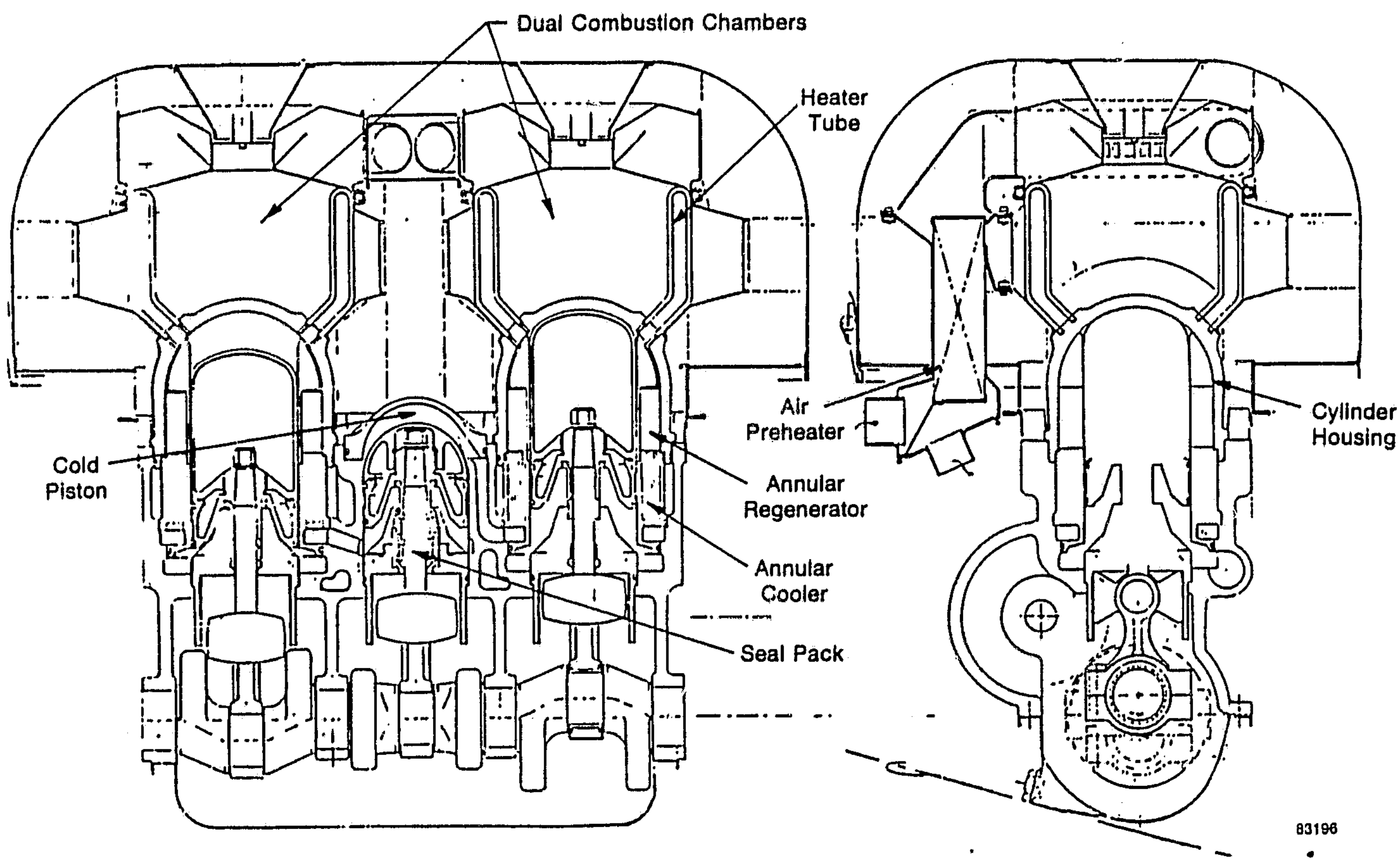

Fig. 2-5 α-II Downsized RESD

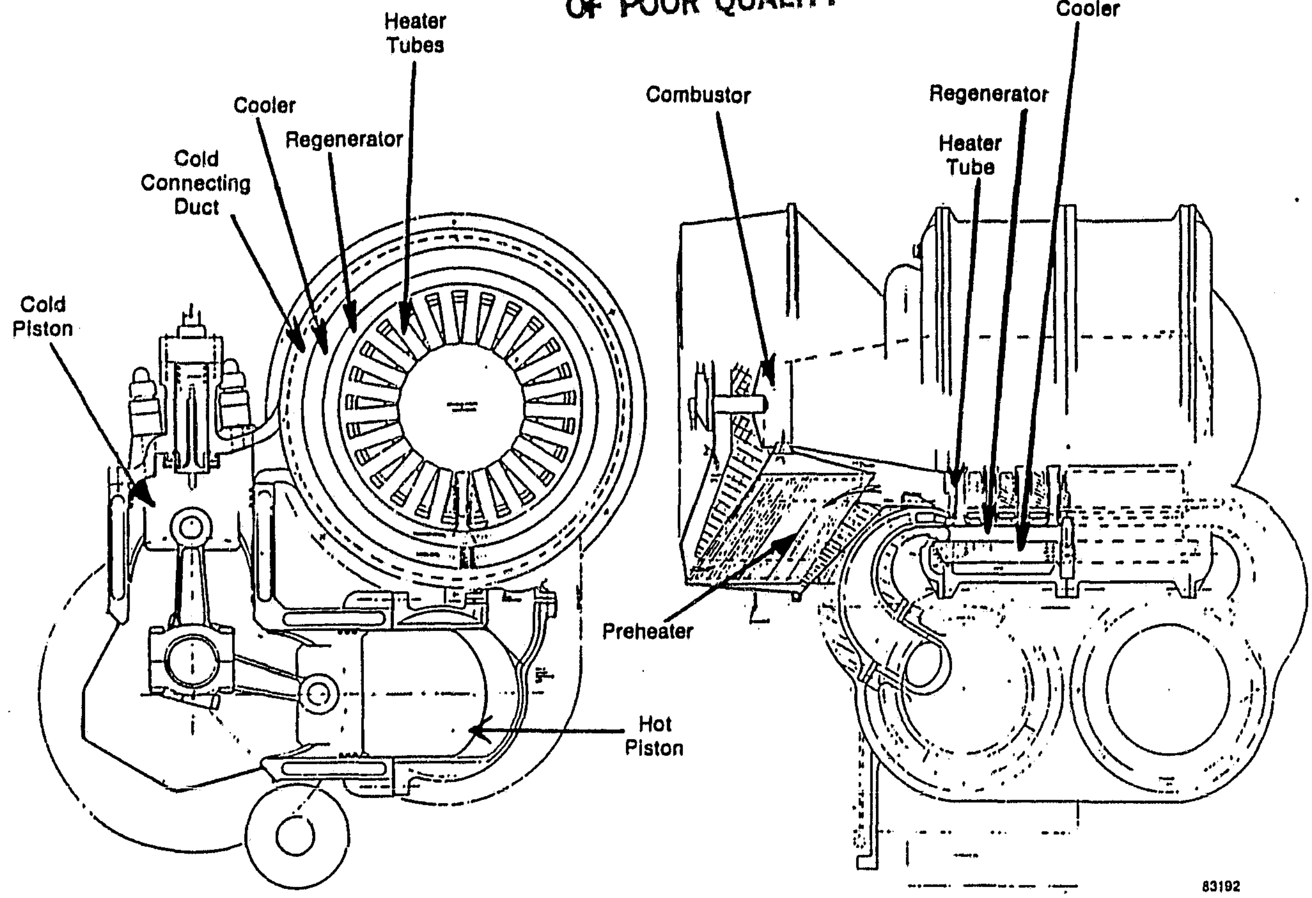

Fig. 2-6 Downsized RESD

In comparing the hydrogen-engine concepts, note that cost and performance for all concepts are almost equal. Since the α-I and α-II engines require a departure from existing four-cycle machine technology with no identifiable benefits, these designs were eliminated. Even though substantial cost/performance improvements are indicated for the air engine, development of several technologies (insulating regenerator matrix, oil contamination prevention, bearing designs) is required for this concept, so it too was eliminated as a primary RESD candidate. As a result, the U-4 and V-4 concepts were selected as the preferred engine designs because both engines can be installed in subcompact vehicles, and can provide significant mileage improvements when compared to spark-ignition engines. A weight summary for the selected designs is shown in Table 2-2.

At the conclusion of this baseline design effort, the following actions were recommended:

- the U-4/V-4 designs be refined to reduce manufacturing costs;
- alternate control techniques such as variable-stroke, bypass control, and delayed-compression power control be further analyzed; and,
- advances in required technology for the air engine be pursued in light of the cost/performance potential indicated in the baseline effort.

TABLE 2-2

WEIGHT COMPARISON OF SELECTED DOWNSIZED RESD CONCEPTS

Component/System	Engine Net Power 45 kW (60 hp)	
Annular-Regenerator	U-4	V-4
Crankcase/Bedplate	8.7	6.1
Crankshaft, Output Shaft, Main Bearing	9.3	7.9
Rear-End Gear	2.7	---
Transmission Intercasing	1.4	---
Oil Pump/Filter Assy	1.0	1.0
Oil Sump	1.5	1.2
Connecting Rod	2.1	2.1
Piston, Rod, Crosshead Assembly	4.3	4.3
Total	31.0	22.6
Crosshead Guide	3.6	3.6
Piston Rod Seal Assy	1.3	1.3
Cylinder Liner	2.2	2.2
Engine Block/Water Jacket	8.6	8.8
Total	15.7	15.9
Heater Head Assy (incl. part. wall & spacer)	11.5	11.8
Housing Attachment Ring and Studs	6.8	6.8
Regenerator	1.7	1.7
Gas Cooler	2.6	2.6
External Heat System	9.9	10.8
Total	32.5	33.7
Gas Compressor, Control Valve System Gas Bottle/Other Items	11.0	11.0
Gas Compressor Drive	0.3	0.3
Water Pump	3.1	3.1
Cooling Fan with Clutch	3.5	3.5
Variable-Drive	3.0	3.0
Combustion Air Blower with Motor	9.0	9.0
Starter	7.0	7.0
A/F Control System (incl. Fuel Pump)	2.0	2.0
Microcomputer	1.4	1.4
Bolts/Tubes/Brackets/ Belts, etc.	7.0	7.0
Total	47.3	47.3
Total Dry Weight	126.5	119.5
Incl. Alternator and P/S Pump - 7.7 kW (10 hp)	134.2	127.2

Value-Engineered Design Effort

A value-engineering design effort was initiated for the V-4 design, emphasizing the need for a manufacturing cost reduction. A baseline cost estimate was established for the V-4 concept, and all proposed design changes were evaluated against this baseline.

Initially, material substitutions that would significantly reduce the cost of specific components, and that could be incorporated without any design changes, were identified:

- heater head tubes - change from Inconel 625 to CG-27;
- gas cooler tubes - change from stainless steel to phosphate-coated carbon steel; and,
- preheater matrix - change from Sandvik 253MA to Armco 12SR or 18SR.

Following this initial study, several designs such as a one-piece, equal-angle, cast-iron block; perforated plate-gas coolers; a one-piece piston dome/rod assembly, and the elimination of the balance shaft were conceived and incorporated in the V-4 concept, resulting in a weight reduction of the total engine system from 119.5 kg (263.5 lbs) to 110.3 kg (243.2 lbs).

A cost comparison for the V-4 baseline design, the value-engineered design, and materials' change, is shown in Table 2-3. As noted, a significant cost reduction (37%) is achievable with this design. The value-engineered V-4 design is shown in Figure 2-7.

TABLE 2-3

RESD V-4 MANUFACTURING COST COMPARISONS

Parts* Descrip.	Baseline Design Feb. '82	Materials Substitution	Value-Engineered Design
Cylinder Housing	$ 50.92	$ 50.92	$ 50.92
Heater Tubes	118.48	94.07	94.07
Regener. Matrix	67.00	67.00	67.00
Gas Cooler Assembly	290.45	87.60	65.93
Preheater Matrix	147.47	122.84	120.09
Combustion Chamber Assembly	41.44	41.44	41.44
Piston & Piston Rod Assembly	75.33	75.33	53.69
Water Jacket*	24.62	24.62	
Crosshead Guides*	27.58	27.58	57.54
Crankcase*	30.26	30.26	
Oil Sump	1.95	1.95	2.28
Main Bearings	3.00	3.00	3.00
Crankshaft	27.89	27.89	27.89
Balance Shaft	13.94	13.94	Deleted
Spur Gears	9.55	9.55	Deleted
Total	929.88	677.99	583.85

*one-piece casting in Value-Engineered design

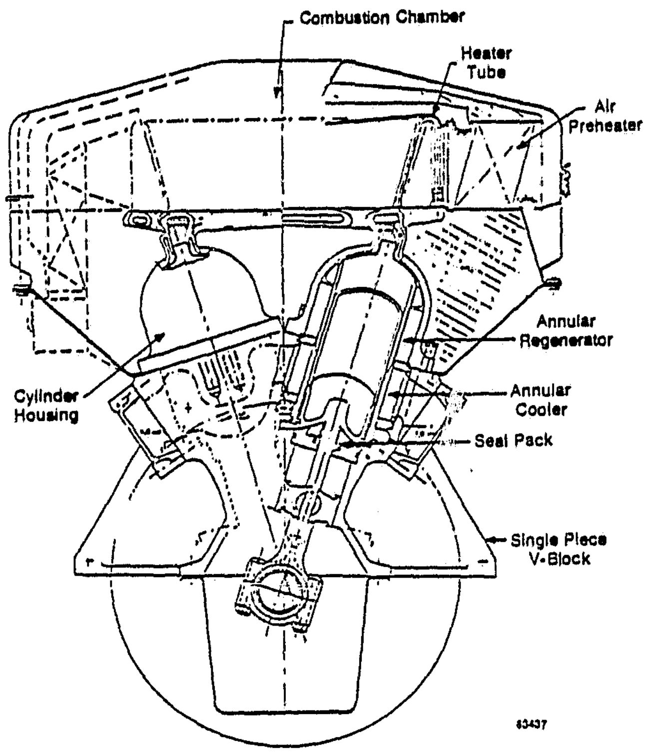

Fig. 2-7 Value-Engineered 45-kW (60-hp) V-4 RESD

III. COMPONENT DEVELOPMENT

Component development activity is organized on an engine subsystem basis, with developmental emphasis on the following: 1) External Heat System (combustor, fuel nozzle, igniter, preheater); 2) Hot Engine System (heater head/regenerators); 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).

Mod I component performance characterization, and initiation of Mod I-A component design/testing was the focal point of development activity during this report period. This has been accomplished through substantial technology transfer and acceleration of the utilization of test facilities in the United States.

During the latter half of 1982, primary emphasis will be placed on the completion of component development for the Mod I-A engine through rig characterization and Mod I engine testing, as guided by the RESD. Each component has either a performance, cost-reduction, or reliability goal for 1982. Components will be developed to the point where they will be suitable for proof-of-concept testing on a Mod I or Mod I-A engine.

