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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

SEMIANNUAL TECHNICAL PROGRESS REPORT FOR PERIOD: JULY 1 - DECEMBER 31, 1981

Mechanical Technology Incorporated

September 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

NOTICE

ERRATA

Report: Automotive Stirling Engine Development Program
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for Period: July 1 - December 31, 1981

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16. Abstract

This is the first semiannual technical progress report prepared under the Automotive Stirling Engine Development Program; it covers the fourteenth and fifteenth quarters of activity after award of the contract. Quarterly technical progress reports reported program activities from the first quarter through the thirteenth quarter; thereafter, reporting was changed to a semiannual format.

This report summarizes activities performed on Mod I engine testing and test results, progress in manufacturing, assembling and testing of a Mod I engine in the United States, P-40 Stirling engine dynamometer and multifuels testing, analog/digital controls system testing, Stirling reference engine manufacturing and reduced size studies, components and subsystems, computer code development activities. The overall program philosophy is outlined, and data and results are presented.

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Semiannual Technical Progress Report

Period Covered:

July - December, 1981

Approved By: N. P. Nightingale
Noel P. Nightingale
ASE Program Manager

Approved/Ratified By NASA: Richard H. Knoll 8/20/82

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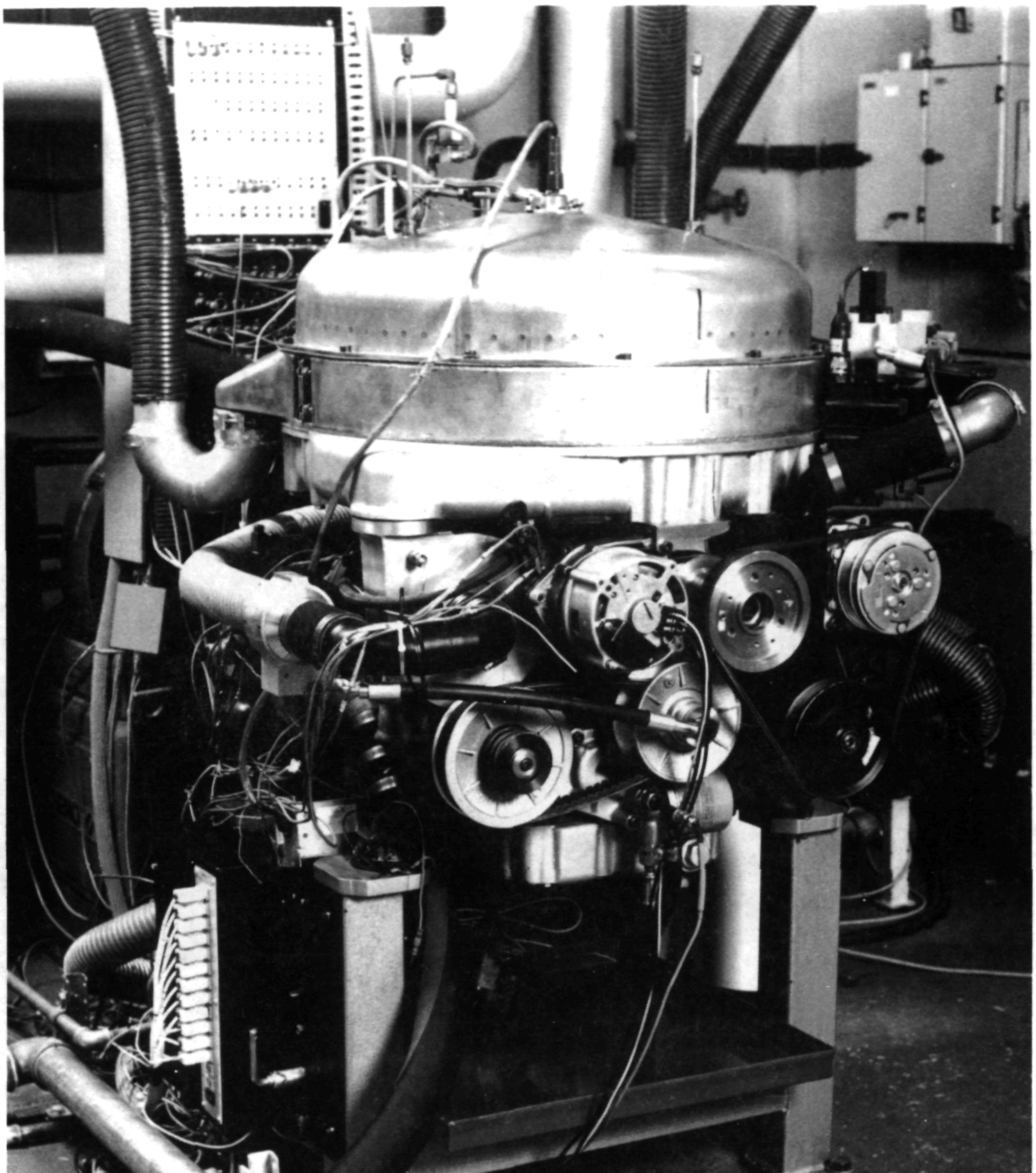

ASE MOD I ENGINE

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INTRODUCTION

In March, 1978, NASA awarded a Stirling engine development contract to Mechanical Technology Incorporated (MTI) for the purpose of automotive Stirling engine development and transfer of Stirling engine technology to the United States under the Deptartment of Energy (DOE) Heat Engine Programs. The program team consists of MTI as prime contractor, contributing their program management/development/technology transfer expertise; United Stirling of Sweden (USSw) as major subcontractor for Stirling engine development; and AM General (AMG) Corp. as major subcontractor for vehicle systems development and engine/vehicle integration. Most existing Stirling engine technology previously resided in Europe, and was demonstrated for stationary applications; therefore, the Automotive Stirling Engine (ASE) Development Program was directed at the establishment/demonstration of a base of Stirling engine technology for automotive application and the transfer of this technology to the United States 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.

The program features five P-40 engines for research and technology purposes only, and four Mod I engines (first-generation automotive engines).

The ASE Program has realized many major achievements in the last four years such as accumulation of engine/rig test hours; the publication (Sept., 1979) of an Initial Technology Assessment Report (NASA CR-15963) that presented the Stirling engine as a candidate automotive power plant; the manufacture/shipment of four P-40 engines* to the U.S. for Stirling engine familiarization testing at MTI/AMG/NASA, of which two were successfully installed in vehicles; a component development program with over 6700 test hours accumulated on rigs; and the manufacture of Mod I engines.

*Over 3500 hours have been accumulated on these P-40 engines.

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 has been changed to a semiannual format. This report, covering the period of July 1 - December 31, 1981, is the 1st Semiannual Technical Progress Report issued under NASA Contract No. DEN3-32. It includes technical progress only. Overall program objectives and major task descriptions are outlined below.

OVERALL PROGRAM OBJECTIVES

The overall objective of the ASE Program is to develop an automotive Stirling Engine System (SES) technology that 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: $NO_x = 0.4$, HC = 0.41, 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 design objectives are considered goals of the program:

- the ability to use a broad range of liquid fuels from many sources, including coal and shale oil;
- reliability and life comparable to powertrains currently on the market;
- a competitive initial cost and life-cycle cost comparable to a

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

conventionally powered automotive vehicle;

- acceleration suitable for safety and consumer considerations; and,

- noise/safety characteristics that meet currently legislated or projected Federal Standards for 1984.

MAJOR TASK DESCRIPTIONS

The overall objectives of the major program tasks are described below:

Task 1 - Reference Engine - This task, intended to guide component, subsystem, and engine system development, involves the establishment and continual updating of a Reference Engine System Design (RESD). The RESD 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 all 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 the engine 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 Technology Development - Guided by RESD activities, component/subsystem development will be conducted in support of various Stirling engine systems generated under the program, 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 and/or vehicle test installation.

Component development will be directed at advancing engine technology in terms of durability/reliability, performance, cost, and manufacturability. The tasks 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 USSw P-40 Stirling engine will be used as a baseline engine for familiarization, as a test bed for component/subsystem performance improvement, to evaluate current engine operating conditions and component characteristics, and to define problems associated with vehicle installation. Three P-40 Stirling engines will be built and delivered to the United States' team members; one will be installed in a 1979 AMC Spirit vehicle. A fourth P-40 Stirling engine will be built and installed in a 1977 Opel sedan. 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 System - A first-generation automotive Stirling engine (Mod I) will be developed using USSw 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 improvement; it will be utilized to verify concepts incorporated in the RESD, and to serve as a stepping stone toward the ASE Mod II engine. In this way, the Mod I will provide an early indication of its potential to meet the final ASE Program objectives.

Three engines will be manufactured in Sweden and tested in dynamometer test cells to establish their steady state performance, durability, and reliability.

Continued testing and development may be necessary in order 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 become a transient test bed to accept one of the above engines for installation. Tests will be conducted under various transient and environmental conditions to establish engine-related driveability (controls), fuel economy, noise, emissions, and durability/reliability.

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

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.

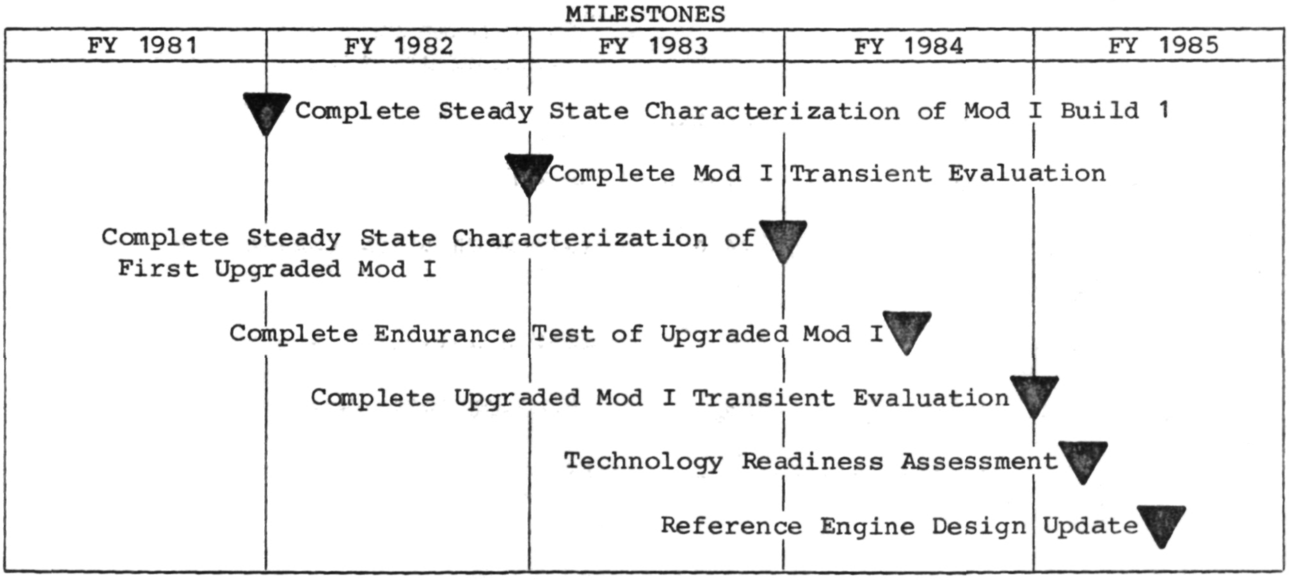
Task 9 - Program Management - Work under this task will provide total program control, administration, and management, including reports, schedules, financial activities, test plans, meetings, reviews, seminars, training, and technology transfer.

PROGRAM SCHEDULE

A 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 ASE Program's accomplishments over the last six months and plans for the next reporting period are presented below:

Major Accomplishments

A major program milestone, characterization of the Mod I engine, has been completed. The Mod I engine has met or exceeded all performance requirements. Performance at the key operating points was:

	Measured	Predicted
• max power	53.9 kW (72.2 hp)	50.0 kW (67.0 hp)
• max efficiency	37.4%	37.5%

As of December 31, 1981, a total of 422 and 7697 hours of operation have been accumulated on all ASE Program Mod I and P-40 engines, respectively.

A P-40 engine has been operated with the following alternative fuels without significant performance/emissions penalties:

- unleaded gasoline;
- commercial diesel;
- shale-oil-derived marine diesel;
- 10%/90% alcohol/gasoline blend; and,
- experimental referee-broadened-specification aviation turbine fuel.

A manufacturing cost study of the Reference Engine, performed by an independent automotive industry consultant, indicates that the cost of a Stirling engine could be comparable to that of a diesel engine.

Alternative low-cost heater head casting and tube materials have been characterized. A heater head constructed of these materials will only be 35% of the cost of the current design.

An alternative regenerator matrix, costing only 25% of the current matrix, and having little performance penalty, has been identified.

A seal material having lower wear and friction characteristics than the currently used materials has been identified.

Analytical codes to aid in engine design and transfer of Stirling engine technology to the United States have been developed and are being validated.

Work has been initiated on the development of new component designs to upgrade Mod I engine performance while reducing its weight and cost.

Work Planned for Next Reporting Period

The manufacture and acceptance testing of two additional Mod I engines will be completed, and Mod I engine development testing will be initiated.

A Mod I engine will be installed in a vehicle test bed to evaluate transient performance characteristics.

Assembly of a U.S.A.-manufactured Mod I engine will be initiated.

The design of an Upgraded Mod I engine system will be initiated.

Upgraded Mod I component development will be conducted in the following areas:

- improved combustion;
- reduced cost/weight preheater;
- low-cost heater head material characterization;
- reduced friction/longer life piston rings and main seals;
- reduced friction drive system; and,
- simpler, more reliable control system components.

A Reference Engine study will be initiated to evaluate the feasibility of applying the Stirling engine to subcompact automobiles.

I. MOD I STIRLING ENGINE

The Mod I Stirling engine is the first engine specifically designed and built for automotive application within the ASE Program. During development, it will be tested on an engine dynamometer as well as in a vehicle test bed.

A major program milestone, characterization of engine performance, was completed on schedule during this report period. Four sets of engine hardware were ordered, three complete engines were assembled, and acceptance testing was initiated. A total of 472 engine test hours were accrued by December 31, 1981. In addition to the above, which was performed in Sweden, work continued on the manufacture of a Mod I engine in the United States.

Engine acceptance testing will be completed during 1982, and development testing will be conducted on engine dynamometers and in a vehicle test bed. Assembly of the Mod I engine fabricated in the United States will also be initiated.

Engine Testing

The first Mod I engine underwent functional testing and completed an acceptance test program at USSw in September, 1981. Currently, the engine has accumulated 337 test hours on an engine dynamometer. A second Mod I engine currently undergoing functional testing has accumulated 86 hours of operation.

During acceptance testing, Mod I engine #1 demonstrated a peak power of 53.9 kW (72.3 hp) and a maximum efficiency of 37.4%. Measured fuel consumption islands are in a more favorable location for automotive application than those of conventional IC engines, and measured exhaust emissions from the Mod I engines are consistent with meeting the emissions requirements (0.4 g/mi NO_x, 3.4 g/mi CO, 0.41 g/mi HC).

The following paragraphs are an analysis of test data from the acceptance test for the Stirling Engine System, including the auxiliaries.

Power Characteristics

All acceptance test data was obtained using hydrogen as the working gas. Heater head temperature, as represented by the average of four rear-row heater tube thermocouple measurements, was set at 720°C ±10°C, and cooling water temperature at the entrance to the engine was maintained at 50°C ±0.5°C.

Engine power characteristics for various mean working gas pressures are shown in Figure 1-1. Selected comparisons were made against power predictions after engine hardware manufacture, but before the start of engine testing. Peak measured power was 53.9 kW (72.3 hp) at 4000 rpm and a 15 MPa mean working gas pressure (predicted peak power is 50 kW (67 hp)). Engine data show somewhat higher power than predicted for most operating conditions, and tend to deviate more from predictions at higher engine speeds. Performance predictions were based on the assumptions of: 1) a constant oil temperature of 85°C;

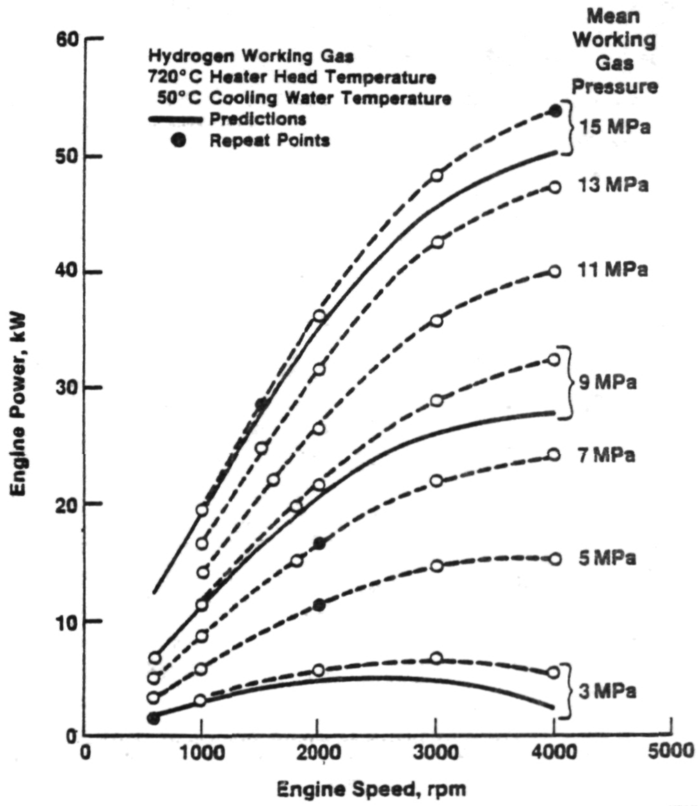


Fig. 1-1 Power Characteristics

2) nominal bearing clearances; and, 3) nominal engine operating conditions. If the actual operating conditions are included in the computer predictions, the agreement between analysis and measurements is improved. To check data repeatability, the operating conditions identified in Figure 1-1 were repeated three times during the acceptance test; for each test condition, data repeatability was within ±2%.

Efficiency Characteristics

Mod I engine efficiency characteristics for selected working gas pressures, as compared with their corresponding predictions, are shown in Figure 1-2. Peak measured efficiency was 37.4% at 1500 rpm and a mean working gas pressure of 15 MPa (compares with a predicted peak efficiency of 37.5%). Measured engine efficiency tends to be significantly higher than predictions for the lower mean working gas pressures, with this difference decreasing as mean working gas pressure is increased to its maximum value at 15 MPa. Engine efficiency falls below the predictions only at low engine speeds. At any given mean working gas pressure, peak efficiency tends to occur at a higher engine speed than predicted. As mentioned previously, if actual bearing clearances, engine oil temperature variations, and engine operating conditions are included in the computer predictions, the agreement between predictions and measurements is improved.

Data repeatability for engine efficiency was examined for specific operation points, as identified in Figure 1-2. For each test condition, the data repeated within ±2.8%, except at the minimum power condition, which showed a somewhat wider variation.

Specific Fuel Consumption Characteristics

Engine efficiency is alternatively represented on a specific fuel consumption basis in Figure 1-3. The Mod I engine demonstrated a specific fuel consumption of less than 0.4 lb/bhp-hr over a significant range of engine speed-load conditions. At any given mean working gas pressure, fuel consumption characteristic curves are rather flat, except for the lowest pressure level (3 MPa), which would see limited operation in an automotive application.

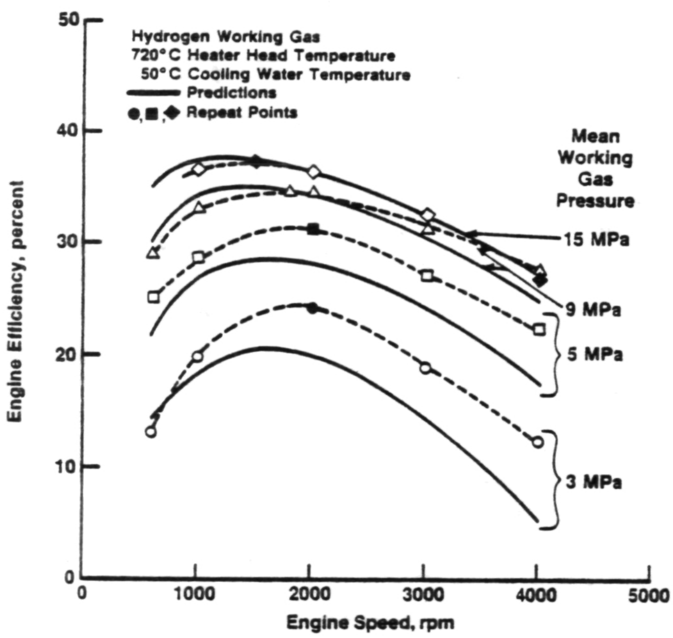


Fig. 1-2 Efficiency Characteristics

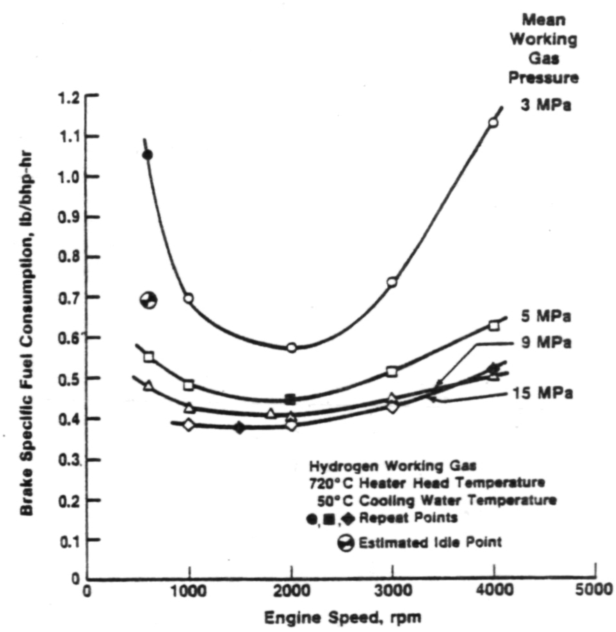


Fig. 1-3 Fuel Consumption Characteristics

Estimated idle condition in the vehicle is 600 rpm with a mean working gas pressure of 4.5 MPa. Based on acceptance test data, this engine idle condition will provide an output power of 2.8 kW (3.7 hp) to run the accessories and overcome torque converter losses.

Specific Fuel Consumption Comparisons

Mod I specific fuel consumption characteristics are compared to those of other conventional engine types in Figure 1-4. The comparisons are made at 2000 rpm, with the assumption that all three engines are scaled to provide the same power. Engine power is normalized with respect to maximum power at 2000 rpm for the engines being compared. This normalization permits a more realistic comparison of the merits of each engine type.

The average engine power required to drive a vehicle (3125-lb inertia weight) over urban/highway driving cycles is also identified. At the average power required by the urban cycle, the Mod I engine provides 30-35% lower fuel consumption than conventional gasoline and diesel engines; for the highway cycle, the Mod I engine provides 10-15% lower fuel consumption than conventional IC engines.

Torque Characteristics

The Mod I engine exhibits good, low-speed torque characteristics (Figure 1-5), with a maximum measured torque of 135 ft-lb at 1000 rpm and 15 MPa working gas pressure. The engine torque boundary shows a decreasing torque characteristic from this maximum value as engine speed is increased. Another engine operating constraint is imposed by limitations of the hydrostatic bearings in the drive. This constraint does not impose any significant restrictions on an automotive application of the Mod I engine. Symbols representing acceptance test data indicate good fuel consumption regions in Figure 1-5. The minimum fuel consumption region is located at engine speeds lower than 2000 rpm and at torque levels above 90 ft-lbs. Specific fuel consumption values, expressed as lb/bhp-hr, are shown adjacent to some of the engine data points in Figure 1-5.