External Heat System (EHS)

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

Development activity during the first half of 1982 focused on the evaluation of nonair-atomized Mod I-A fuel nozzles, the design of an improved CGR combustor for the Mod I-A, Performance Rig baseline tests, an analysis of Mod I engine EGR emissions, and the design of a ceramic Mod I-A preheater.

Primary objectives during the latter half of 1982 will focus on the development of a reduced size/cost EHS through rig testing, and verification of EHS performance in a Mod I-A engine. A metallic preheater will be used as an interim design until a ceramic matrix can be developed late in 1983.

MOD I-A FUEL NOZZLE EVALUATION

The Mod I-A fuel nozzle must be capable of operation over a turndown range of 18, allow the engine to idle at 0.25 g/s fuel flow, not use an external source of atomizing air, and demonstrate a beneficial affect on emissions and temperature profile. Based on a spray-quality evaluation, two nozzles (a piloted air-blast and a dual-orifice) were selected for further development in the Free-Burning and Combustion Performance Rigs. The Free-Burning Rig test setup, utilizing a CGR combustor modified to accept the fuel nozzles and external igniter, is shown in Figure 3-1. Testing was conducted to determine and compare the nozzles' ignition/blowout limits to those of the Bill-of-Materials (BOM) Mod I nozzle. Typical results are illustrated in Figures 3-2 and 3-3.

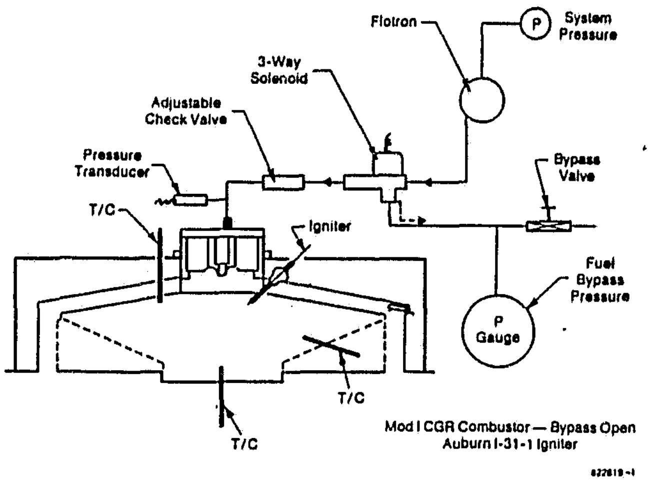

Fig. 3-1 Free-Burning Rig Configuration Ignition Test

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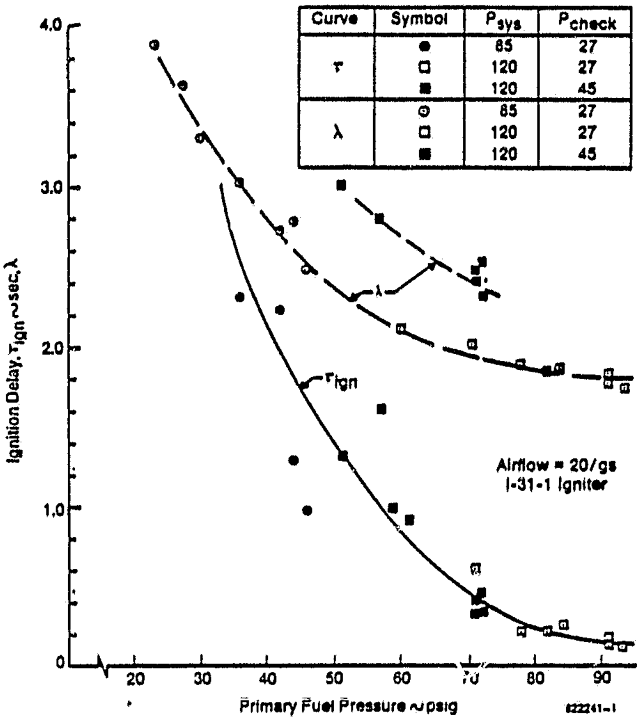

Fig. 3-2 Ignition Characteristics of Delavan Dual-Orifice Nozzle #35814

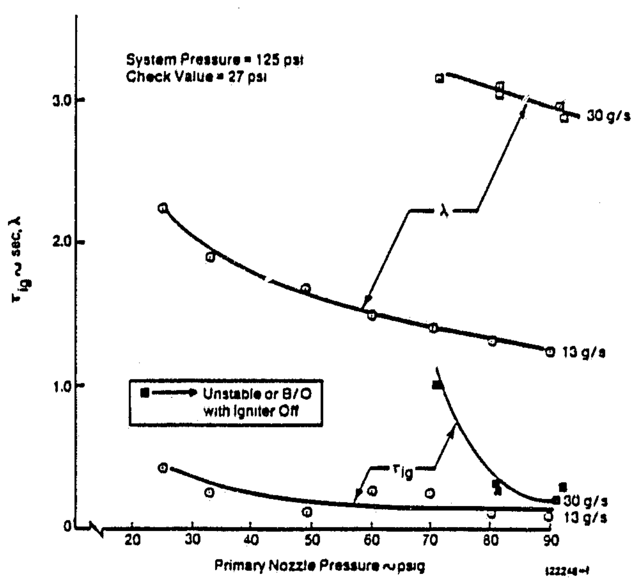

Fig. 3-3 Ignition Characteristics of Delavan Air-Blast Nozzle #37112

Both the air-blast and dual-orifice nozzles demonstrated ignition delay times less than 0.2 seconds, and a 33% improvement over the Mod I air-atomized nozzle. Blowout performance was also satisfactory, occurring at fuel flows ranging from 0.1-0.25 g/s. One of the nozzles will be selected for the Mod I-A during the second half of 1982 based on Performance Rig testing. Development of additional air-blast and dual-orifice nozzles, as well as an ultrasonic atomizer, will continue if cost and performance improvements are demonstrated in the Rig.

MOD I-A CGR COMBUSTOR

Goals for the CGR combustor, to ensure CVS cycle emissions compliance and maximize the EHS (e.g., increase engine efficiency while allowing the engine to operate stably at low idle conditions), will be achieved through recirculation of exhaust products to minimize emissions, a well-mixed reaction zone to ensure temperature profile uniformity, rapid ignition to minimize cold-start penalty, and structural integrity to withstand thermal transients.

Two methods of controlling emissions have been used - EGR and CGR. CGR is desirable because it eliminates the external plumbing, valving, and control requirements of EGR, extends preheater life, and minimizes blower power requirements. The Mod I straight guide vane CGR combustor and its design variations are unacceptable because of temperature profile nonuniformity and thermal transient-induced parts' life limitations.

A tubular and radial CGR combustor were designed during this report period, and are currently being developed for the Mod I-A. The radial combustor does not contain discrete CGR passages; instead, it uses an annular, converging/diverging ejector. The tubular combustor (Figure 3-4) uses discrete ejectors; however, the CGR channel cross-sectional area is constant instead of diverging. To prove the feasibility of the radial CGR concept, 1/10-segment test pieces were fabricated in three geometric variations and tested for recirculation capability (Figure 3-5). Based on these cold-flow results,

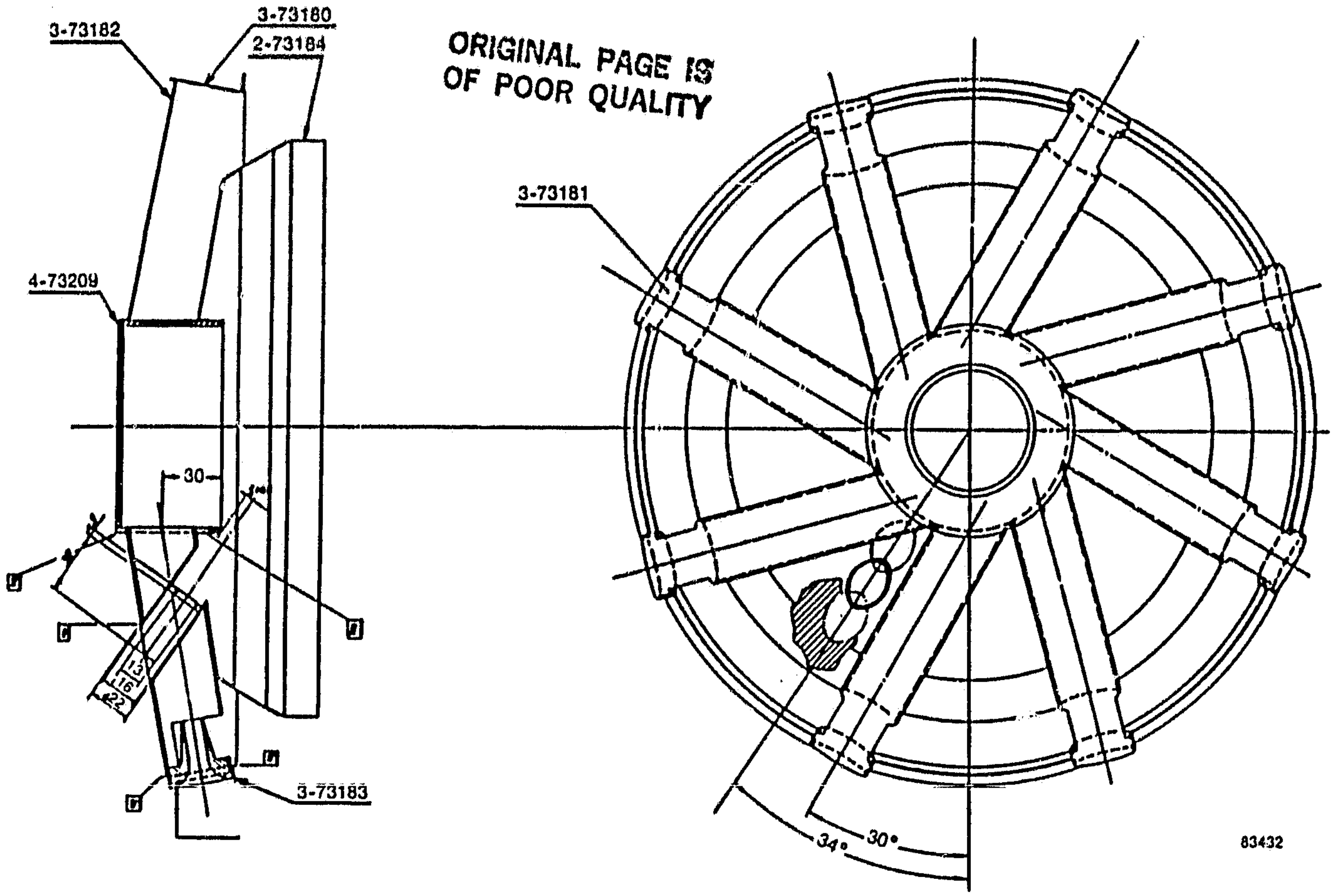

Fig. 3-4 USAB Tubular CGR Combustor (Early Version)

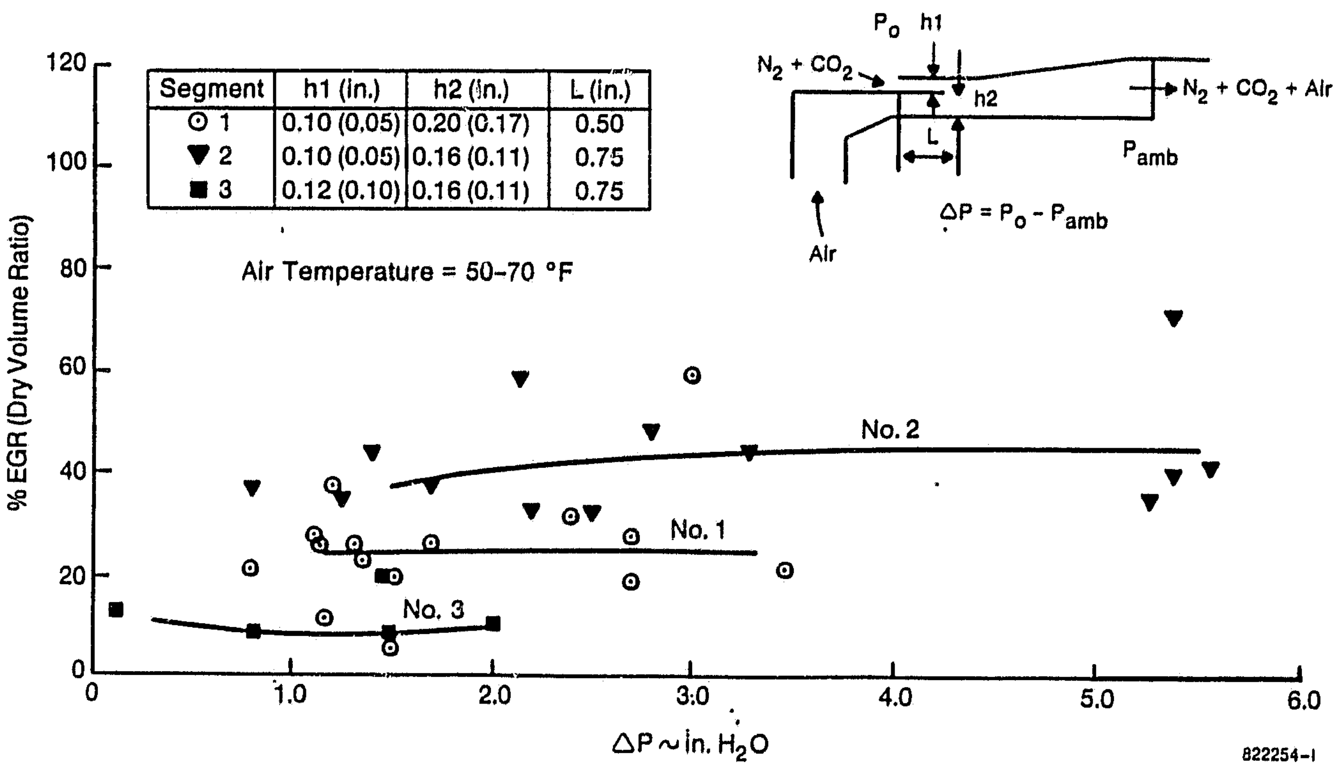

Fig. 3-5 1/10-Segment Mod I-A CGR Combustor Test Results three full-size combustors will be fabricated and tested in the Combustion Rigs with the Mod I-A fuel nozzle.

Rig/engine testing of the tubular CGR combustor was conducted with both 12- and 13-mm diameter ejectors. CVS cycle emissions are projected (Table 3-1) based on steady-state Mod I engine data. Blower power requirements at maximum airflow and average spread in working gas temperature are given in Table 3-2, which shows that the the 12-mm ejector tubular CGR combustor requires about the same blower power as the current BOM CGR combustor, but significantly reduces the variation in working gas temperature. Finally, rig start-up times, defined as fuel-on to 600°C tube temperature without cooling, are about the same as the EGR combustor without gas recirculation (e.g., 25 seconds). The results of these tests indicate favorable progress.

TABLE 3-1

TUBULAR CGR COMBUSTOR
ESTIMATED EMISSIONS (g/mi)

Combustor	Drawing	NO_x	CO	HC
12-mm (cell)	1-S-2079	0.29	0.46	0.03
13-mm (cell)	1-S-2079	0.34	1.55	0.02
13-mm (rig)	1-S-2079	0.30	0.39	0.05
BOM CGR (cell)	1-17831	0.23	0.52	0.02
60% EGR (cell)		0.43	3.40	0.41
CVS Cycle Requirement		0.40	3.40	0.41

PERFORMANCE RIG BASELINE TESTS

The Performance Rig (shown in Figure 3-6), crucial to fuel nozzle, combustor, and preheater development, contains all the critical components necessary to evaluate the EHS (shown in Figure 3-7), including a heater head/heat extraction system that duplicates the engine geometry and heat transfer capability. Through the use of Performance Rig emissions, heater head temperature profile, and preheater performance, EHS efficiency can be determined.

Fabrication of the Performance Rig has been completed, and the operational capability of all systems has been verified. The heater head (assembled entirely in the United States), which has the same tube geometry and fins as the Mod I engine, has also been successfully fabricated.

The heat extraction system has been assembled and made operational, and automation of data acquisition has begun. Shakedown and Mod I CGR baseline testing were initiated using both a "dummy" and a Mod I preheater. Numerous problems with the operation, control, and measurement of this complicated system were identified and corrected in preparation for Mod I-A EHS development during the second half of 1982.

TABLE 3-2

COMPARISON OF PREDICTED AOP MOD I PERFORMANCE FOR
SCREEN, OPTIMUM METEX REGENERATOR, AND MULLITE-FOAM REGENERATOR

		Urban AOP			Highway AOP			Combined AOP		
		Scr	Met	Mul	Scr	Met	Mul	Scr	Met	Mul
Speed	(rpm)	1350	1350	1350	2150	2150	2150	1620	1620	1620
P_{mean}	(MPa)	5.64	5.67	6.56	7.11	7.03	7.94	5.62	5.58	6.12
Ind. Power	(kW)	12.5	12.5	12.5	24.1	24.1	24.1	14.7	14.7	14.7
	(hp)	16.7	16.7	16.7	32.3	32.3	32.3	19.7	19.7	19.7
Ind. Efficiency	(%)	45.9	43.9	35.3	47.6	44.3	33.4	46.4	44.0	35.0
Regen. Efficiency	(%)	99.7	97.2	91.9	99.6	96.5	89.5	99.7	97.1	91.7

Notes: 1. Screen - Sintered Wire, 0.002-inch Diameter
2. Metex - Triple-Strand, 0.0035-inch Diameter

MOD I ENGINE EGR EMISSIONS

In order to comply with ASE Program emissions goals for the CVS cycle, both EGR and CGR combustion systems were designed and tested in the Mod I engine. EGR was selected for the Mod I-A based on superior combustor life/temperature profile. The impact of EGR on emissions and engine efficiency was then evaluated through testing on Mod I engines No. 2 and 3.

The first test series was conducted on engine No. 2. Emissions were measured for various levels of constant EGR (defined as the moles of recirculating exhaust gas divided by the moles of inlet air on a dry basis). Results for NO_x emissions are illustrated in Figure 3-8. The affect of EGR on NO_x is seen to be exponential, as expected. Testing was confined to the lower power regions typical of the CVS cycle, e.g., more than 96% of that cycle is run at fuel flows equal to or less than 2 g/s. CO and HC emissions were low and relatively unaffected by EGR, leading to the conclusion that compliance to the 0.4 g/mi ASE Program goal would require 60-70% EGR.

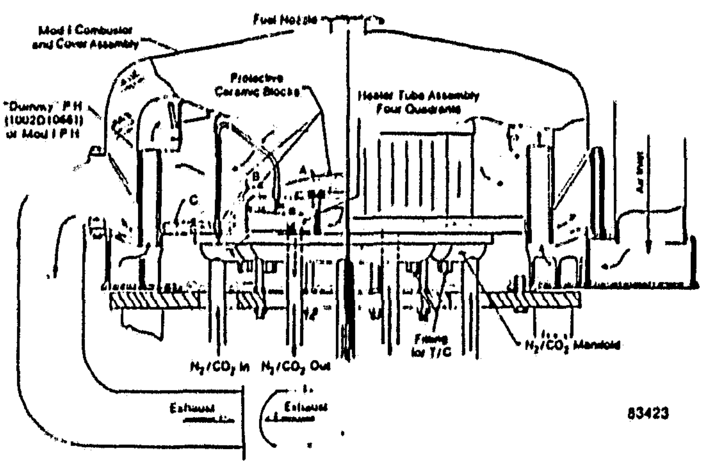

Fig. 3-6 Performance Rig

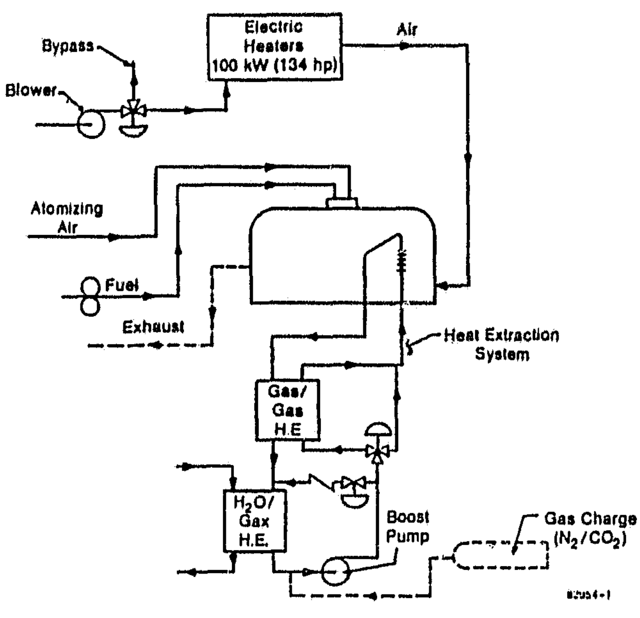

Fig. 3-7 Schematic Showing Fuel, Air, and Heat Extraction System

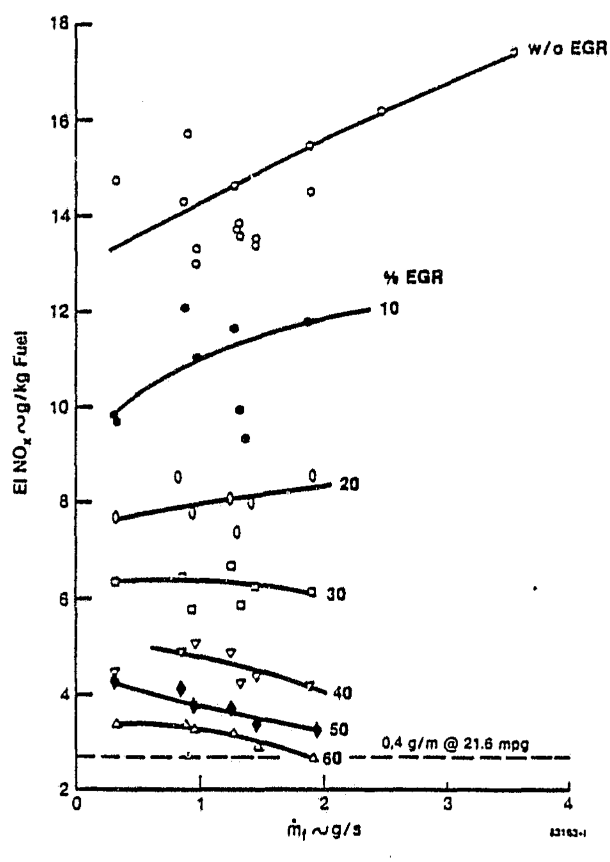

Fig. 3-8 Mod I NO_x Emissions versus %EGR

The second test series was conducted on engine No. 3 to determine the affect of EGR on engine efficiency, and the impact of constant metering area on EGR and emissions. The simplest EGR system, from a controls and hardware point of view, consists of a metering orifice with an on/off valve; however, with such a system, EGR will vary with engine load. Initial testing was conducted with no restriction in the EGR line (EGR ranged from 40 to 70%).

NO_x was well under the 0.4 g/mi requirement, but engine efficiency was adversely affected. A criteria was established stating that engine efficiency could not decrease more than 0.4% compared to non-EGR. The metering valve was then adjusted to satisfy this criteria (results shown in Figures 3-9 and 3-10). The two λ (defined as $A/F \div (A/F)_{stoich}$) schedules had no affect on EGR, NO_x or HC; however, a dramatic affect on CO was observed. The EGR schedule illustrated is consistent with CVS cycle NO_x emissions of 1.0 g/mi. CO and HC are well under the CVS requirements if λ > 1.2. A summary of the affect of EGR on NO_x is shown in Figure 3-11.

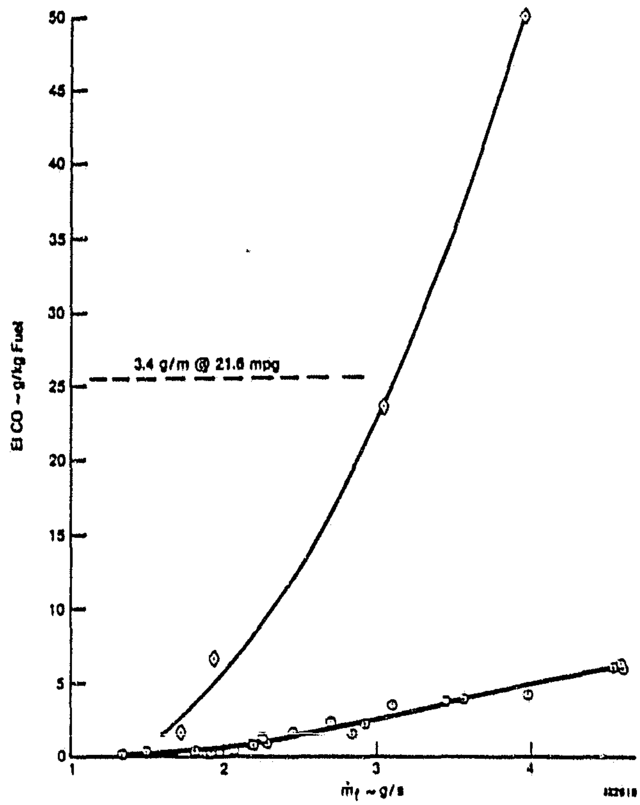

Fig. 3-10 Mod I Engine No. 3 CO Emissions with EGR June 9-18, 1982

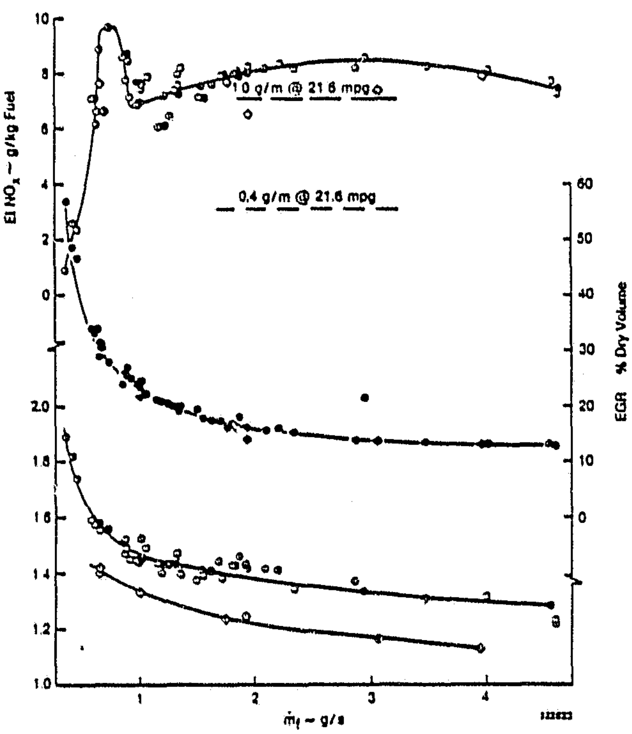

Fig. 3-9 Mod I Engine No. 3 NO_x Emissions with EGR June 9-18, 1982

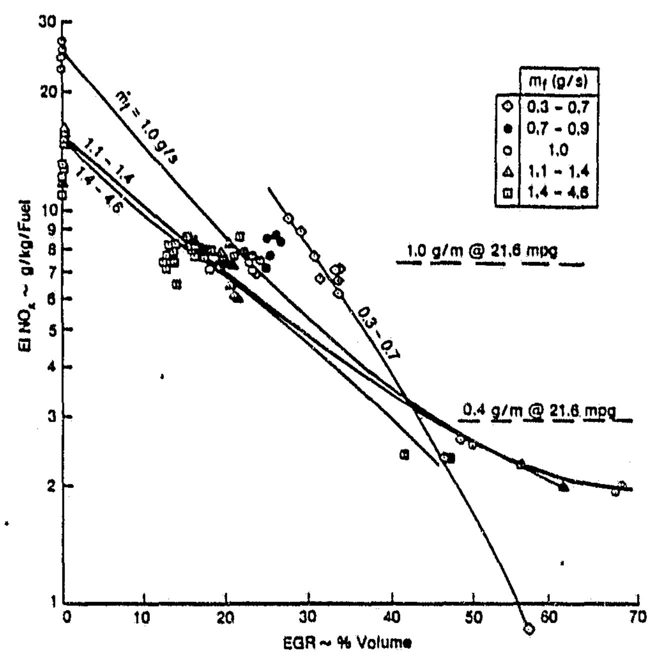

Fig. 3-11 Mod I Engine No. 3 NO_x Emissions with EGR June 4-18, 1982

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During this semiannual report period, the emissions measurement capability of the gas analyzer was upgraded as follows:

- a paramagnetic oxygen analyzer was added to provide redundancy to the determination of air/fuel ratio, and simultaneous determination of EGR/exhaust emissions;
- automation of data acquisition using the HP-1000 was initiated; and,
- a smoke measurement system was designed and procurement initiated.

MOD I-A CERAMIC PREHEATER

The metallic preheater represents one of the most expensive components of the Mod I engine due to the inherent expense of fabricating 1100-1200 individual metallic platelets into a matrix. Ceramic material is relatively inexpensive to fabricate. In addition to cost considerations, ceramic preheater performance and thermal mass must be consistent with high EHS efficiency, minimal cold-start penalty, and packageability. These requirements imply extremely high surface/volume ratio (300 ft^2/ft^3), thin walls, and low mass times specific heat.

Contact with three ceramic vendors indicated that a preheater fabricated from ceramic blocks was feasible. As a result, a conceptual design of a Mod I-A ceramic preheater with ten blocks has been completed. The ceramic matrix can be either a counterflow-Z configuration or a crossflow concept (shown in Figures 3-12 and 3-13, respectively).

The Z-flow concept appears more favorable at this time based on wall thickness and effectiveness. A ceramic counterflow block, developed for a gas turbine, will be evaluated for performance and durability in the Preheater Rig (Figure 3-14) during the latter half of 1982.