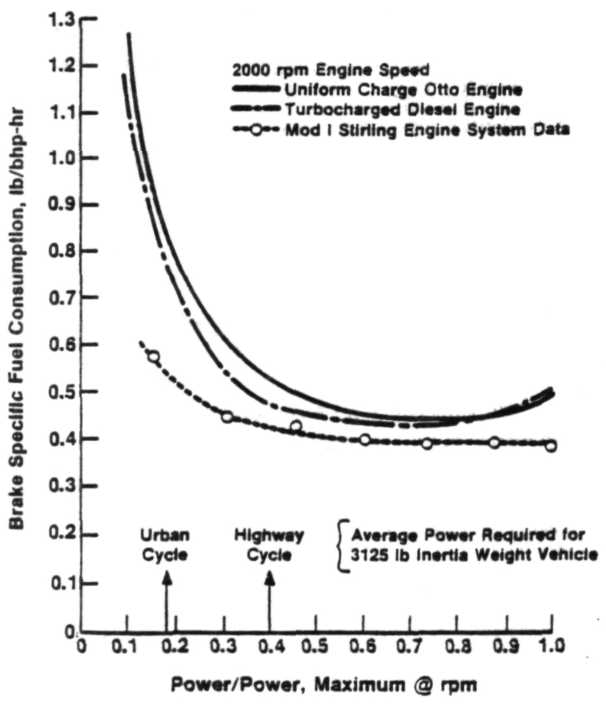


Fig. 1-4 Engine Fuel Consumption Comparisons

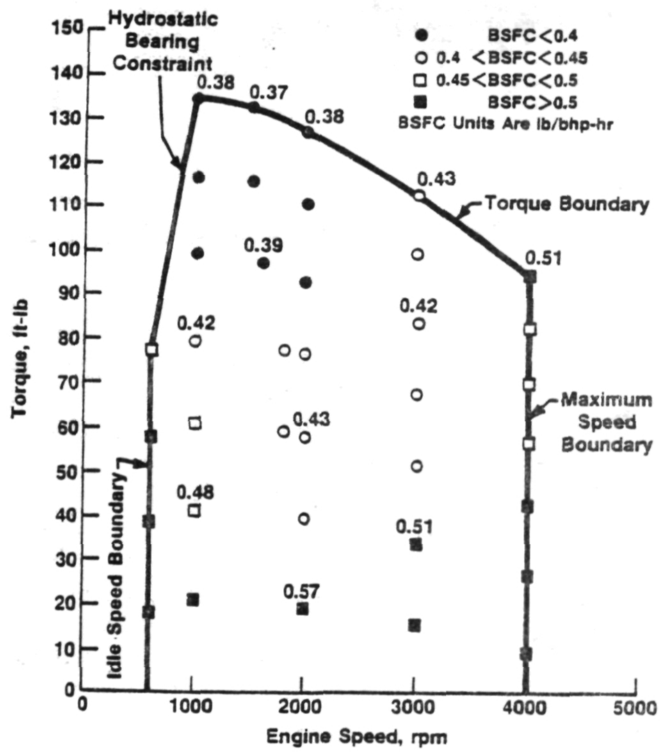


Fig. 1-5 Torque Characteristics

Torque-Speed Comparisons

Torque-speed characteristics for the Mod I engine are compared in Figure 1-6 to those of a conventional, naturally aspirated diesel engine. Engine torque and speed are normalized with respect to torque and speed at maximum power. For automotive applications, engines with a high torque ratio at relatively low engine speeds are advantageous since they exhibit better acceleration characteristics. This comparison shows that the Mod I engine has a better torque characteristic than conventional diesel engines, and also indicates that it will provide greater vehicle performance than a conventional engine of the same horsepower.

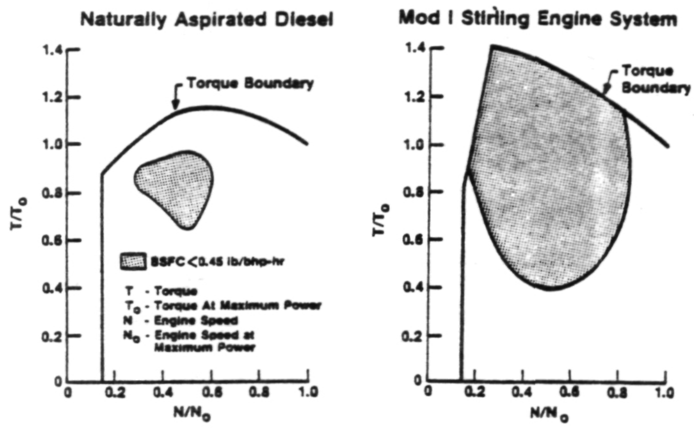


Fig. 1-6 Engine Torque-Speed Comparisons

Regions where specific fuel consumption is less than 0.45 lb/bhp-hr are identified on the torque-speed maps for the two engines. The region of low specific fuel consumption (bsfc <0.45 lb/bhp-hr) for the Mod I engine is about seven times as large as the corresponding region for the naturally aspirated diesel engine. This excellent part-load fuel consumption characteristic of the Mod I Stirling engine is particularly attractive for automotive applications.

Energy Balance Comparisons

The characteristic difference between the energy balance for the Mod I engine and that of a conventional gasoline engine is shown in Figure 1-7. This comparison, made at 2000 rpm, and at a power level that falls midway between the average levels required to drive a vehicle (3125 lbs. inertia weight) over urban/highway driving cycles, shows that in the Mod I engine, a significantly larger fraction of the energy is lost to the cooling water; however, at the same time, a smaller fraction of energy is lost in the exhaust gases. As a result, exhaust gas temperatures for the Mod I engine are significantly lower than those for a conventional gasoline engine.

Engine Speed = 2000 rpm
Engine Power = 0.3 × Maximum Power at 2000 rpm

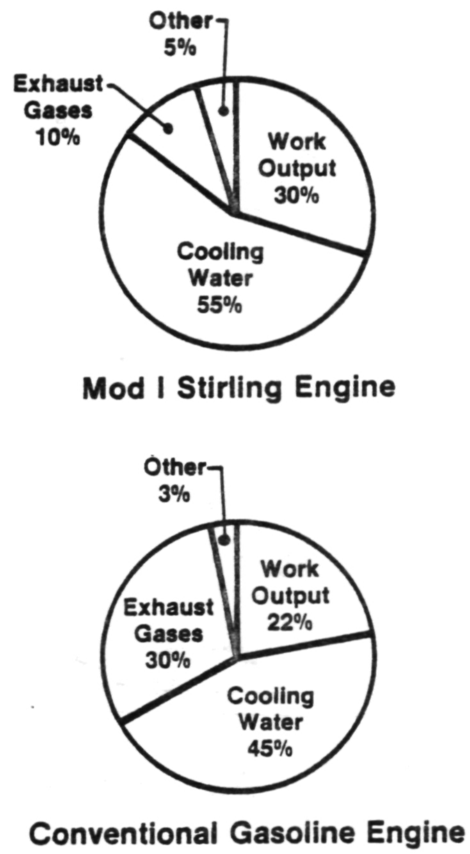


Fig. 1-7 Energy Balance Comparison

Nitrogen Oxide Emissions

The nitrogen oxide (NO_x) emissions characteristics of the Mod I engine as a function of fuel flow are shown in Figure 1-8. The emissions are given in terms of the emissions index expressed in grams of pollutant per kilogram of fuel used. The average value of emissions index needed to meet the 0.4 g/mi NO_x emissions requirement is shown for comparison purposes.

Except for three data points, all measured data fall below the required levels. On an average basis, the measured values indicate that the Mod I engine should be capable of meeting the NO_x emissions requirement with some margin.

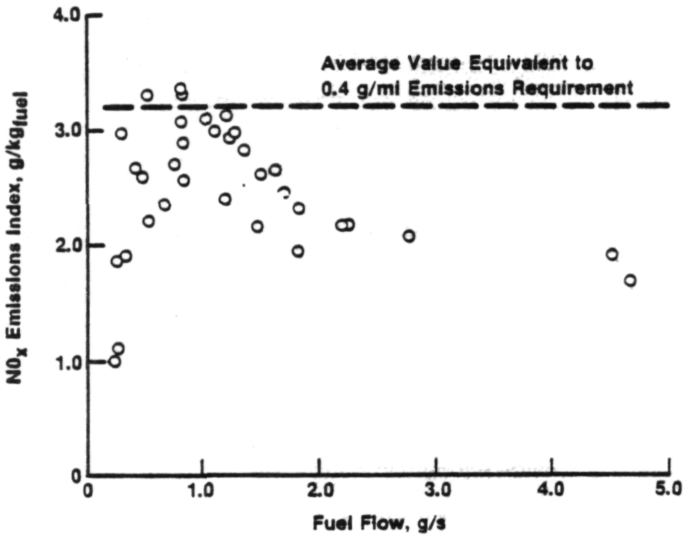


Fig. 1-8 Nitrogen Oxide Emissions

Carbon Monoxide Emissions

Figure 1-9 shows the carbon monoxide (CO) emissions characteristic of the Mod I engine relative to the emissions index needed to meet the 3.4 g/mi emissions requirement. During functional testing of the Mod I, the CO emissions from the Mod I combustor were sensitive to air/fuel ratio, especially when the λ value was below 1.2. Except for one data point, all measured data fall well below that required to meet the CO emissions requirement.

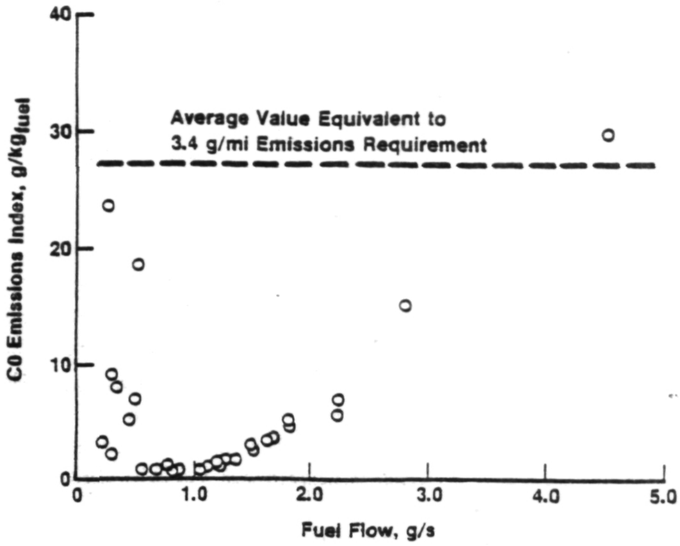


Fig. 1-9 Carbon Monoxide Emissions

Hydrocarbon Emissions

Figure 1-10 shows the hydrocarbon (HC) emissions characteristic of the Mod I engine relative to the emissions index needed to meet the 0.41 g/mi emissions requirement. The steady state HC emissions from the Mod I combustor are very low; however, the primary contribution to HC emissions is known to be the ignition and early warm-up of the combustion system (an area not evaluated in the engine dynamometer tests reported here). The measured HC emissions are encouraging since they are an order of magnitude less than the emissions index needed to meet the HC emissions requirement.

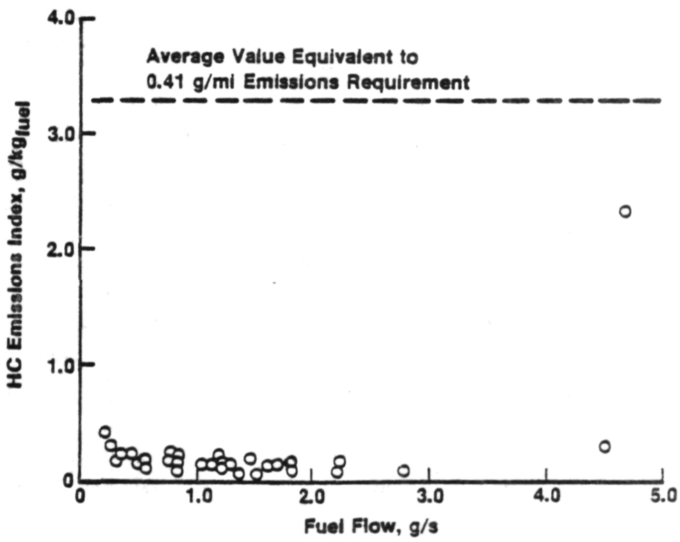


Fig. 1-10 Hydrocarbon Emissions

U.S.A. Mod I Manufacturing

The major purpose of this subtask is to manufacture, assemble, and test a Mod I engine in the United States. The manufacture of the Basic Stirling Engine (BSE) (less controls and auxiliaries) was initiated in August, 1980. With procurement overlapping actual Mod I European engine testing, attempts were made to obtain the most recent hardware, incorporating updates of original hardware based on engine testing in Sweden.

The drive unit for the basic engine was procured from U.S.A. vendors, using raw castings from Europe when necessary. The only items not yet delivered to MTI are the drive gears. Figures 1-11 through 1-17 are photographs of current in-house major drive unit components.

The partially completed U.S.A. Cold Engine System (CES) features 430 stainless steel liners in place of the cast-iron USSw design, and Inconel 718 piston domes (deep-drawn caps) in place of the machined USSw Nimonic 80 domes. Major CES components manufactured in the U.S.A. are shown in Figures 1-18 to 1-21.

The External Heat System (EHS) is partially complete at this time. A metallic-plate assembly preheater will not be manufactured because of its high-volume production costs; therefore, a major effort will be made to fabricate a ceramic preheater. The other major component of the EHS, the combustor (shown in Figure 1-22), has been fabricated and is in-house.

The Hot Engine System (HES) has one major component, heater head quadrants (four per engine), which consist of an investment-cast regenerator/cylinder housing. During this report period, MTI received nine regenerator castings and two cylinder housing castings from the U.S.A. vendor.

The castings delivered to date are currently in use on the operable Mod I engine in Sweden because of a failure of USSw's European vendor to supply good-quality castings. Of the remaining quadrant items, heater tubes and fins, the fins are currently in-house; the tubes are expected in March, 1982. Figures 1-23 and 1-24 are photographs of the regenerator and cylinder housing.*

Buildup of the U.S.A. Mod I Engine Drive Unit is expected to begin in January of 1982. Delivery of other items not yet in-house is expected in early 1982, with testing scheduled for late 1982.

In the coming year (1982), the following events are scheduled:

- the Mod I Lerma test bed will become operational with Mod I engine #1;
- present Mod I engine hardware will be upgraded to obtain:

  PWR = 58 kW (78 hp),

  power-to-weight ratio = 7.5 lb/hp,

  BSFC = 0.355 lb/hp-hr; and,
- MTI will receive Mod I engine #3 for endurance testing.