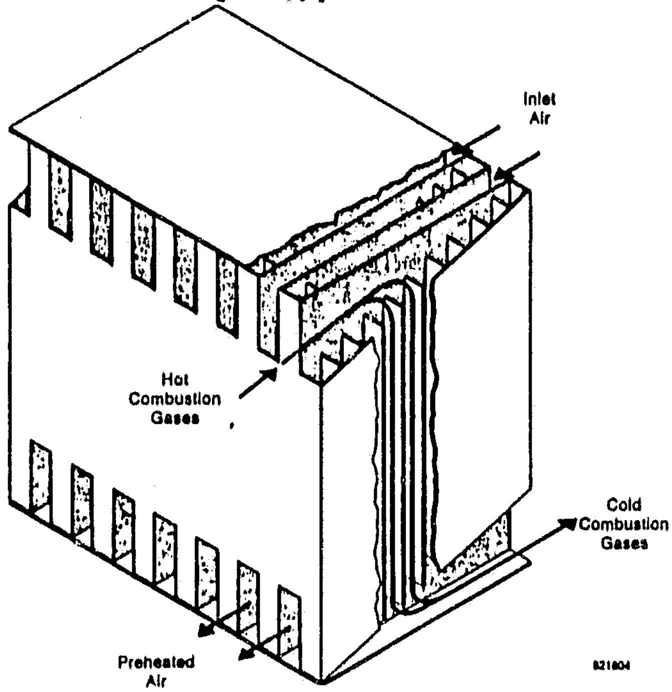

Fig. 3-12 Ceramic Air Preheater: Counterflow Concept

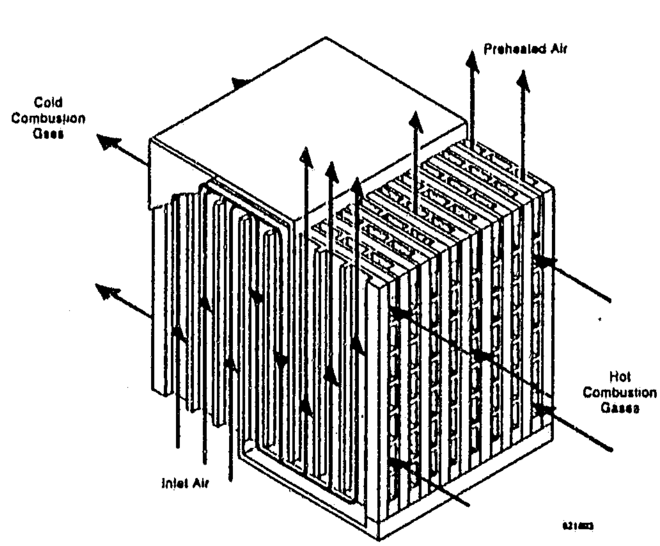

Fig. 3-13 Ceramic Air Preheater (Mu H-Pass): Crossflow Concept

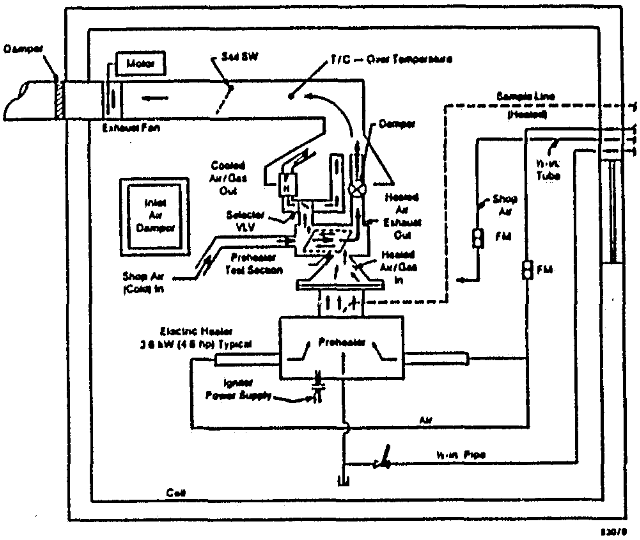

Fig. 3-14 Schematic of Preheater Rig

In support of the Mod I-A preheater, a computer code for EHS thermal analysis was completed based on empirical engine data. This code has been used to predict the affect of atomizing air on Mod I-A efficiency (see Figure 3-15).

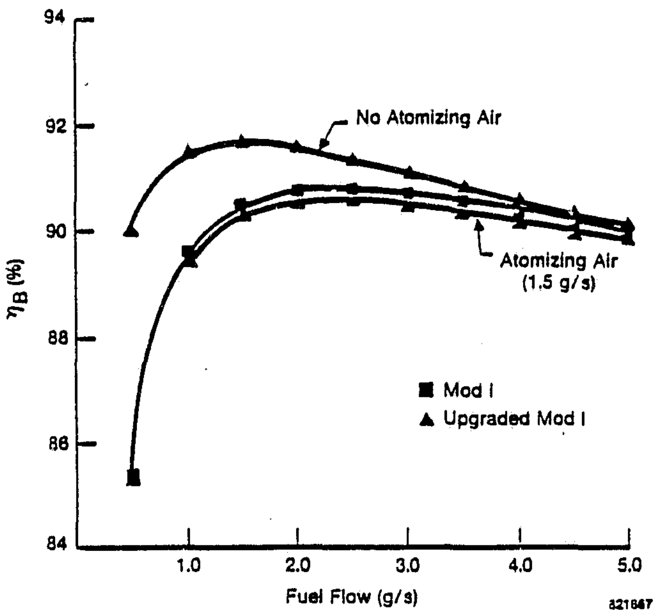

Fig. 3-15 EHS Efficiency versus Fuel Flow - Tubular Combustor (No G.R.)

Hot Engine System Development

The primary goal of this task is to provide a low-cost regenerator design that has a mileage performance comparable to the current design. To date, the requirement for high regenerator efficiency (typically ~98%) has been reflected in the use of high-cost matrix material such as sintered woven, high-count stainless mesh.

Activity during this semiannual report period was aimed at Heat Transfer Rig testing of alternate regenerator matrix materials (costs 70% or less than sintered screen), and the selection of an optimum material and design for the Mod I-A.

Regenerators will be fabricated (with the selected material) and tested in a Mod I engine during the second half of 1982, and an evaluation of heater head performance in both the engine and Combustion Performance Rig will begin.

REGENERATOR RIG TESTS

The regenerator stores the thermal energy of the working gas as it passes from the hot expansion space to the cold compression space. Both its heat transfer coefficient (Nusselt Number) and pressure drop (friction factor) must be determined accurately to evaluate its performance. A unique Heat Transfer Rig (shown schematically in Figure 3-16) is used for these measurements. In this rig, air is blown through a sample test section (see Figure 3-17) in such a way that a fast-acting gate valve can achieve a step rise in air temperature.

A typical trace of the resultant matrix test section inlet/outlet temperatures is shown in Figure 3-18. The Nusselt Number of the sample matrix material can be determined from analysis of the outlet temperature trace.

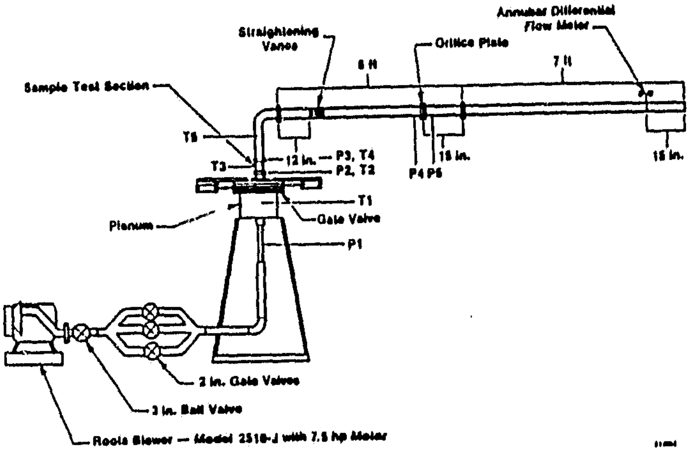

Fig. 3-16 Regenerator Matrix Test Rig Schematic

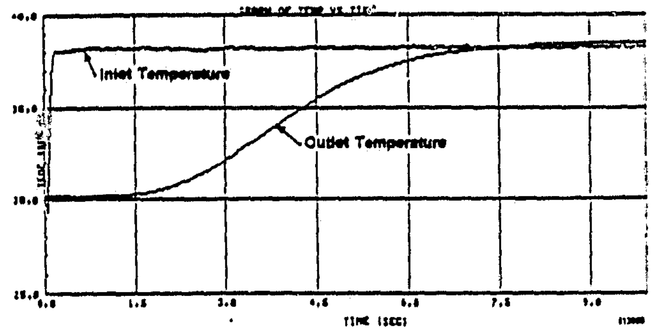

Fig. 3-18 Typical Temperature History

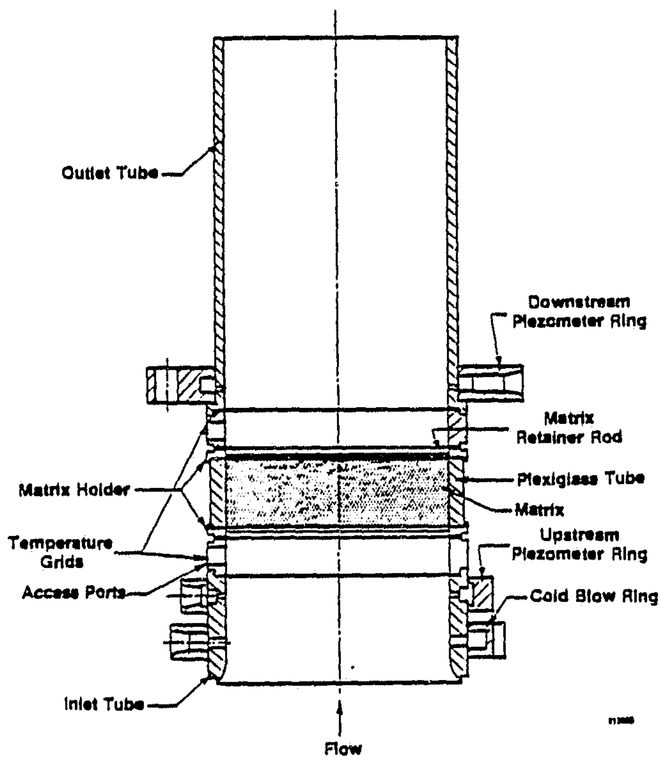

Fig. 3-17 Sample Test Section

METEX REGENERATOR MATRIX EVALUATION

Previous testing of Metex knitted wire (0.006-inch diameter) indicated that a finer wire would improve performance without significantly increasing cost. During the first half of 1982, Metex test sections (0.0035-inch diameter wire) were fabricated and tested in the Regenerator Rig. Results indicated an improvement in performance compared to the previously evaluated, larger wire diameter Metex, and that both heat transfer and friction increases with porosity (Figures 3-19 and 3-20).

The test results were entered in MTI's First-Order Engine Code and used to predict the affect on CVS cycle Average Operating Point (AOP) (see Table 3-2). Compared to the Mod I sintered screen, regenerator and engine efficiency decreased 5 and 3%, respectively, while cost decreased by about 1/4. The First-Order Code also indicates the optimal porosity to be 70% (Figure 3-21).

Two sets of Metex regenerators, 55 and 70% porosity, have been ordered for the Mod I engine. Testing scheduled for the latter part of 1982 will verify the applicability of the rig data and First-Order Code to engine performance.

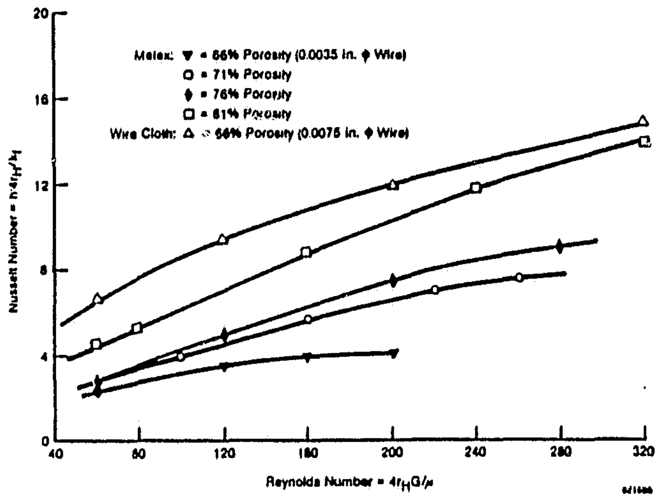

Fig. 3-19 Nusselt Number versus Reynolds Number

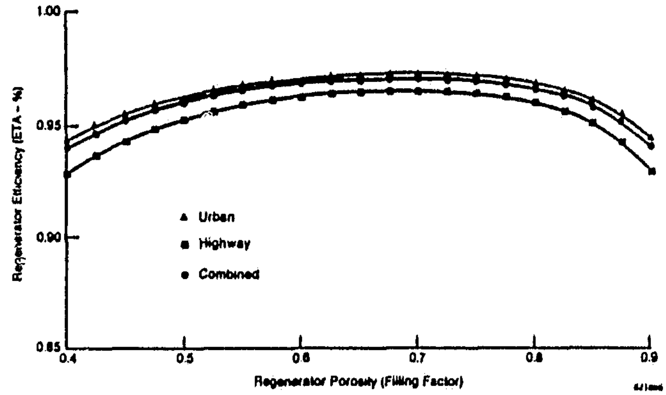

Fig. 3-21 Optimum Metex Regenerator for Mod I

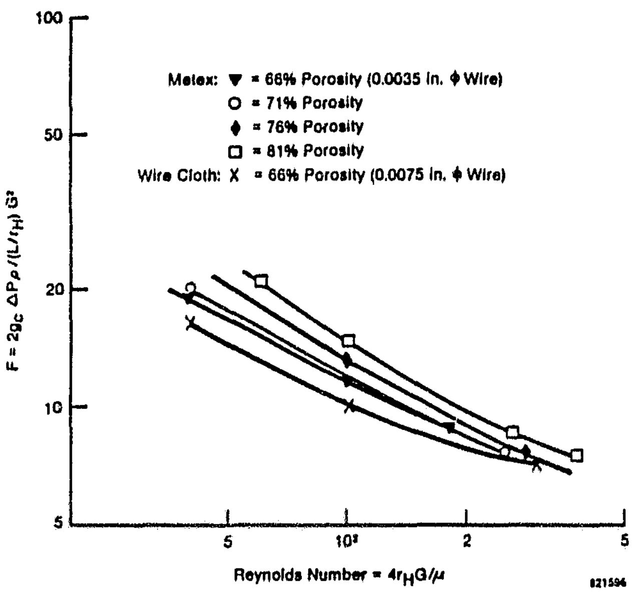

Fig. 3-20 Friction Factor versus Reynolds Number

CERAMIC REGENERATOR MATRIX TESTING

Continued analysis of materials was performed to identify regenerator matrix candidates that would be cost-effective and not cause a performance degradation. Two such materials have been identified - Mullite-foam ceramic and silicon carbide Duocel. Testing of Mullite (Figures 3-22 and 3-23) in the Regenerator Rig has been completed; the First-Order Code predicted effect is shown in Table 3-2.

The influence of regenerator NTU (number of transfer units) on effectiveness (Figure 3-24), illustrates that Mullite is not an acceptable material. Silicon carbide Duocel will be evaluated later in 1982. The major technical question concerning ceramic regenerators is their ability to survive in a cyclic-flow environment. A Durability Rig will be fabricated using a diesel-engine block to cycle pressurized nitrogen through the test section before committing to engine hardware.

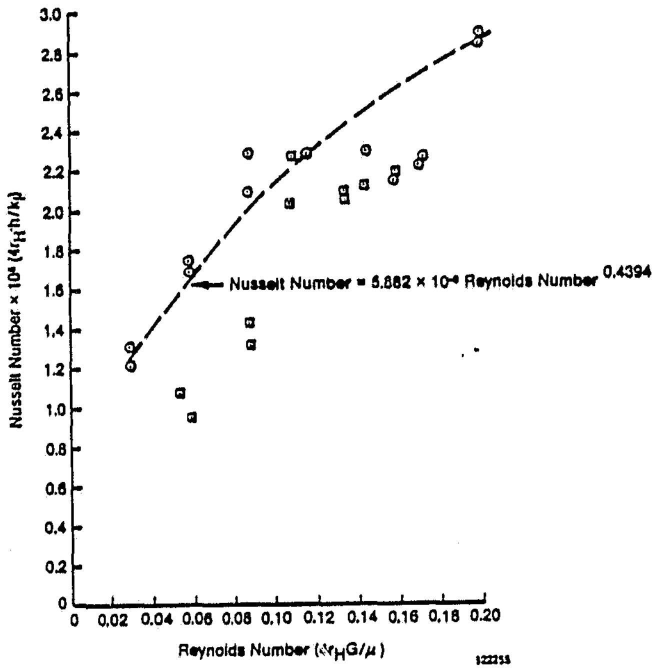

Fig. 3-22 Mod I-A Regenerator Heat Transfer Results Mullite Foam-Ceramic Test (60 Pores/Inch)

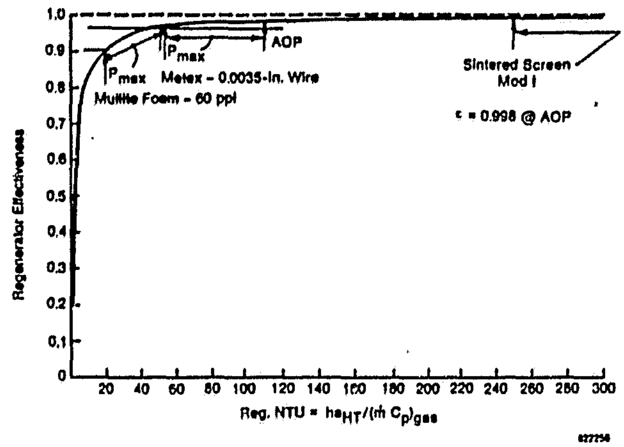

Fig. 3-24 Regenerator Effectiveness versus NTU

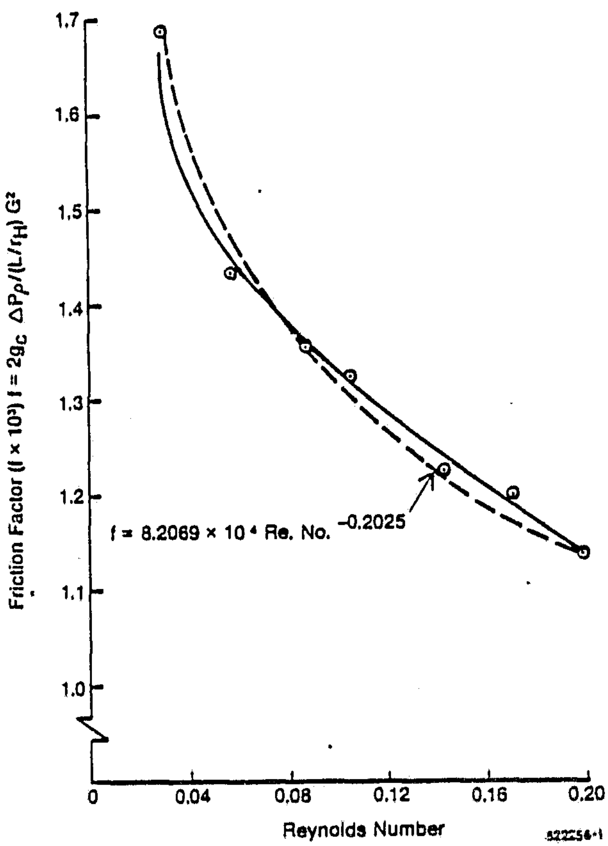

Fig. 3-23 Mod I-A Regenerator Friction Factor Results Mullite Foam-Ceramic Test (60 Pores/Inch)

Materials 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 and pressure environment, as well as the presence of both high and low-cycle cyclic stresses, have contributed to the difficulty of meeting this objective.

Development efforts during this reporting period focused on Phase II testing of two specimens of alternate-casting materials and five alternate-tube materials; reducing the strategic-element content of the Mod I-A engine; selection of material for the Mod I-A heater head; and engine-testing two heater heads from these and other materials.

During the last half of 1982, testing of the alternate-material heater heads will be completed on an engine in a simulated duty cycle, and fatigue tests of the casting materials will be performed in a hydrogen environment.

DESIGN PROPERTIES TESTING

Testing is underway to determine the statistical scatter of the initial fatigue data, the affects of mean stress on fatigue life, the affects of hold time (creep-fatigue interaction), and the affects of a hydrogen atmosphere on fatigue life. In addition, tensile and fatigue data of welded XF-818 are being generated to further aid in the design and repair processes.

Long-time (up to 5000 hours) creep testing of the alternate heater tube materials is being conducted in argon at 750 and 850°C. The data collected to date indicates that alloy CG-27 is the best candidate, ahead of Inconel 625 and Sandvik 12RN72. The results to date are shown in Figures 3-25, 26, and 27. Sanicro 32 and 31H appear to have a creep strength less than desirable for long engine life. Figure 3-28 compares all five alternate tube alloys and N155. Lines are drawn through the points for each tube alloy to give a relative ranking.

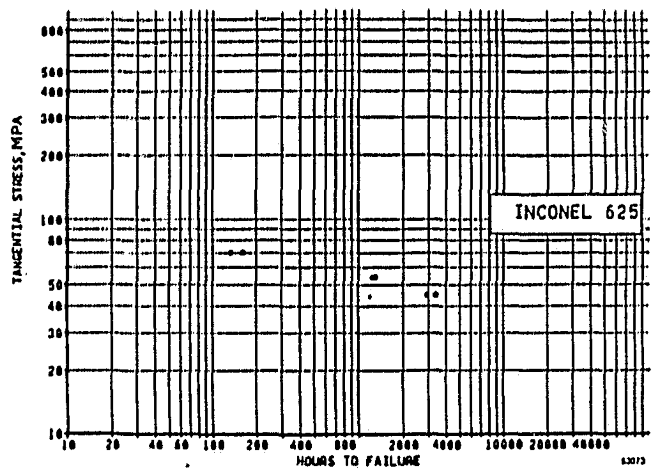

Fig. 3-26 Creep Rupture Testing at Sandvik - 850°C

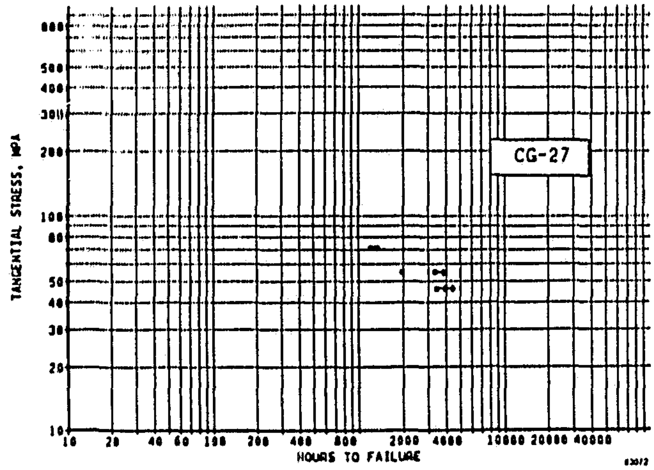

Fig. 3-25 Creep Rupture Testing at Sandvik - 850°C

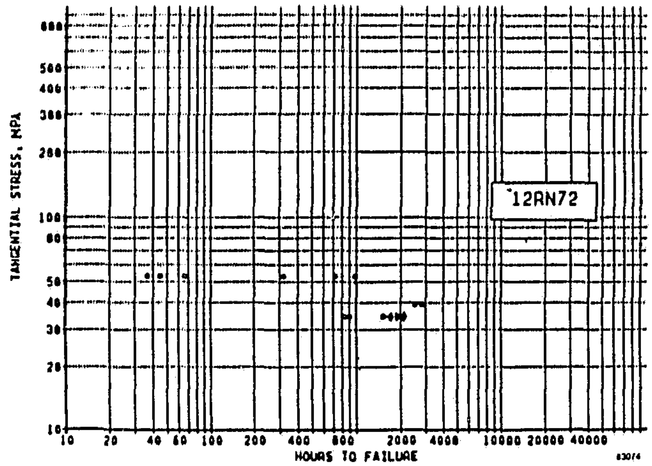

Fig. 3-27 Creep Rupture Testing at Sandvik - 850°C

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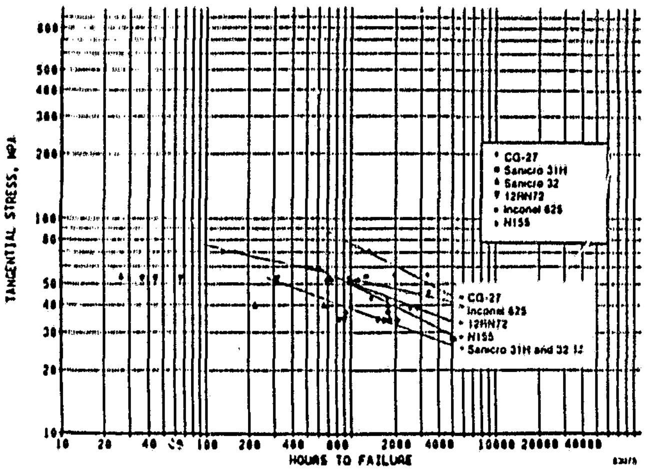

Fig. 3-28 Creep Rupture Testing at Sandvik - 850°C

STRATEGIC MATERIALS

Strategic materials have been defined[1] as chrome, cobalt, columbium, and tantulum. Current efforts have been aimed at eliminating or reducing the amounts of these elements in the various engine systems, especially the heater heads/heater tubes. Investigations of other strategic material reductions revealed the possibility of using CG-27 (13% chrome) in place of Inconel 718 (19% chrome) in the piston dome, and also substituting a lower chrome-containing ferritic stainless steel (18SR or 408 Cb, 18, and 12% chrome, respectively) for 310 SS (25% chrome) or Sandvik 253MA (21% chrome). Serious consideration is being given to using silicon carbide foam in place of 304 SS for the regenerator, eliminating 1.78 lbs. of chrome from each Mod I engine. A ceramic air preheater is also being considered that would eliminate another 3.075 lbs. of chrome from the current Mod I engine.

Table 3-3 summarizes the current and potential reductions in strategic elements for the Mod I-A engine. Mod I engine parts weighing 55.557 lbs. contain 23.33 lbs. of cobalt, 13.378 lbs. of chrome, and .047 lbs. of columbium; those same parts in the Mod I-A design contain no cobalt, 8.08 lbs. of chrome, and .199 lbs. of columbium.

Efforts will continue during the latter part of 1982 to examine the material requirements of each engine component to ensure that the material with the least amount of strategic elements is used.

TABLE 3-3

STRATEGIC MATERIALS REDUCTION

Component	Engine Weight (lbs.)	Mod I Alloy	Mod I-A Alloy
Cylinder & Regenerator Housing	41.51	HS-31	XF-818
Heater Tubes	4.70	N155	CG-27
Regenerator	9.347	--	Silicon Carbide*
Total	55.557		

*potential

Mod I/Mod I-A Strategic Materials Weight (lbs.)

Component	Co	Cr	Cb	Ta
Cylinder/ Regen. Housing	22.4/0	10.60/7.7	0/166	0/0
Heater Tubes	0.93/0	0.998/0.1	.047/033	0/0
Regen.	0/0	1.780/0	0/0	0/0
Totals	23.33/0	13.378/7.8	.047/199	0/0

ALTERNATIVE MATERIALS' SELECTION

Although testing of alternative casting/heater tube materials continues, the selection for the Mod I-A engine has been

[1]Stephens, Joseph R., "A Status Review of NASA'S COSAM (Conservation Strategic Aerospace Materials) Program", NASA Tech. Memorandum.

made. The criteria for selection was, in order of priority (first being highest):

- strategic material content;
- weldability;
- fatigue strength;
- castability;
- creep strength;
- cost;
- density/weight; and,
- availability in U.S.A.

After comparing the above qualities of the three alternate alloys, XF-818, CRM-6D, and SAF-11, XF-818 was selected. The main selection criteria was weldability; XF-818 is very weldable by the TIG process, aiding in manufacturing and repair. The selection criteria for alternate heater tube material were, in order of priority (first being highest):

- strategic material content;
- creep strength;
- oxidation resistance;
- hydrogen permeation resistance;
- cost;
- brazeability; and,
- availability in the U.S.A.

The crucial factors in the selection* of CG-27 were creep strength and hydrogen permeability. As testing of alternate materials continues, and as the Stirling engine design evolves, the selection for future engine builds may change.

HIGH-TEMPERATURE TESTING OF ALTERNATE CASTING/HEATER TUBE MATERIALS

Another test to aid in the selection of alternate materials is high-temperature engine testing of heater head casting and tube materials. The primary objective of this test is to rank the candidate casting materials by testing them to failure in an engine environment. Type of failure and time to failure will aid in the selection process for the RESD and future engine builds.

Two four-quadrant heater heads have been manufactured to a modified P-40 Stirling engine design. The different combinations of casting and tube materials is shown in Table 3-4.

TABLE 3-4

TUBE/CASTING MATERIALS

Quadrant No.	Material	
	Casting	Tube
1	HS-31	Inconel 625
2	CRM-6D	Sanicro 32
3	XF-818	12RN72
4	SAF-11	CG-27
5	HS-31	Sanicro 32
6	CRM-6D	Inconel 625
7	XF-818	12RN72
8	SAF-11	Sanicro 31H

The heater head contains a cast cylinder/regenerator housing, along with cold extruded heater tubes that are brazed into the housings. The baseline cast material was cobalt-base alloy HS-31; the three iron-base alternate materials currently being investigated are XF-818, CRM-6D, and SAF-11. Because of breakage problems during casting, SAF-11 has been removed from future testing programs. The nominal compositions of these alloys are shown in Table 3-5. The baseline tube material was the cobalt-chrome-nickel alloy Multimet (N-155). The five alternate alloys are iron-nickel-base CG-27, nickel-base Inconel 625, and iron-base Sanicro 32, Sanicro 31H (Incoloy 800H), and 12RN72. The compositions of these alloys are shown in Table 3-6.