*The housing shown on the left side of Figure 1-23 is a casting, and the housing shown on the right side is a wax mold; the housing shown in Figure 1-24 is a casting.

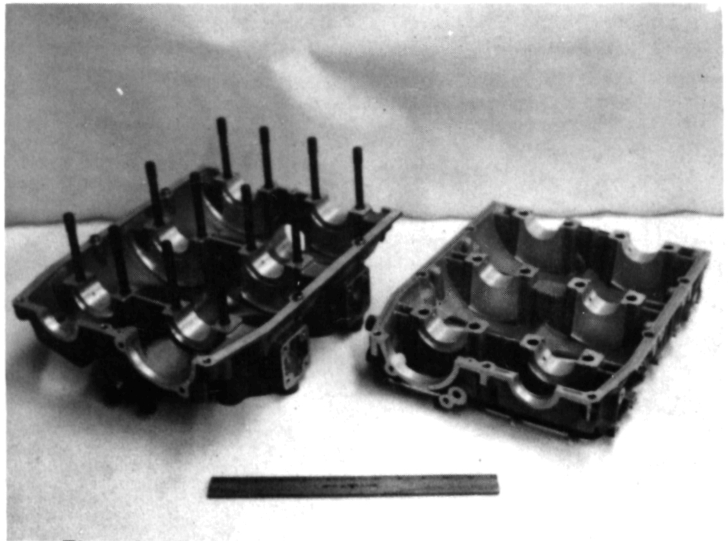


Fig. 1-11 Crankcase/Bedplate Assembly

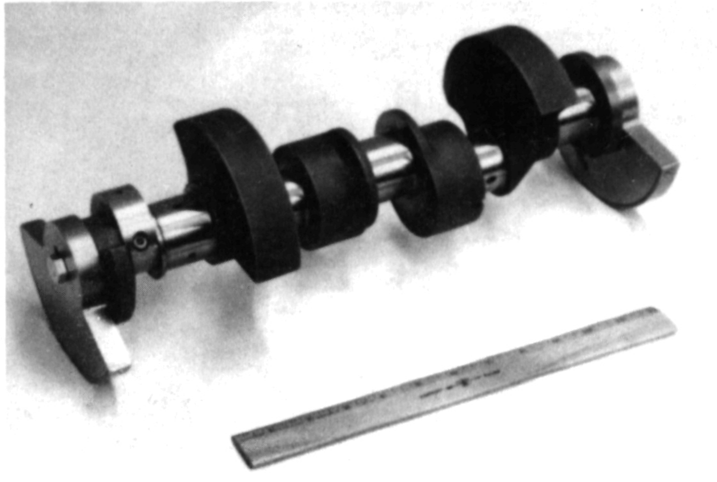


Fig. 1-12 Crankshaft

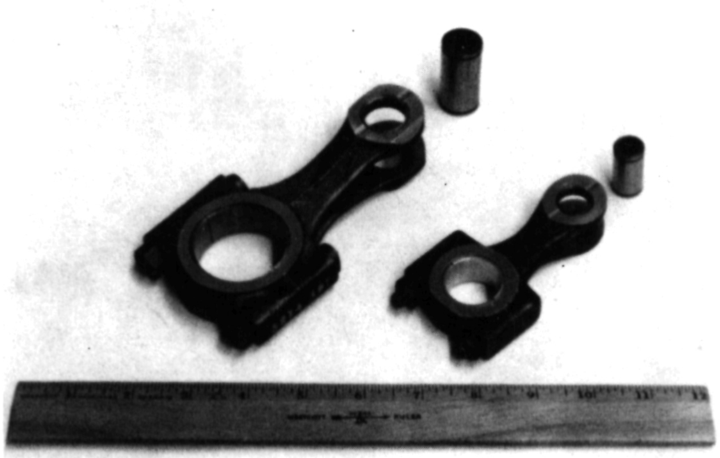


Fig. 1-13 Connecting Rods

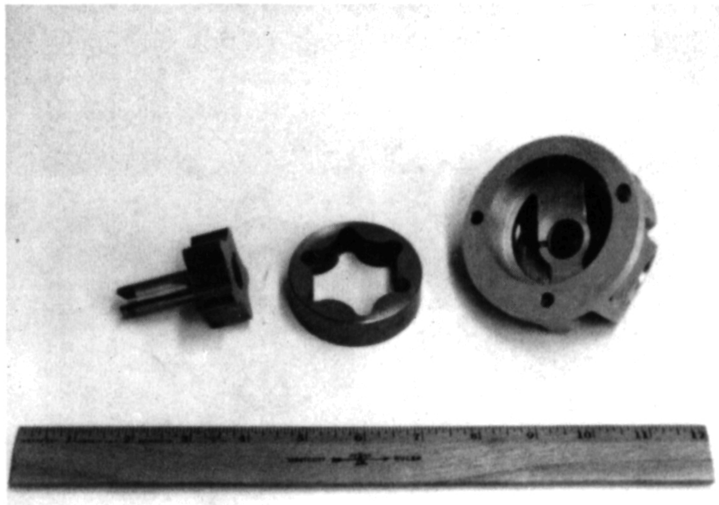


Fig. 1-14 Lube Oil Pump Assembly

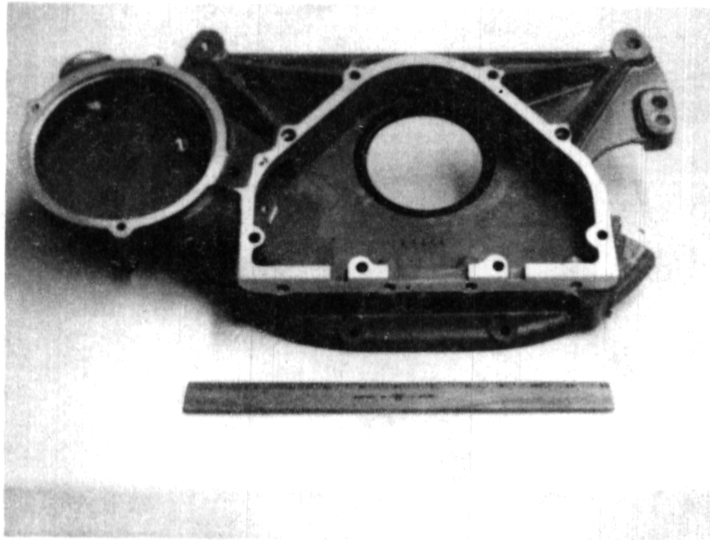


Fig. 1-15 Intercasing

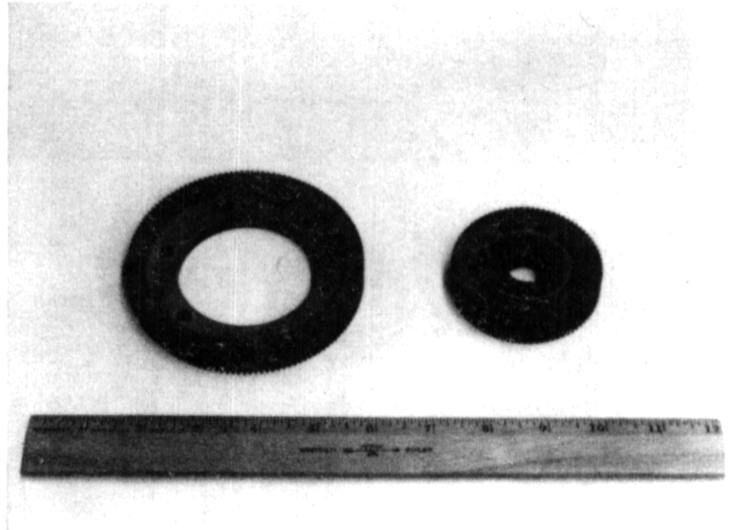


Fig. 1-16 Plastic Gears

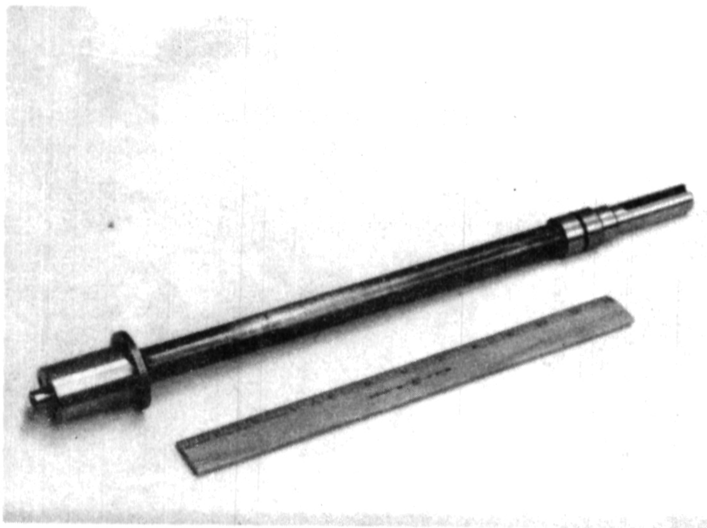


Fig. 1-17 Main Drive Shaft

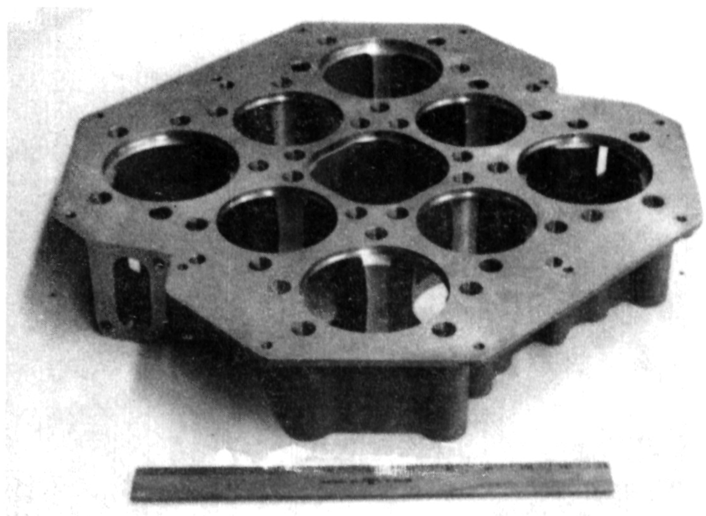
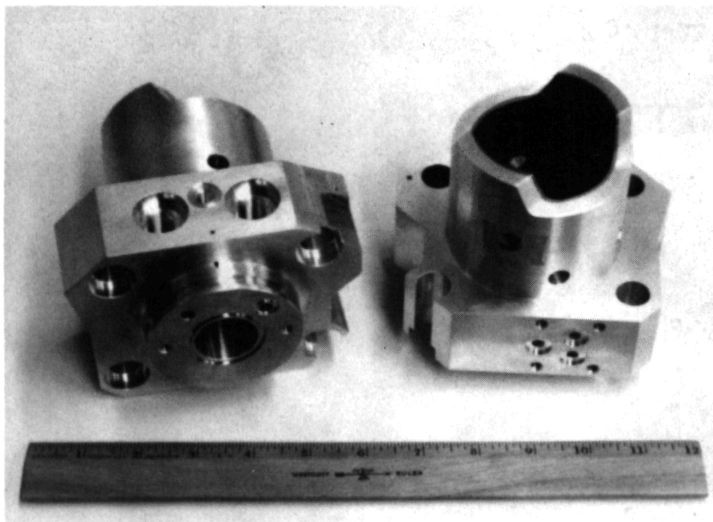