The test, currently in progress, is being run at a mean heater tube temperature of 820°C with 1% CO_2 doped hydrogen as the working gas. As of the end of June, over 1000 of the total 2000 hours had been accumulated. The first 1000 hours were conducted with a mean pressure variation of 4 to 7.5 MPa every 90 seconds; the next 500 hours will be run with a mean pressure variation of 4 to 9 MPa. The mean pressure variation of the last 500 hours will depend on the failure history over the first 1500 hours.

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*The selection memo is attached as Appendix A in MTI Report No. 82ASE265MT49.

TABLE 3-5

HEATER HEAD CASTING ALLOYS' NOMINAL CHEMISTRY

Alloy	C	Si	Mn	Ni	Cr	Mo	W	Co	Cb	B	N	Fe
HS-31	0.50	0.50	0.50	10.5	25.5		7.5	54.0				1.0
CRM-6D[1]	1.05	0.55	4.75	5.0	21.0	1.0	1.0		1.0	.005		64.5
SAF-11[2]	0.60	0.70	0.70	16.0	24.0		13.0			.45		44.5
XF-818[3]	0.20	0.30	0.15	18.0	18.0	7.5			0.4	.70	.12	54.0

[1]Chrysler Research
[2]Sorcery Metals
[3]Climax Molybdenum

TABLE 3-6

ALTERNATIVE HEATER TUBE MATERIALS' NOMINAL CHEMISTRY

Base Material	Co	Cr	Ni	Mo	W	C	Al	Ti	B	Cb	Mn	Fe	Si	N
Multimet (N155)	19.75	21.25	20	3.00	2.5	.12	-	-	-	1.0	1.50	29.7	1.0	.15
Alternatives:														
CG-27[1]	None	13.0	38	5.75	-	.05	1.6	2.5	.01	0.7	--	38.0	-	-
Inconel 625[2]	None	21.5	61	9.00	-	.05	0.2	0.2	-	-	0.25	2.5	0.2	-
Sanicro 32[3]	None	21.0	31	-	3.0	.09	0.4	0.4	-	--	0.60	42.8	0.6	-
Sanicro 31H[3]	None	21.0	31	-	-	.07	0.3	0.3	-	-	0.60	46.13	0.6	-
12RN72[3]	None	19.0	25	1.40	---	.10	---	0.5	.006	---	1.80	51.80	0.4	---

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

A 500-hour inspection revealed that one of the CRM-6D regenerator housings had developed a crack where the manifold meets the housing. This crack did not grow after the next 500 hours of testing, and no other failures occurred during the first 1000 hours of testing.

Cold Engine System (CES) Development

The focus of CES activity is the development of reliable, low-friction, reciprocating seals combined with long life in an automotive duty cycle. During the first half of this report period, effort was concentrated on the design of efficient piston rings, and the evaluation of these concepts in a single-cylinder test rig. During the latter part of this report period, emphasis will be placed on baseline evaluation (carried out in a reciprocating test rig) of the Pumping Leningrader (PL) seals (current main seal for the Mod I engine).

The goals for 1982 are to evaluate the new piston ring design in a motored Mod I engine, and evaluate main seal designs that can operate without a capseal.

PISTON RINGS

The most promising piston ring design developed in the ASE Program, the H-ring, is pressure-balanced so that the force holding the piston ring in contact with the cylinder is independent of the working gas pressure. A separate, internal expander ring is then used to exert a constant radial force on the piston ring so that the ring stays in contact with the cylinder. By changing the piston ring dimensions and/or expander ring, designing for a specific, predictable friction level is possible.

The first H-ring design (H-1) had the same overall dimensions as the Mod I solid piston ring. A 15° scarf-cut joint was introduced, and the individual piston rings were fitted to the cylinder diameter with no interference. Initial gas leakage was high, but decreased with time. The low contact pressure in this design extended the run-in period. Testing was terminated after 68 hours.

In the second H-ring design (H-2), the rings were made to have an initial interference of 0.1 mm with the cylinder bore, and the center edges of the rings were chamfered to reduce initial contact area. Both changes served to increase initial contact pressure and speed up the run-in process. As run-in wear took place, contact force decreased and contact area increased. These effects were apparent from a gradual reduction in friction and leakage.

The third H-ring design (H-3) is basically the same as H-2 except that a butt joint is used instead of a scarf joint. Overall H-3 ring performance was very similar to that of the H-2 rings. H-2 and H-3 ring performance is summarized in Table 3-7 where equivalent data for Mod I solid and split/solid rings is included for comparison.

TABLE 3-7

PISTON RING DATA

Design[1]	Life (hrs)	Friction Force[3] (#)	Hydrogen Leakage (ℓ/min)
Mod I Solid	42[2]	200	< 1
Mod I split/ Solid	19[2]	100	< 1
H-1	> 68	25	<10
H-2	>110	60	< 8
H-3	> 30	45	< 8

[1]All piston rings made from Rulon LD.
[2]Life determined by complete failure to seal gas pressure.
[3]After run-in.

The substantially lower friction forces with the H-rings represent a direct reduction in friction power loss. On the basis of Exploratory Seals Rig testing*, the H-rings should give a power loss that is less than 50% of that with the solid rings. If this performance is reproduced in the engine, piston ring losses would

*see MTI Report No. 82ASE248SA1

decrease by at least 4 kW (5.4 hp) at maximum speed and pressure. In the H-ring design, the piston ring is held in contact with the cylinder wall; therefore, unless the ring is damaged or excessively worn, a rapid loss of sealing action will never occur (the major mode of failure with the Mod I ring designs). Leak rates for the H-rings were an order of magnitude greater than the Mod I ring designs, but an analysis showed leak rates of the order of 10-20 ℓ/min. will have a negligible affect on the power developed by the engine.

With the scarf and butt joints, the gap between the ends of the rings provides a potentially large leak path that will increase with ring wear. In a further refinement of the H-ring concept, the rings were designed with a double-lap joint, allowing the ring to expand freely, but having no direct leak paths. Further H-ring testing will be carried out in both a motored and live engine.

MAIN SEALS

A special test head was designed to evaluate main seals in the Exploratory Seals Rig; the overall configuration of the test head is shown in Figure 3-29.

Two seals were tested simultaneously in a back-to-back arrangement. The section between the two seals was pressurized with hydrogen, and hydrogen leakage past the seals was measured via a mass spectrometer leak detector using the same technique employed for the piston ring tests. The reciprocating rod (equivalent to the piston rod in the engine) is guided by two hydrostatic bearings that straddle the seal section, and the test head is instrumented to monitor seal temperature and friction force.

To provide baseline data for main seal development, the Mod I PL seals were tested first. These seals are made from HABIA PTFE (polytetrafluoroethylene), which contains carbon and graphite. The first pair of seals had no measurable static hydrogen leakage at installation; they were then run-in for 3 hours at 1000 rpm and 500 psi hydrogen pressure. At the end of this period, gas leakage under dynamic conditions was 0.012 ℓ/min per seal. Following this, the seals were run at different speed/pressure combinations in ranges of 1000-4000 rpm and 500-1500 psi.

During the first 15 hours of testing, gas leakage rates did not exceed 0.023 ℓ/min; however, from that point on, gas leakage rates increased rapidly with time. To illustrate this, measurements made over the 17-to-21-hour period are given in Figure 3-30.

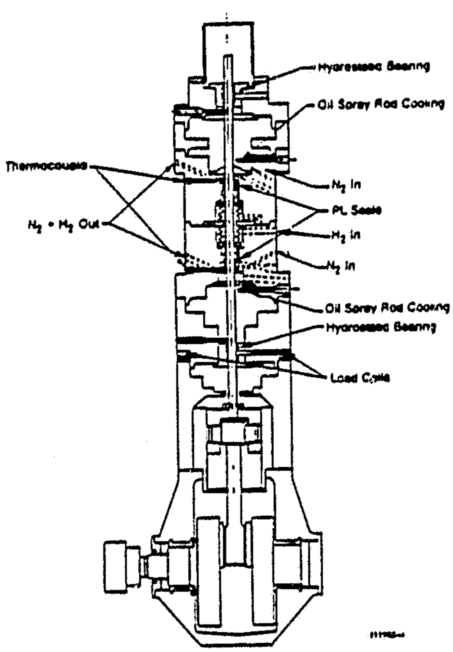

Fig. 3-29 PL Seal Test Head

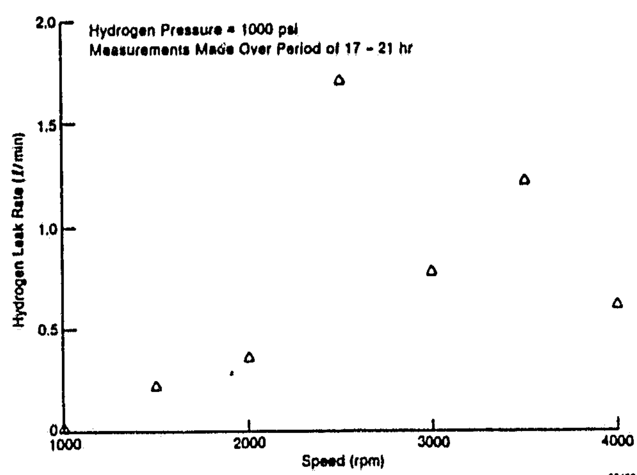

Fig. 3-30 Hydrogen Leak Rate for One PL Seal

Gas leakage rates were found to be somewhat erratic, but the general trend was increasing leakage with time. After 40 hours of testing, both seals had consistently high hydrogen leakage under dynamic conditions. When testing was discontinued after 65 hours, the seals were removed, and the seal cavity and its components were found to be saturated with oil that had leaked past the seals. Repeat tests on a second pair of PL seals gave results similar to those of the first pair of seals. These tests were terminated after 25 hours, when both seals had dynamic gas leakage rates in the 1-2 ℓ/min range. Again, there was substantial oil leakage past the seals over the short duration of testing.

Following the baseline PL seal tests, two pairs of Modified Leningrader seals were evaluated. These seals (shown in Figure 3-31), also made from HABIA PTFE, performed under test in a manner very similar to the PL seals, i.e., gas leakage rate increased rapidly with time, and there was significant oil leakage during the short test duration.

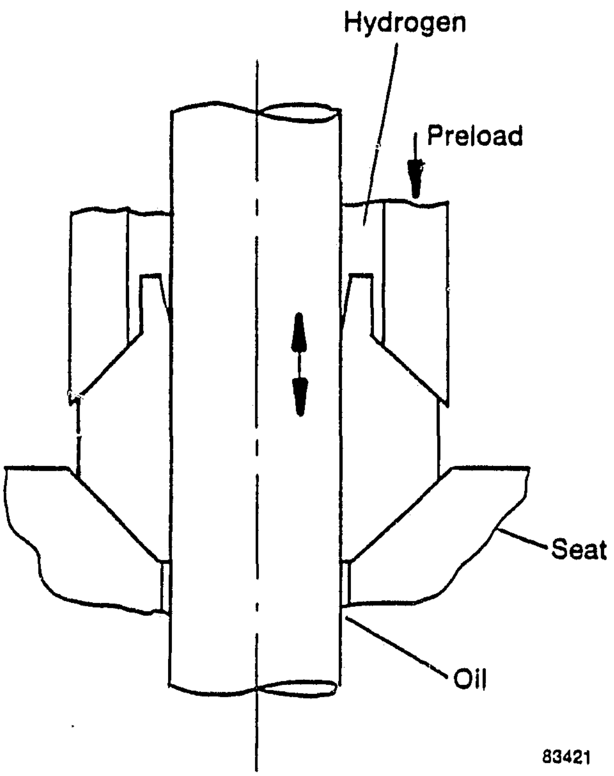

Fig. 3-31 Modified Leningrader Seal

During the first half of this semiannual report period, two pairs of Rulon LD Modified Leningrader seals were tested. These seals proved to be superior to the seals tested previously. Over the first 60-80 hours of testing, hydrogen leakage rates were mainly in the 0-0.005 ℓ/min range, with isolated higher readings; however, leakage rates increased rapidly, so tests were terminated after 90-100 hours. Some oil leakage also occurred during testing, but this was small when compared to that seen with the PL seals tested over a shorter time period.

An analytical study of lubricated piston rod seals (such as the PL seal) has been initiated. This analysis will provide basic data for the design of effective piston rod seals that will be fabricated and evaluated in the Exploratory Rig during the latter half of 1982.

Engine Drive System (EDS) Development

The primary goal of this task is to develop a Reduced Friction Drive, 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 baseline characterization of the EDS motored friction, design and characterization of cylinder liner P_{min} venting hardware, the Reduced Friction Drive design, and initiation of hardware fabrication.

During the latter half of 1982, work will continue on seal testing/evaluation with the EDS Motoring Unit, and evaluation of the Reduced Friction Drive.

MOD I MOTORING RIG

A Mod I engine drive with a dummy heater head has been installed as a Motoring Rig and checked out in preparation for the determination of baseline drive and seal friction losses. The EDS and drive motor are mounted on a test table. The motor is a 20-hp DC unit with a speed controller that provides motor speeds from 83.3 to 2500 rpm. A toothed belt/pulley system with a 1.6:1 step-up ratio provides motoring speeds in the range of 133 to 4000 rpm at the EDS input shaft. Engine shaft and drive shaft speeds are monitored with speed transducers; the speed signals are compared to prevent safety clutch slippage. A torque transducer mounted in the driveline provides continuous torque input readings, and flexible couplings isolate the torque transducers from shaft side loading. Working gas pressures P_{mean}, P_{max}, and P_{min} are measured for each cycle.

A cross section of the Mod I EDS Motoring Unit is shown in Figure 3-32. The actual arrangement is shown in the cross-sectional view where the expansion space of one cylinder is connected to the compression space of an adjacent cylinder through the cold connecting duct. Gas flow paths are the same for the Motoring Rig and engine.

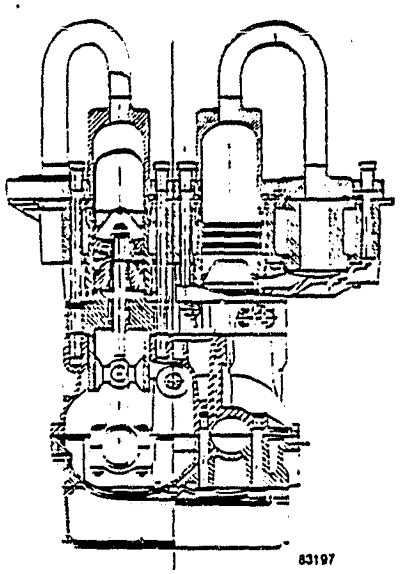

Fig. 3-32 Mod I EDS Motoring Unit

Baseline engine drive friction (seals and mechanical losses) was determined in both helium and hydrogen over a range of gas pressures; the results (see Figure 3-33) compare favorably with USAB data at 9 MPa. Helium was used as the working gas to increase the drive power requirements due to increased pumping losses. The test results will be used as a basis for evaluation of the Reduced Friction Drive (now in its final fabrication phase), which will utilize split-race, rolling element bearings that project to yield a 1.3-kW (1.7-hp) reduction in engine drive friction (exclusive of seal losses).

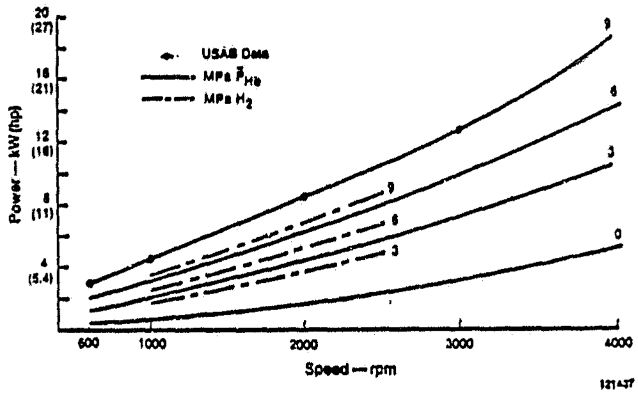

Fig. 3-33 Mod I EDS Baseline Drive and Seal Friction Losses

REDUCED FRICTION DRIVE

The preliminary design of a Lightweight Reduced Friction Drive with an EDS weight of 31 pounds less than the current EDS (a breakdown of projected weights savings is shown in Table 3-8) was completed during this report period. This goal is consistent with the Proof-of-Concept RESD Program.

TABLE 3-8

MOD I EDS WEIGHT REDUCTION

Component	Mod I EDS (lbs)	LRFD (lbs)	Δ (lbs)
Crankshaft 1 & 2	23.75	13.0	-10.75
Crankshaft 3 & 4	28.90	13.9	-15.00
Drive Shaft	5.80	2.9	- 2.90
Drive Shaft Gear	6.20	3.7	- 2.50
Total			-31.15

P_{min} CYLINDER LINER

Design of the cylinder liner P_{min} system hardware* for the H-ring was completed during this report period. Figure 3-34 shows the general arrangement of the key elements of the system.

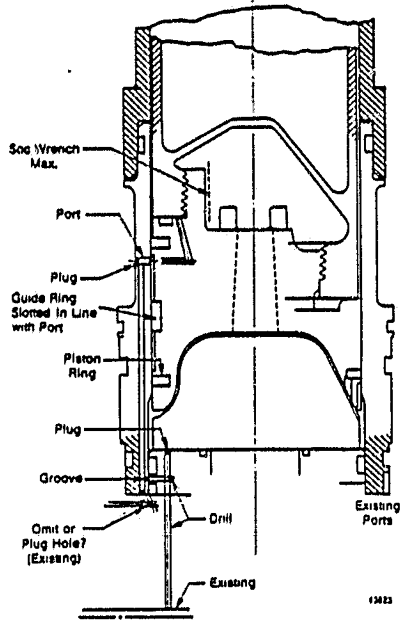

Fig. 3-34 Cylinder Liner P_{min}

EDS MECHANICAL LOSS BREAKDOWN

The objective of these tests was to determine individual mechanical component friction power on the Mod I Motored EDS Test Rig.

The approach used was to record friction power versus oil temperature at specific levels of assembly, remove a component, and then repeat the tests. Dynamic force and pressure balance considerations dictated mechanical configuration/operating envelope limitations, i.e., motoring speed was limited to 600 rpm with pistons removed. Data was double-checked by repeating the above procedure in reverse.

The lowest level of assembly for which reliable friction power data could be obtained was comprised of the main shaft and crankshaft 1 and 2 with connecting drive gears. Crankshaft 3 and 4 was left in place to restrict oil flow to normal levels; however, the drive gear between the main shaft and crankshaft 3 and 4 was removed. Driving crankshaft 1 and 2 was necessary to provide lubricant to the shaft journals, since the oil pump drives off the front end of crankshaft 1 and 2.

The system in this configuration was operated at 600 rpm over an oil temperature range of 24-40°C, and torque data was recorded. The 3 and 4 crankshaft drive gear was reinstalled, and the test cycle was repeated. A torque increase of ~14 in-lb with the 3 and 4 crankshaft coupled to the drive shaft is attributed to the crankshaft journal friction and the drive gear mesh losses. Results of the above series of tests are shown in Figure 3-35.

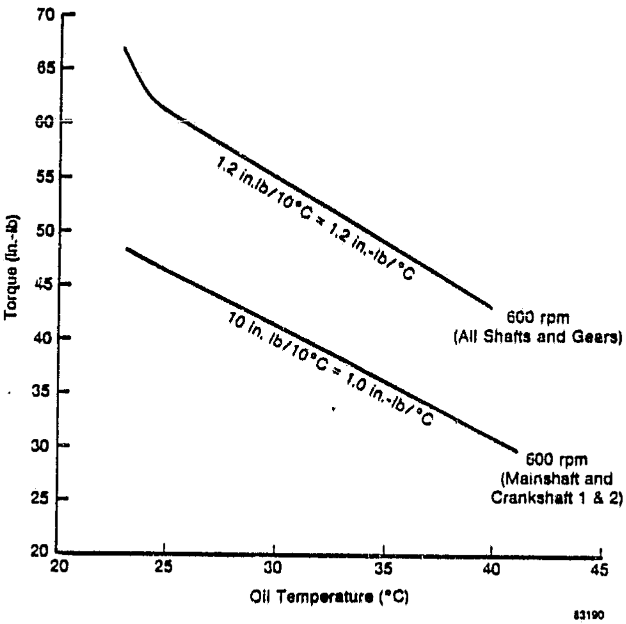

Fig. 3-35 Mod I EDS Shafts and Gears (Mainshaft/Crankshaft 1-2)

The next series of tests was conducted with the connecting rods, piston rods, crosshead guides, PL seals, and pistons reinstalled; the cap seals, cylinder liners, and dummy heads were not installed. There was no practical way to break the system down to a lower level for this

*modified domes, new pistons, piston feet, cap seal housings, and modified crosshead guides

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test series because the connecting rods required guidance at the upper end. Guidance is provided by the crosshead, which is part of the piston rod. The piston rod, in turn, is guided at the lower end by the crosshead guide, and at the top end by the guide bushing in the seal housing. Installation of the PL seal is required for oil control; the pistons were installed for balance considerations only.

The incremental torque increase recorded in this test sequence was thus comprised of friction from the connecting rod journal bearings, crossheads, PL seals, and piston rod guide bushings. The combined torque for these elements was ~18 in-lb. Figure 3-36 shows the results of this test plotted over the previous test results.

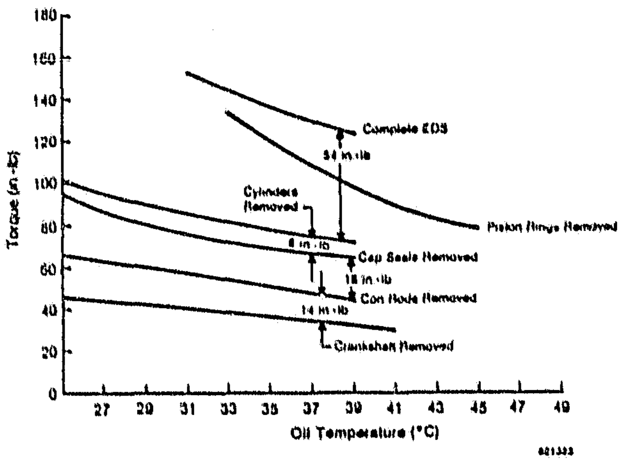

Fig. 3-36 Mod I EDS Tests

The next test series was conducted with the cap seals reinstalled. The system was not pressurized since the cylinders were not installed; thus, the 8.4-in-lb torque contribution attributed to the cap seals (shown in Figure 3-37) is not indicative of the torque that occurs in a pressurized operating environment. Attempts to determine the torque contribution of the cap seals under actual operating conditions in subsequent tests was unsuccessful due to excessive torque buildup with relatively low system pressure.

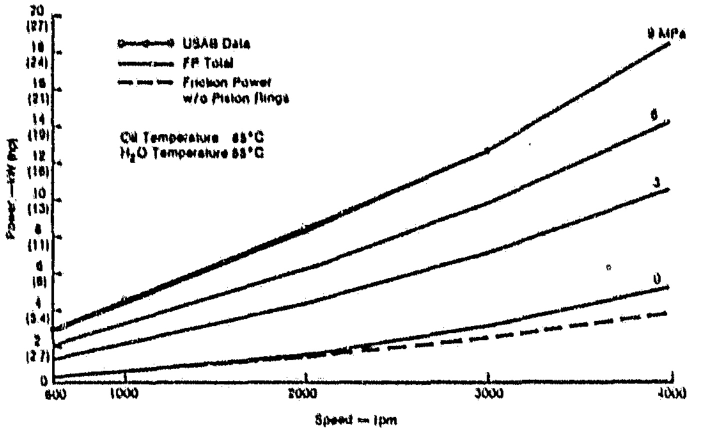

Fig. 3-37 Mod I EDS Tests

The first such test was conducted with all mechanicals in place, with the exception of the piston rings. The high torque noted under these conditions was thought to be the result of increased pumping losses caused by excessive piston blow by. The pistons were removed entirely, and the tests were repeated to verify this theory. No appreciable reduction of friction was noted versus that with the pistons installed; therefore, it was concluded that the cap seal was sensitive to the pressure balance across it - a condition that did not exist without properly sealed pistons in place. This unfavorable pressure balance caused the cap seal to squeeze the rod, resulting in high friction and sufficient heat generation to slightly discolor the rod surfaces in the cap seal contact area.

The next test series was conducted with a complete EDS. The torque increase associated with the piston rings (see Figure 3-38) was ~60 in-lb at 0 psig charge pressure. This operating condition is obviously not representative of actual engine operating conditions where the piston ring torque contribution, like the cap seal/PL seal torque contributions, increases as a function of system charge pressure. Subsequent tests were conducted to determine the relationship of friction losses to charge pressure. Unfortunately, there was no way to isolate individual seal elements to identify the torque contribution of each element under these conditions.

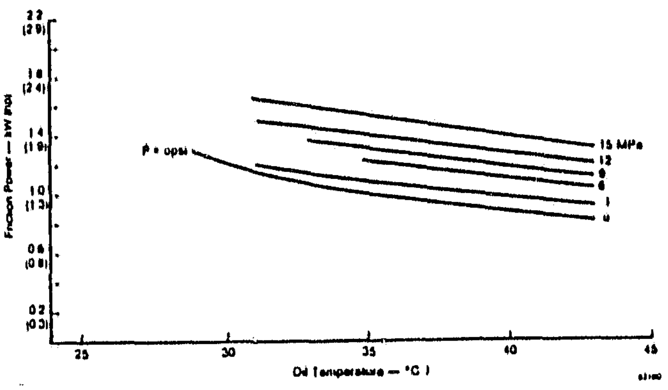

Fig. 3-38 Mod I EDS PL Seal (1000 rpm)

The third test sequence was conducted with the complete EDS over the full speed range (600-4000 rpm) at mean pressures of 3, 6, and 9 MPa. Attempts to obtain data at higher charge pressures were unsuccessful due to the drive motor torque limitations. Inlet cooling water temperature was maintained at ~55°C, and oil sump temperature at ~85°C. Figure 3-37, a plot of the resulting data, contains an overplot of the USAB data at 9 MPa that is in close correlation with MTI data.

The fourth test series was conducted with the pistons and cap seals removed to determine PL seal losses. All data (plotted in Figure 3-38) was taken at 1000 rpm for charge pressures of 0, 3, 6, 9, 12, and 15 MPa. The increase in power loss at full speed (4000 rpm) is estimated to be ~4 kW (5.4 hp) for the PL seals at 15 MPa versus 0 MPa. The friction loss of the unpressurized PL seal could not be segregated from the rod bearing and crosshead losses because lubricant control considerations precluded operating the EDS without the PL seals installed.

Friction loss for the three elements combined at 4000 rpm (no pressure loading) was ~0.8 kW (1.07 hp). No more than 0.3 kW (0.40 hp) should be attributed to the PL seals; thus, maximum PL seal losses are in the order of 2.2-2.4 kW (2.9-3.2 hp).

The fifth test was conducted to determine the relationship of the cap seal friction losses to system pressure. All tests were conducted at 1000 rpm at charge pressures of 0, 3, 6, 9, 12, and 15 MPa with and without cap seals installed. Figure 3-39 compares the torque values obtained for both conditions. From this data, the maximum cap seal loss was extrapolated to be ~2.0-2.5 kW (2.6-3.3 hp) at 15 MPa and 4000 rpm.

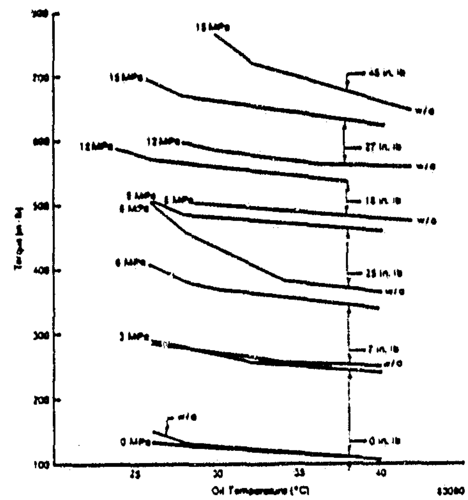

Fig. 3-39 Mod I EDS With and Without Cap Seals (1000 rpm)

The sixth test series conducted were to complete the EDS characterization up to full pressure (15 MPa). This required a change in the motor to EDS drive ratio (from 1.0:1.6 to 1.0:1.0) to provide adequate torque to motor the EDS at the higher pressures. As a consequence, maximum motoring speed was limited to 2500 rpm for these tests. Test results and overall system losses/breakdowns are summarized in Figure 3-40. The large indifference in friction power attributed to piston rings and gas losses for helium versus hydrogen suggests that further testing should be conducted to determine a more precise breakdown between mechanical friction and gas hysteresis effects.