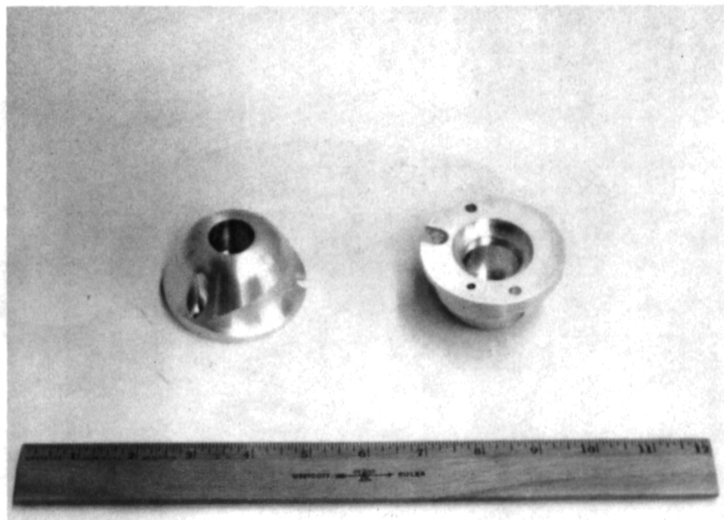
Fig. 1-18 Water Jacket

Fig. 1-19 Piston Dome

Fig. 1-20 Piston Bases



Guides



Covers

Fig. 1-21 Crosshead Guides and Covers

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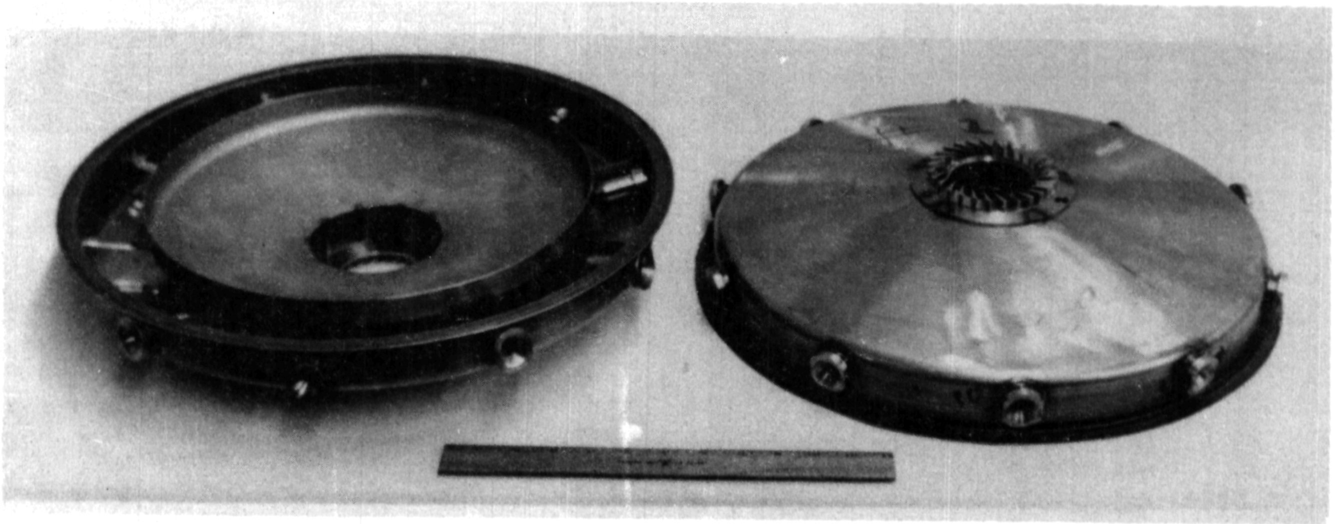

Fig. 1-22 CGR Combustor

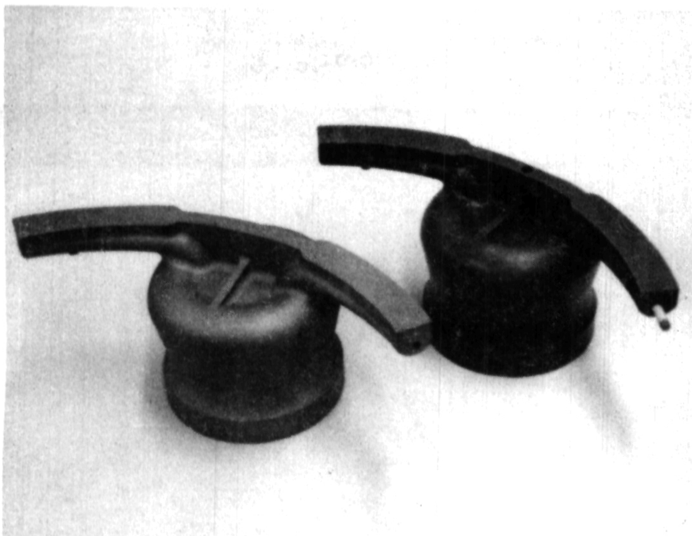

Fig. 1-23 Mod I Regenerator Housings

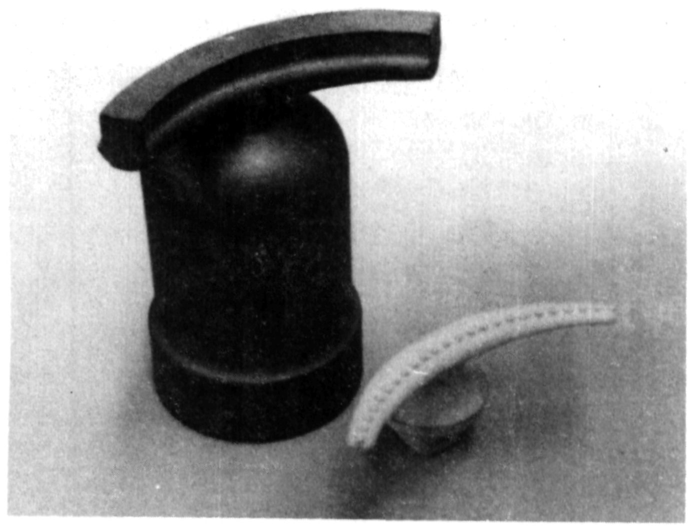

Fig. 1-24 Mod I Cylinder Housing and Internal Ceramic Core

II. P-40

Technology Familiarization (Baseline Engine)

The objective of the technology familiarization effort was to test the P-40 engine* on a dynamometer and in a vehicle in order to familiarize U. S. program participants with engine/component operating characteristics, and with problems associated with vehicle installation.

During this semiannual report period, the major focus of this task was on engine dynamometer testing to gain experience with the Mod I Digital Control System, and to assess the impact of alternative fuels on engine performance and emissions. Vehicle testing of ASE P-40-8 was discontinued, and the above effort completed the testing under this task. The Stirling engine test facilities will continue to be upgraded and maintained during the program. An overall summary of the operating time on ASE P-40-7 is given in Figure 2-1.

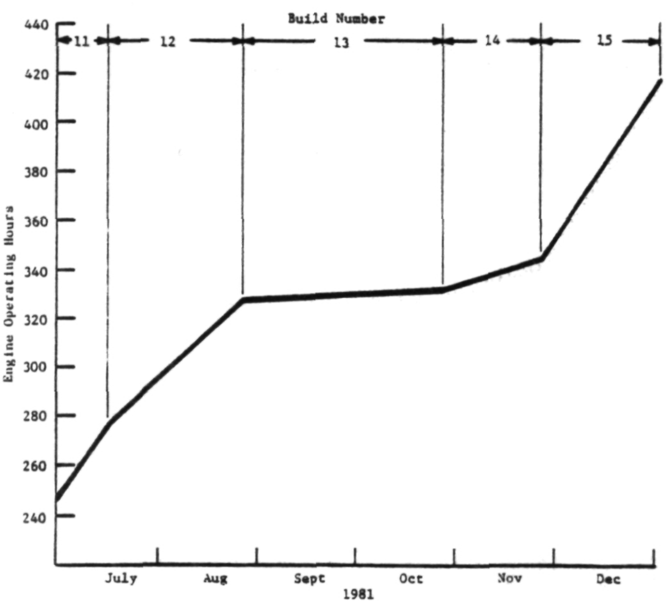

Fig. 2-1 ASE P-40-7 Operating History

*At the start of the ASE Program, the P-40 engine represented baseline Stirling engine technology at USSw.

Analog/Digital Control System Testing

A new microprocessor-based Digital Control System has been developed for the Mod I engine. In order to become familiar with this system, back-to-back tests were run on ASE P-40-7 with both the typical P-40 Analog Control and the Mod I Digital Control Systems. In general, transient and steady state operation with the Digital System compared favorably with that of the Analog System. A detailed description of the test results is contained in Section III. At the completion of control system testing, the engine was disassembled and a "dirty" inspection was held. The engine was found to be in good general condition; however, discrepancies included:

- extensive rust contamination in working gas tubing;
- damaged threads on the piston base (#3 cylinder);
- preheater sheet metal cracks; and,
- red deposits on heater head fins (later analyzed as oxides of iron).

In order to build up operating time, the seal housing was not disassembled.

Multifuels Testing

ASE P-40-7 was rebuilt in order to perform testing that would measure the effect of operating a Stirling engine with a number of alternative fuels. Discrepancies found during Builds 13 and 14 were addressed in the following manner:

- working gas tubing was coated with electroless nickel to prevent future oxidation;
- #3 cycle piston was replaced with a spare; and,
- a new preheater was installed.

In addition, P-40-7 was rebuilt with reworked heater heads, where cylinder housing web material was removed to relieve stress in this area. After rebuild #15, the engine was successfully operated with the following alternative fuels:

- unleaded gasoline;
- shale-oil-derived marine-diesel;
- ET3 (10/90% alcohol/gasoline blend);
- ERBS (experimental referee-broadened specification aviation turbine fuel); and,
- commercial diesel.

During testing, the engine's performance and emissions characteristics were recorded for EGR/non-EGR operation, and start-up and steady state tests were conducted for each fuel. The emissions results are discussed in detail in Section III. In general, operation on all fuels was acceptable, with the exception of cold starts with diesel fuel (slow and smokey).

Baseline and final checkout tests were run with unleaded gasoline to determine if any significant performance deterioration had occurred during testing. Figure 2-2, a comparison of power at maximum pressure (15 MPa) for each of these runs, indicates that maximum power output dropped about 2 kW (2.7 hp) during the course of multifuels testing.

Engine power and efficiency at maximum pressure are plotted for each fuel in Figures 2-3 and 2-4. Note that when compared to Figure 2-2, performance did not vary significantly with any of the fuels tested. This was expected, since engine cycle operating conditions (heater temperature, mean pressure, etc.) were held constant during each test, and any differences would only be due to the affect of the different fuels on external heating system efficiency.

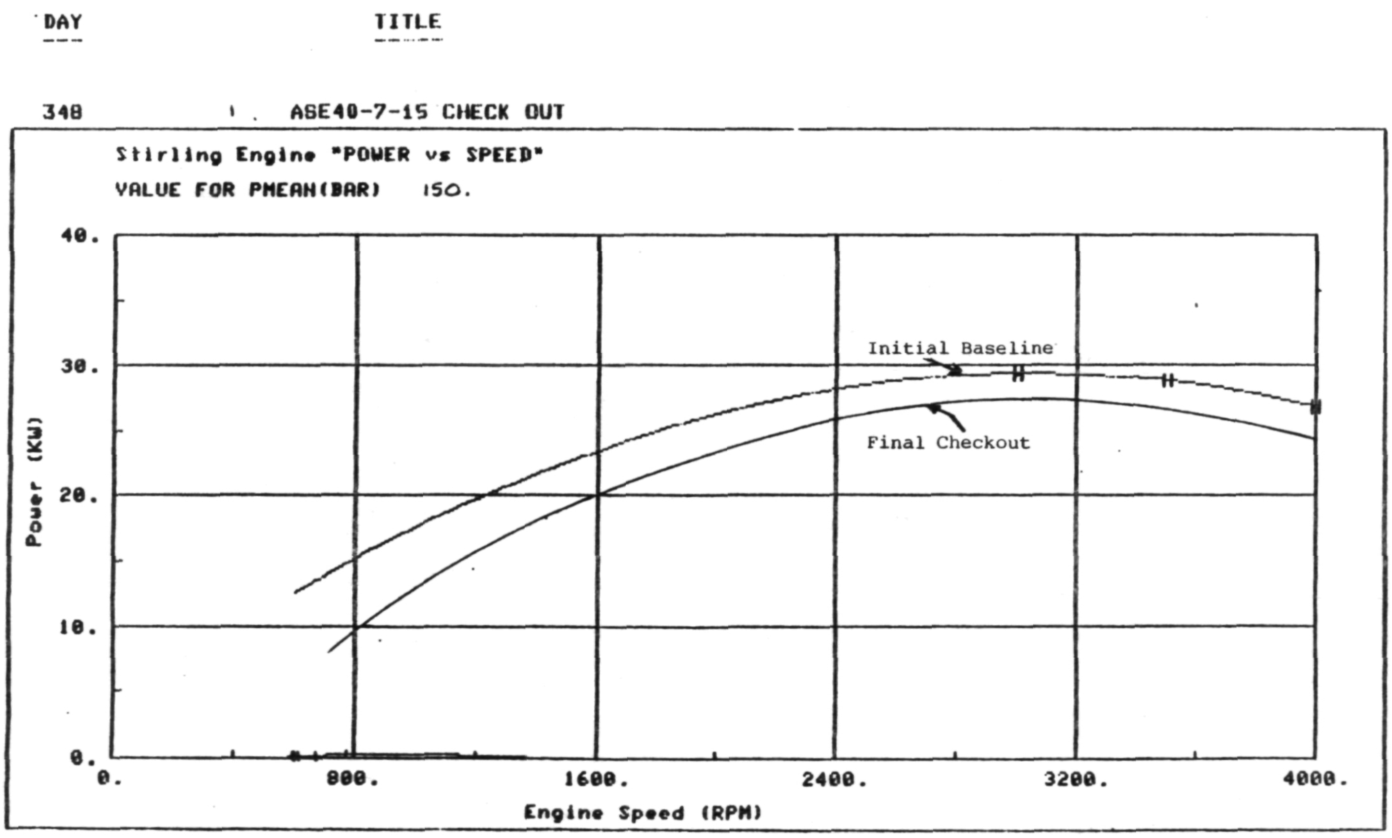

Fig. 2-2 Baseline/Final Checkout Comparison

DAY	TITLE	LABEL
349	ABE40-7-15 EMIS/GAS	+
351	40-7-15 DIESEL/SHALE	*
356	40-7-15 EMIS/ET-3	X
357	40-7-15 EMIS/ERBS	H
363	40-7-15 EMIS/DIESEL	#