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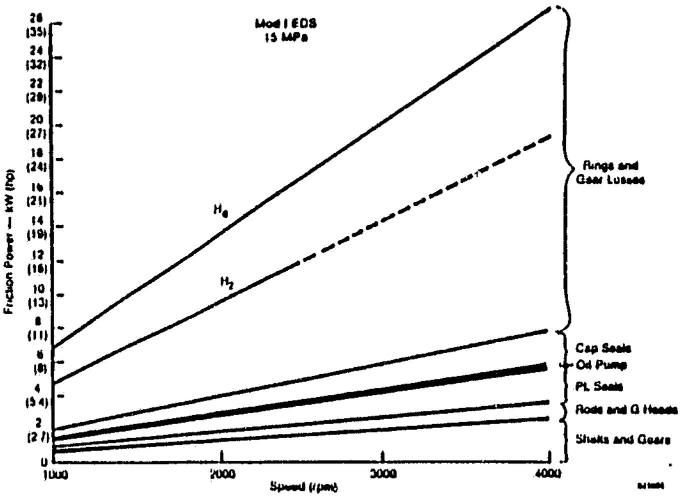

Fig. 3-40 Loss Breakdown Curves for 15 MPa Mean Pressure

H-ring testing was attempted next; however, the tests were inconclusive since initial testing was conducted with hardware that provided a P_{mean} reference pressure. H-rings are designed to operate with a P_{min} reference pressure, a condition that is provided by the P_{min} cylinder liners. Both butt joint and double-lap joint configurations were tested. These tests will be repeated with the P_{min} cylinder liner hardware.

Control System Development

The major goals of this task are to develop and evaluate a simple, reliable, driver-acceptable, microprocessor-based electronic control, and to develop an electronic air/fuel control with low pressure drop, low minimum fuel flow, and a programmable air/fuel ratio. These hardware designs must be compatable with the extremes of an automotive operating environment.

Development during this report period focused on the testing of the baseline air/fuel control, electronic combustion control transducer evaluation and system hardware, Digital Engine Control repackaging, and installation of the prototype digital control in the Lerma transient test bed.

During the last half of 1982, activity will focus on the fabrication, testing, and eventual selection of a prototype air/fuel control, upgrade of digital control software, and transient characterization of combustion control hardware.

AIR/FUEL CONTROL SYSTEM

During Engine Controls testing, two characteristics of the Air/Fuel Control System were noted; both low fuel flow stability and transient response could be improved. In addition, the potential for controlling λ as a function of an appropriate engine parameter accelerated the effort to characterize the K-Jetronic System first, followed by the design and functionalization of an alternate Air/Fuel System.

Testing of the K-Jetronic on a Fuel Flow/Airflow Test Rig confirmed the performance observed during earlier engine testing (Figure 3-41). The transient characteristics (change in fuel flow for a step change in airflow) on the strip chart recording are sluggish, taking more than one second to reach the desired flow. Unlike the test configuration shown in Figure 3-42, the initial test did not include an in-line pressure regulator upstream of the fuel nozzle.

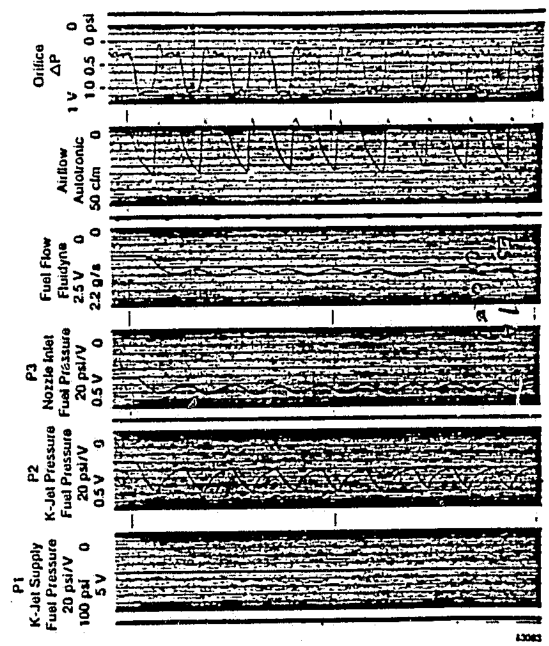

Fig. 3-41 K-Jetronic Testing in Fuel Flow/Airflow Rig

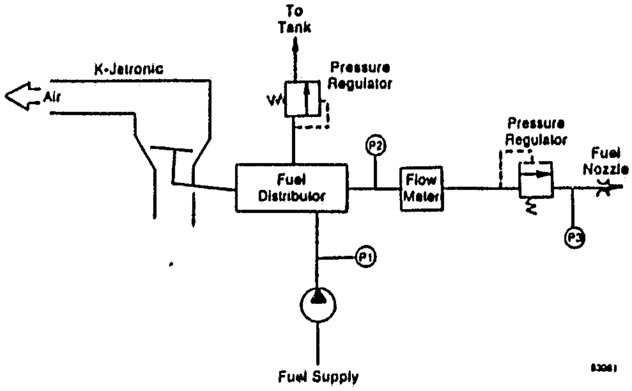

Fig. 3-42 Fuel System - K-Jetronic

A second set of tests, run with the in-line pressure regulator, indicated a dramatic improvement in fuel system response to a step change in airflow (Figure 3-43), leading to the conclusion that with the proper setting for the in-line pressure regulator, transient response of the K-Jetronic system can be good. The ability to achieve low fuel flows and vary λ were still not satisfied by that system.

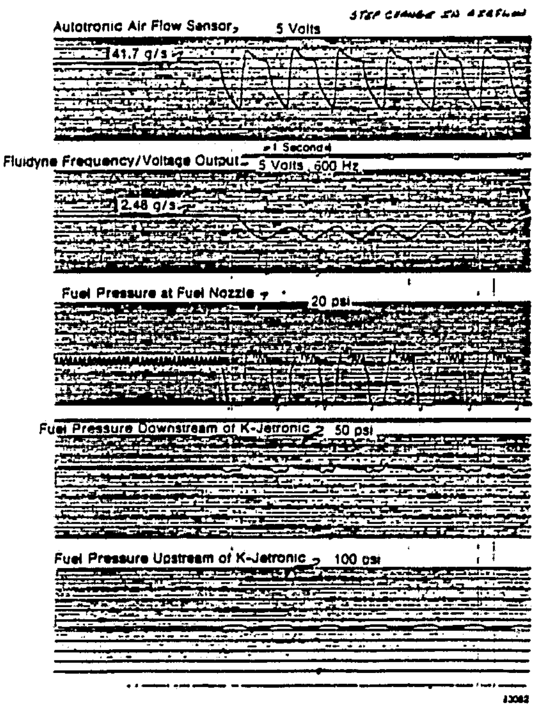

Fig. 3-43 K-Jetronic Testing With In-Line Pressure Regulator

COMBUSTION CONTROL SYSTEM

In the development of the Combustion Control System, several flow sensing devices for both airflow and fuel flow were evaluated on test rigs. Table 3-9 lists the components evaluated; the components selected and the system design are shown in Figure 3-44.

TABLE 3-9

FLOW MEASURING/METERING DEVICES UNDER EVALUATION

Airflow	
Chrysler	Rotating Vortex
J-Tec	Vortex-Shedding-Type
Autotronic	Turbine-Type
Fuel Flow Sensors	
Fluidyne	Piston Positive-Displacement
American Flow Systems	Momentum Absorption
Flow Scan	Turbine
Flow Technology	Turbine
Fuel Metering	
Micropump	

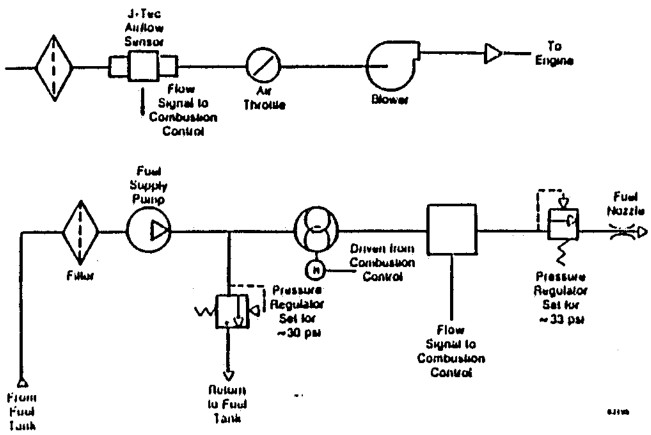

Fig. 3-44 Air/Fuel System Schematic

The microprocessor-based electronic controls for this system were fabricated in a system that permits parameter displays, on-line changing of control constants, and selection of different control algorithms. This development tool has proved very useful during rig testing of the Combustion Control System, where control hardware/electronics were first integrated. Ultimately, the Electronic Control package for the Combustion Control will be integrated with the Engine Control.

The next phase of development for this sytem is operation on an engine in the Engine Test Cell.

DIGITAL ENGINE CONTROL

The Digital Engine Control developed by USAB was first tested at MTI during controls testing conducted in 1981. Recomendations were made at that time for minor modifications to the control (see MTI Report No. 82ASE240ER33).

In January, 1982, acceptance testing of Mod I engine No. 1 was conducted at USAB with the modified Digital Engine Control. During testing, both engine performance and Digital Control performance were evaluated; the control provided a functional replacement of the Analog Control, with some enhanced features permitted because of the flexibility of digital systems.

Additional digital controls were found to be necessary to support the other ASE Program Mod I engines. As an element of technology transfer, the decision was made to build the additional units at MTI. In conjunction, the Digital Control would be built with components available in the United States, and packaged in a manner more suitable for automotive application.

Slight hardware design modifications have been made as a result of packaging considerations and component availability. The control will, however, remain basically the same as that produced by USAB. The software applied to this control will be the same as that found in the original control, except for two areas: 1) where hardware changes are required, parameter addressing must change; and, 2) sections of software might be removed or reduced because of the need for the original control to operate the P-40 as well as the Mod I engines. This reduction in code should permit more efficient coding.

The plan to accomplish this task involves three stages. The first stage is to design a multiboard system implemented with an STD buss structure. The design will be a fabrication of the required functional elements on single cards. The multiboard designs provides a system in which individual component compatibility can be confirmed, and also allows isolation of functions, facilitating hardware design debug. The second stage is fabrication of a compact prototype Digital Engine Control made of the hardware design specified from the multiboard system. Completion of the prototype system will complete the repackaged design effort. The third stage is the production of systems to support the ASE Program Mod I engines. A further effort, the fabrication of additional monitor/simulator systems, is aimed at utilizing components available from U.S. suppliers for additional units as required in the program.

LERMA VEHICLE INSTALLATION

The engine and controls have been installed in the Lerma vehicle, and functional checkout has been completed. Preparation of the vehicle involved installation of the wiring harness and connectors to accept the engine, control blocks, instrumentation, battery system, and Electronic Control. The dashboard and instrumentation were then installed. Installation, checkout, and alignment of the electronics and control components completed this task.

When the Combustor System was changed from CGR to EGR, a control valve and valve control were required. Following engine testing, a fixed orifice valve was selected that was to remain closed until ignition, and then open throughout the engine operating range. Digital Control logic was modified to provide the EGR valve control function.

IV. ENGINE OPERATING HISTORY

Quality Assurance Overview

Below is a status of the Quality Assurance Reports for this report period:

MQ's issued (113, through and including 158)	46
Total QAR's issued	70
Closeout Meetings 82-1 (3/9/82)	71
82-2 (6/30/82)	34
Total QAR's closed	105

The only trend in the 70 QAR's generated during this six-month period was related to the heater head quadrants, Part No. 17036.

S/N 1/1	272 hrs.	Braze Joint (Regenerator)
S/N 4/2	Initial Pressure Test	Casting Porosity (Cylinder)
S/N 4/4	Initial Pressure Test	Casting Porosity (Cylinder)
S/N 2/1	185 hrs.	Braze Joint (Regenerator)
S/N 3/3	28 hrs.	Tube Failure

The program has also experienced continuing delays in the receipt of acceptable castings from both Swedish and United States vendors.

The problem areas have been identified, and a redesign has been accomplished in the regenerator housing. The manifold has been altered to provide a single-center entry to the regenerator in order to improve castibility and working gas flow distribution.

An alternative casting material (XF-818) has been chosen for the Mod I-A design that has good weldability characteristics to allow for repair (this has not been the case with our present casting material). In addition, CG-27 has been chosen for the Mod I-A heater head tube material because of its superior characteristics exhibited in the areas of creep, oxidation resistance, hydrogen permeation, and brazeability.

Table 4-1 is a summary of operating times and mean time between failures for all ASE Program engines as of June 30, 1982. The primary useage of each engine is:

- 40-4 - High-Temperature (820°C) Endurance Testing at UASB;
- 40-12 -- Vehicle Demonstrations at MTI;
- Mod I No. 1 - Dynamometer Testing at USAB; and,
- Mod I No. 2 - Dynamometer Testing at USAB.
- Mod I No. 3 - Dynamometer Testing at MTI.

TABLE 4-1

SUMMARY OF MEAN OPERATING TIMES AND MEAN TIME BETWEEN FAILURES FOR ALL ASE PROGRAM ENGINES

ASE Engine	Operation Time*	Mean Operating Time Between Failures
40-4 (USAB)	7690	114.70
40-12 (MTI - Concord)	190	13.60
Mod I No. 1 (USAB)	456	76.00
Mod I No. 2 (USAB)	249	62.25
Mod I No. 3 (MTI)	261	261.50

*time prior to acceptance tests to the present.

V. FACILITIES

Below is a summary of the major facilities activities accomplished during this semiannual report period:

- The Engine Test Cell and DAS software were prepared for the installation and testing of Mod I engine No. 3. A timer was added to the engine's cooling system loop pump, and airflow measurement was changed from a hot-wire anemometer to a Chrysler airflow meter.

- Several transducers and a visicorder were temporarily installed for use in the Motoring Cell for a generation of P-V diagrams.

- A system for hydraulically and pneumatically pressure-testing engine parts was designed and installed in the High-Pressure Test Cell. A design to use a Lister Diesel Engine to durability-test regenerators was developed and completed, and parts were ordered.

- Parts were ordered for the Exploratory Preheater Test Rig; installation is 95% complete in the High-Temperature Heat Transfer Cell.

- The Seals Cell is operational.

- The CO_2 cooling loop was connected to the Combustion Performance Rig in the Combustion Cell. The Rig was fired for the first time, and is operational. Hookup to the HP-1000 DAS is in progress.

APPENDIX - TERMS AND DEFINITIONS

Symbol/Term	Definition
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
CVS	constant volume sample
DAS	Data Acquisition System
DC	direct current
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 Sytsem
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
m	meter
mi	mile
mm	millimeter
Mn	manganese
Mo	molybdenum
Mod I	first-generation automotive Stirling engine
Mod I-A	first upgraded Mod I engine design
Mod I-B	second upgraded Mod I engine design
MTI	Mechanical Technology Incorporated
MPa	megapascals
MQ	material quote
N	nitrogen
NASA	National Aeronautics and Space Administration
Ni	nickel
NO_x	oxides of nitrogen
NTU	number of transfer units
PL Seal	Pumping Leningrader Seal
P_{max}, P_{mean}, P_{min}	working gas pressures
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 Design
rpm	revolutions per minute
s	second
SES	Stirling Engine System
Si	silicon
S/N	serial number
SS	Stainless Steel
Ta	tantalum
TTB	Transient Test Bed
USAB	United Stirling of Sweden
VE	Value Engineering
W	tungsten
λ	$\frac{air/fuel}{air/fuel_{stochiometric}}$
η	efficiency
$\Delta\eta$	efficiency difference
ΔWt.	weight difference
ΔPower	power difference
α-I	alpha engine I
α-II	alpha engine II
ℓ/min	liters per minute