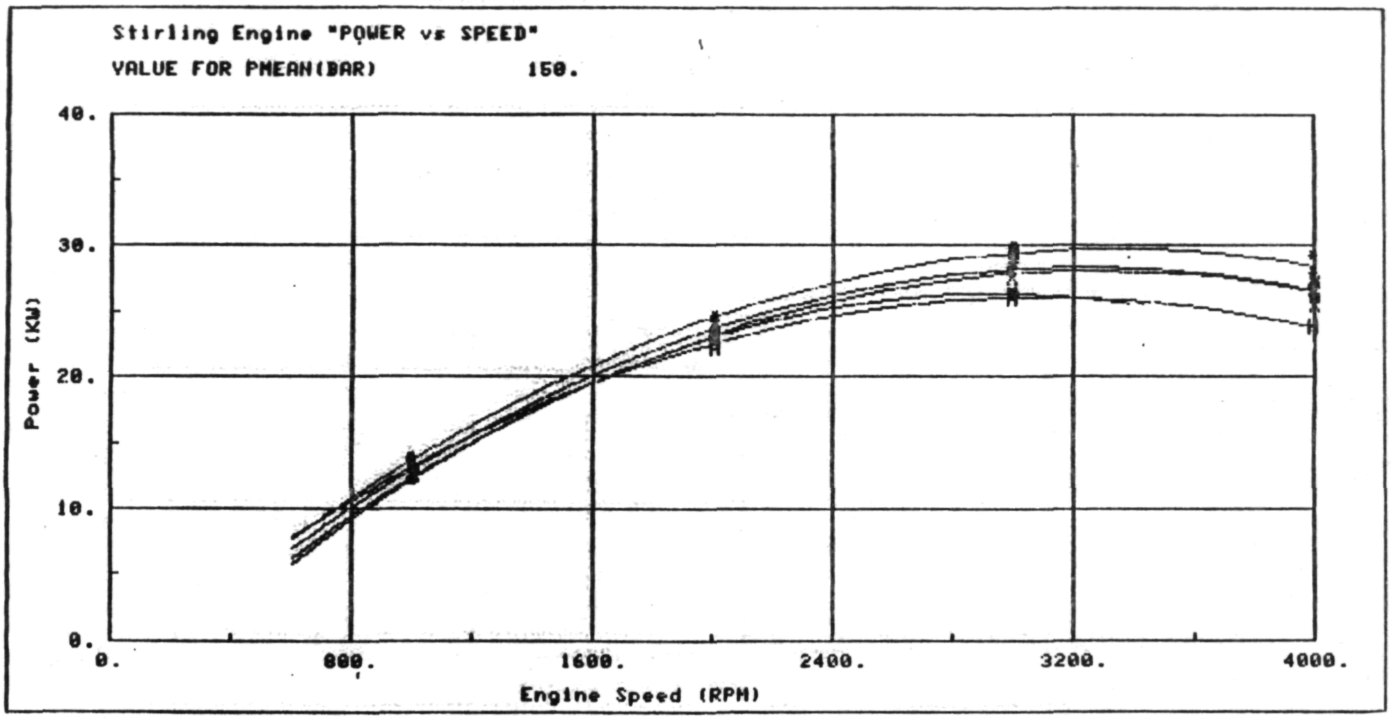

Fig. 2-3 Alternate Fuel Power Comparison

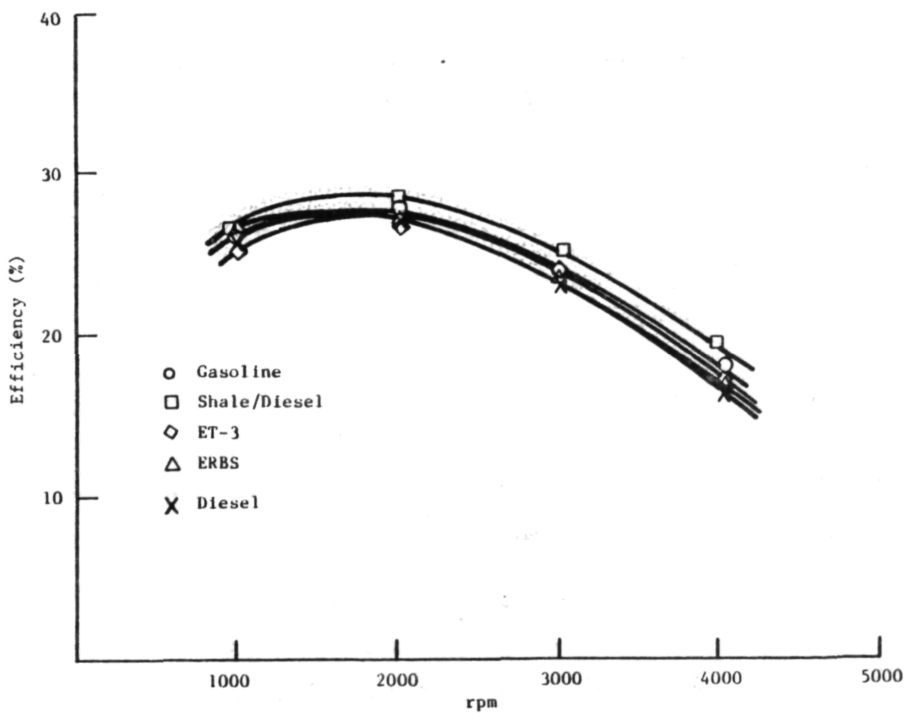

Fig. 2-4 Engine Efficiency for Alternate Fuels

As shown in Figure 2-5, the engine's overall performance level was down during multifuel testing. Since the objective of the testing was to obtain a relative comparison between operation with different fuels, the decrease in performance should not have impacted the test results or conclusions. After the completion of testing, the regenerators were removed from the engine and flow tested to measure pressure drop. Figure 2-6 shows that the measured pressure drop was significantly higher than for the new regenerators. Preliminary calculations indicate that the effect of this difference in pressure drop explains the loss in power indicated in Figure 2-5.

The reworked heater heads were inspected at the completion of testing, with no indications of cracking in the web area. In addition, the electroless nickel coating was successful in preventing rust buildup in the engine.

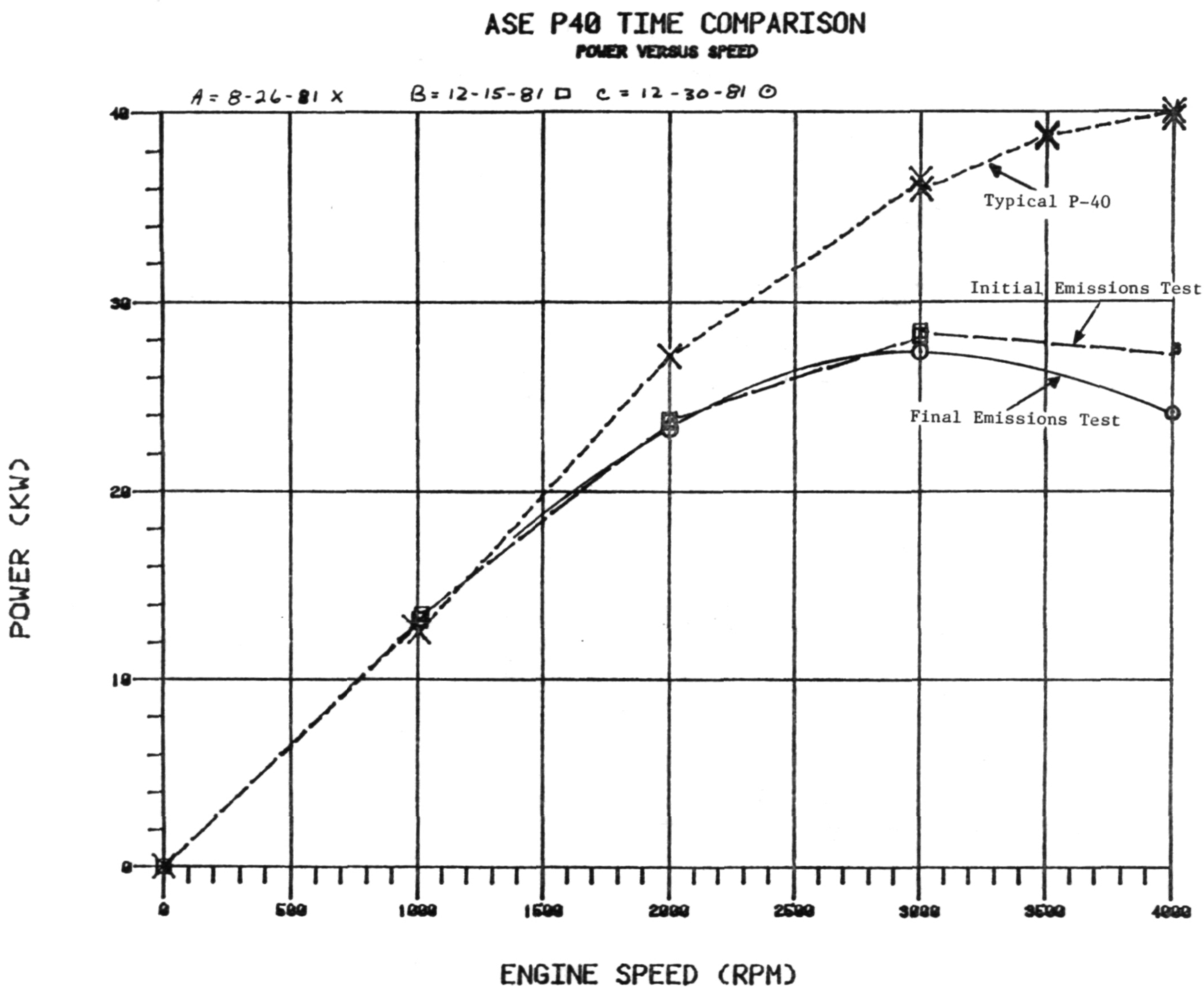

Fig. 2-5 ASE P-40-7 Time Comparison Power Versus Speed

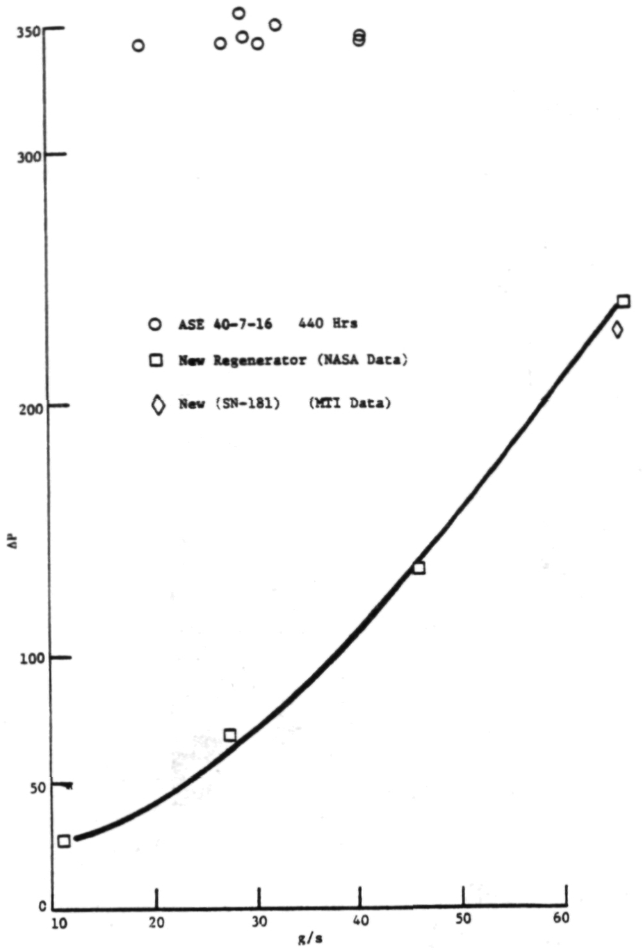

Fig. 2-6 Regenerator Flow Test

Stirling Engine Test Facilities

During this semiannual report period, a number of improvements were made to upgrade MTI's Stirling engine/component test facilities:

- Engine Test Cell (Figures 2-7 and 2-8) - The data acquisition system plotting/storage capabilities were improved. The fuel cubicle, which supplies and measures fuel from three alternative tanks (two in ground, one in fuel cubicle), was also completed.

- Emissions Measurement System - Figure 2-9 illustrates the emissions measurement equipment which was checked out during alternative fuels testing on ASE P-40-7.

- Motoring Cell - The Engine Drive System Motoring Cell (Figures 2-10 and 2-11) was completely checked out.

- Seals Rig (Figures 2-12 and 13) - The rig drive system was upgraded from an electric drive to a hydraulic drive system to increase low-speed torque.

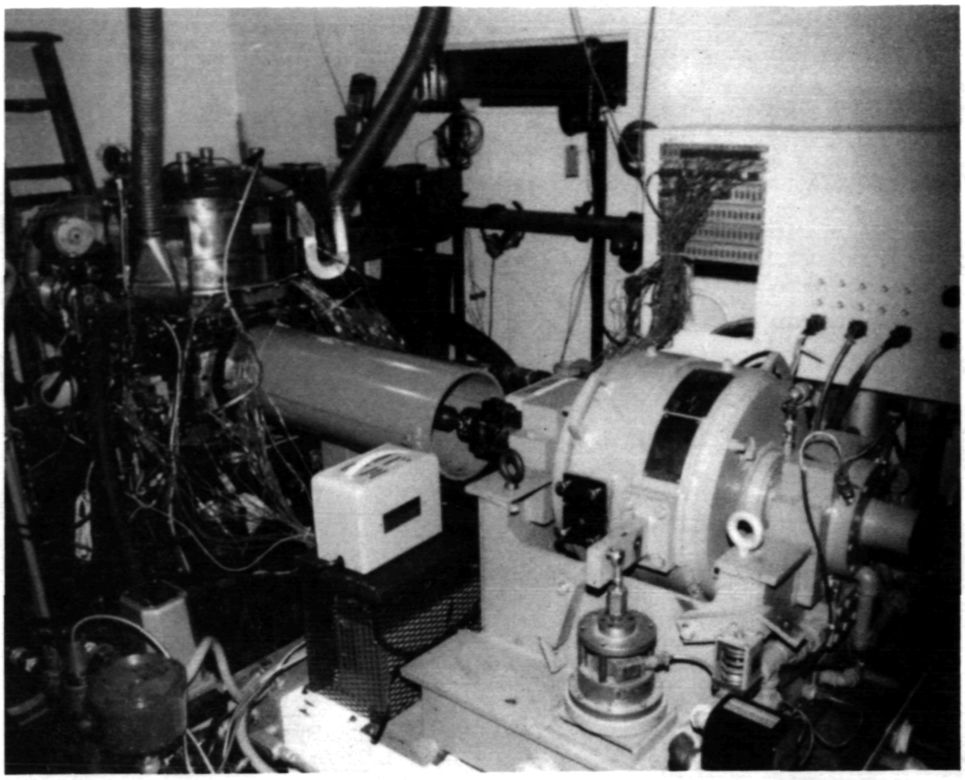

Fig. 2-7 P-40 Engine in Test Cell

Fig. 2-8 Engine Control Room

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Fig. 2-9 Emissions Measuring Equipment

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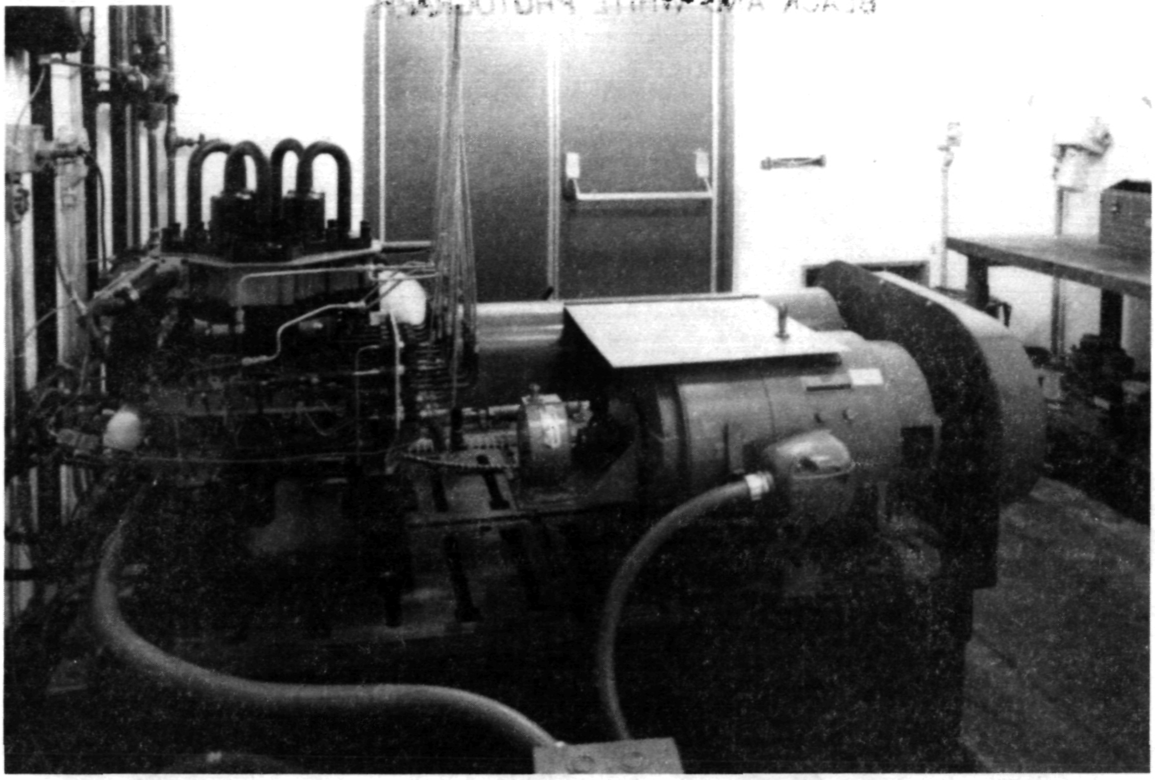


Fig. 2-10 Motoring Cell Rig

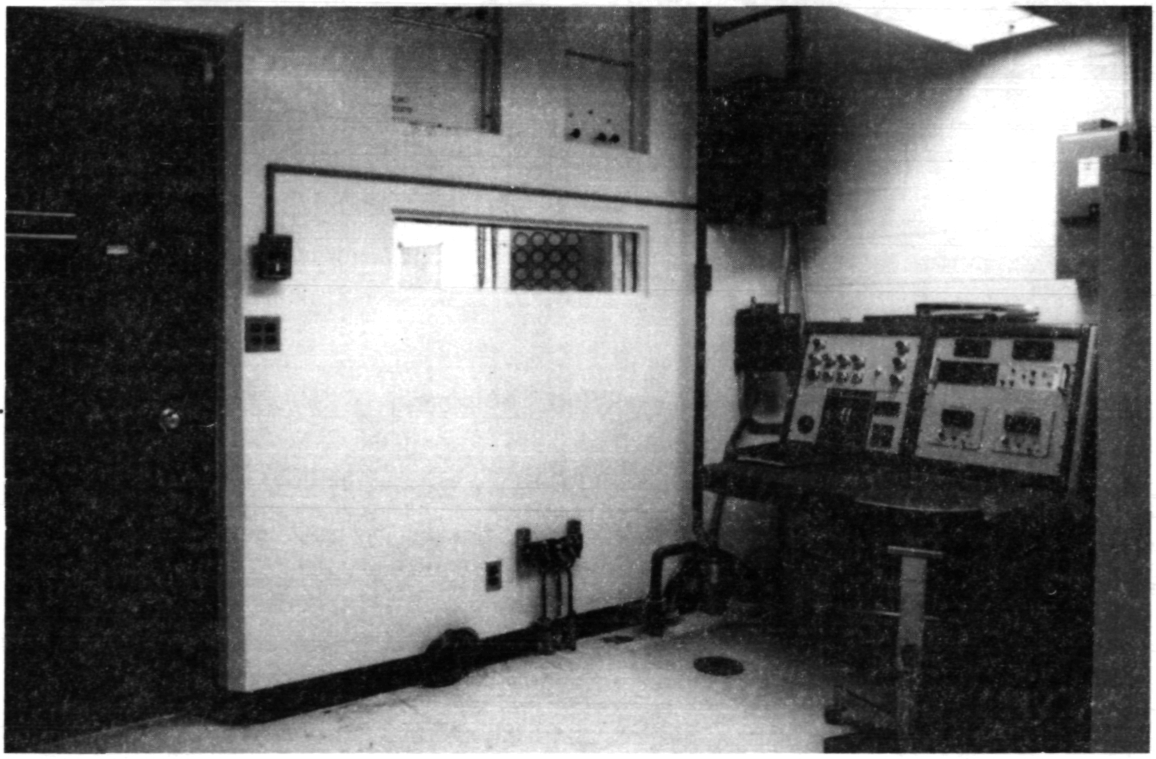


Fig. 2-11 Motoring Rig Control Room

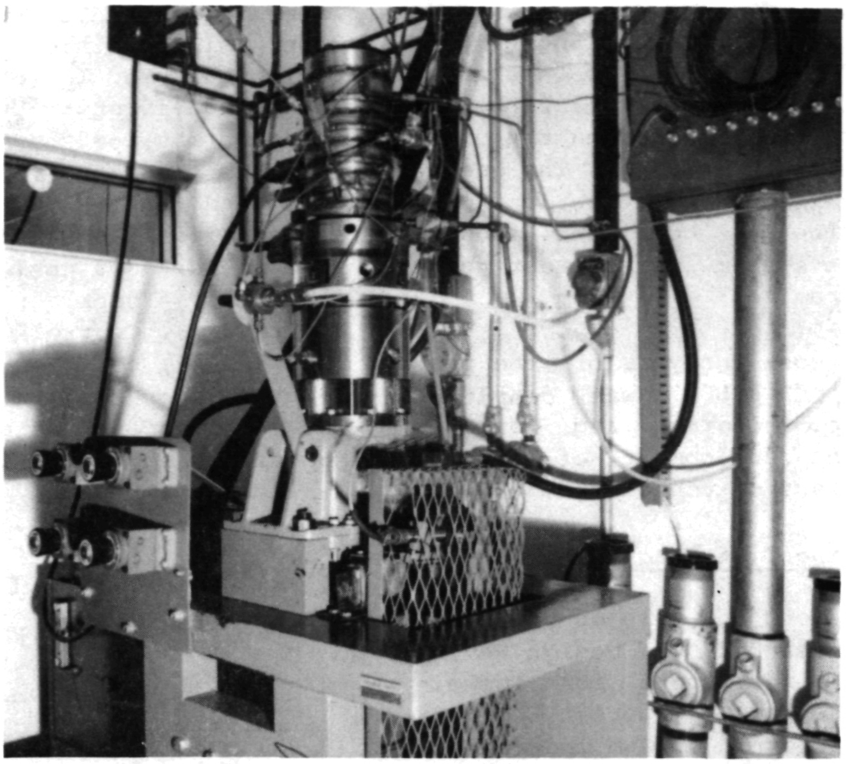


Fig. 2-12 Seals Rig

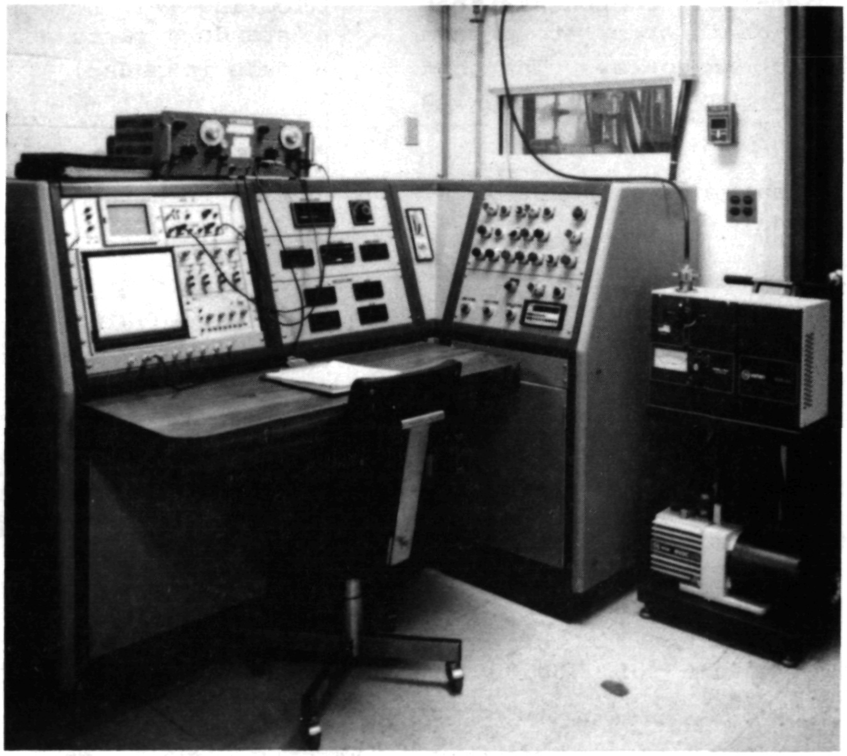


Fig. 2-13 Seals Rig Controls

III. REFERENCE ENGINE SYSTEM DESIGN

The objective of the Reference Engine System Design (RESD) task is to design a mass-producible engine that will represent the optimum approach for meeting the overall ASE Program objectives. The engine design effort will be used as a focal point for guiding all component, subsystem, and engine system development tasks within the program.

A manufacturing cost analysis of the previously developed RESD* was completed during this report period, and work was initiated on the development of an RESD for a smaller class of vehicles.

RESD Manufacturing Cost Study

A manufacturing cost analysis of the 60-kW (80-hp) RESD (presented at a Design Review in March, 1981), developed for installation in a General Motors "X"-Body vehicle, was performed by an independent automotive-industry consultant to compare its cost to that of a current-production spark-ignition engine. A second objective was to identify high-cost components that may need modification in order to attain the overall ASE Program goals. The cost study was performed under the following guidelines:

- the standard automotive industry costing methodology;
- 1981 automotive industry direct labor rates;
- manufacturing burden rates typical of current automotive industry practice;
- manufacturing (transfer) cost defined as the sum of materials, labor, scrap allowance, and manufacturing burden; and,
- a production volume of 300,00 units per year.**

*Automotive Stirling Reference Engine Design Report DOE/NASA/0032-12; NASA CR-165381, June 1981

Since detailed drawings of the RESD engine were not available, the manufacturing cost of a value-engineered version of the experimental Mod I Stirling engine was determined first. The cost of the RESD was then obtained by multiplying the Mod I cost by the ratio of the weights of the RESD and the Mod I engine. The basic engine weights projected for the Mod I and RESD were 413.1 lbs. and 265.2 lbs., respectively (both engines are shown in Figure 3-1). Controls and auxiliaries were not scaled, so their costs are the same for both the RESD and Mod I. The capital and tooling costs established for the RESD engine design are summarized below in Table 3-1:

TABLE 3-1

CAPITAL AND TOOLING COSTS

Item	Cost
Building ($80/ft^2) (905,000 ft^2)	$ 72,400,000
Tooling (all but standard parts made in-house)	$ 82,288,600
Equipment (all but standard parts made in-house)	$292,921,600

The cost study was broken down into five distinct engine subsystems:

- External Heat System;
- Hot Engine System;
- Cold Engine System;
- Drive Unit; and,
- Controls and Auxiliaries.

Layouts of these subsystems, and a summary of their features and costs, are shown in Figures 3-2 to 3-5 and in Table 3-2.

**considered to be representative of a high-volume productive facility that can be sustained on a two-shift basis.

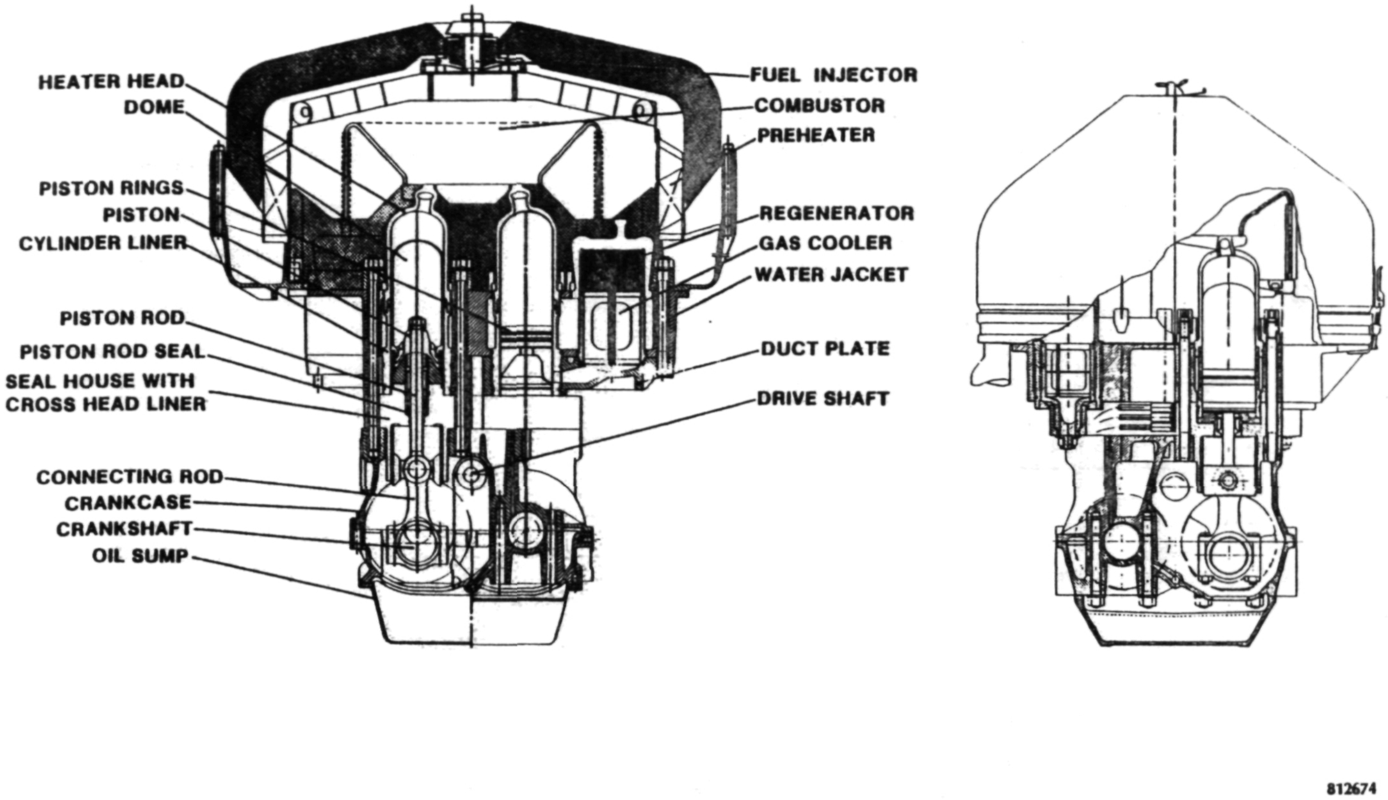


Fig. 3-1 Mod I (left) and Reference Engine Designs (right)

TABLE 3-2

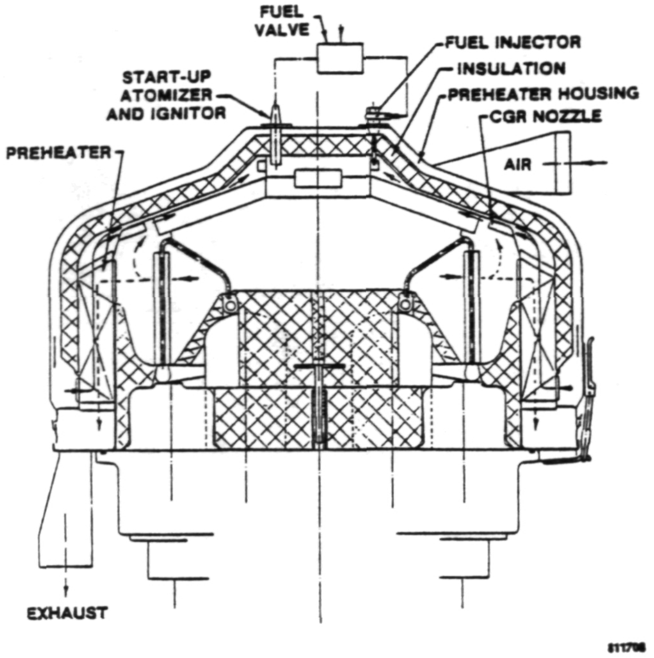
CONTROLS AND AUXILIARIES

Features

- Electronic Control
- Two Pressure Transducers
- Three Displacement Transducers
- Air Pump
- Fuel System
- Alternator
- Power Steering
- Air Conditioning
- Power Brakes
- Mean Pressure Control System
- Hydrogen Compressor and Storage Tank
- Starter, Water Pump, and Oil Pump

1981 RESD Cost

$543.32



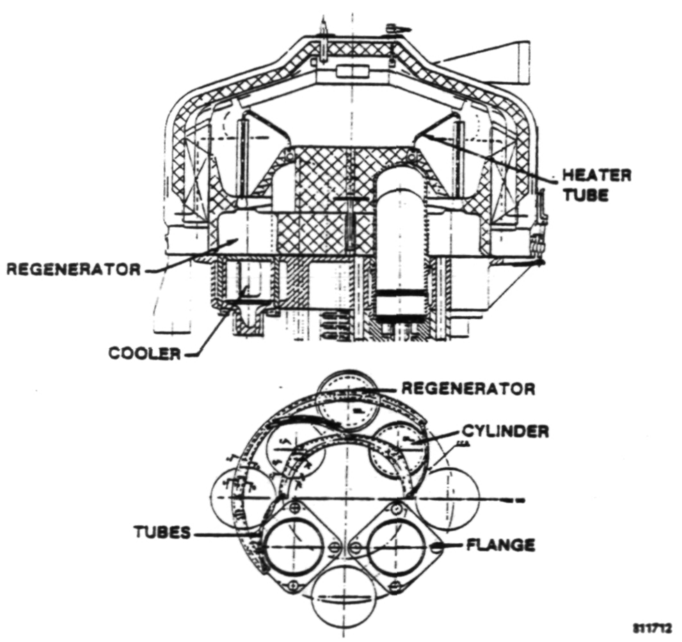
Features

- Ceramic "Z"-Flow Recuperator
- Ceramic Combustor Lower Cone, Remainder Stainless Steel
- Remaining System Parts Aluminum and Plain Steel Sheet Metal

1981 RESD Cost

$186.69

Fig. 3-2 RESD External Heat System



Features

- XF-818 Heater Head Regenerator and Cylinder Housings
- Inconel 625 Heater Tubes
- Single-Piece Fins
- "Steel Wool"-Type Regenerators with 1 to 2% Engine Efficiency Penalty
- Plain Steel Gas Coolers - Galvanized for Corrosion Resistance

1981 RESD Cost

$395.85

Fig. 3-3 RESD Hot Engine System

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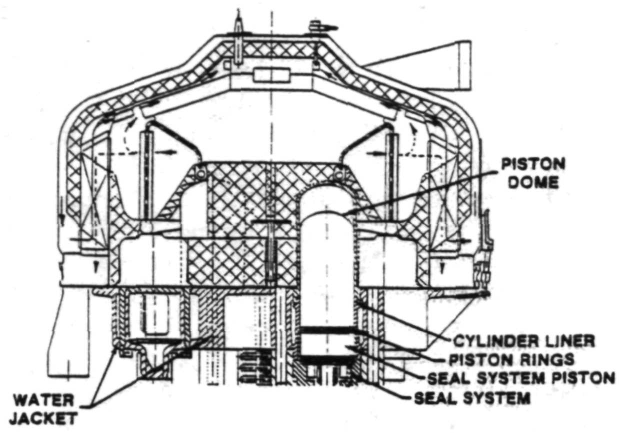


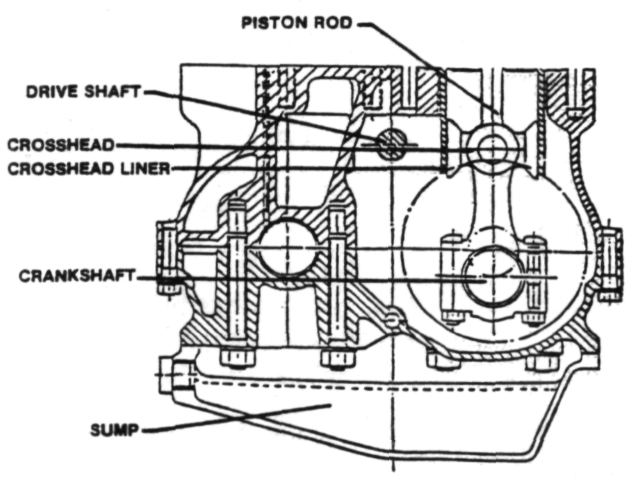
Features

- Inconel 718 Capped Pistons
- Cast-Aluminum Water Jacket/Duct Plate
- Cast Piston Rods
- Cast-Iron Cylinder Liners.

1981 RESD Cost

$109.43

Fig. 3-4 RESD Cold Engine System



Features

- Integral Crankcase/Crosshead Guide Aluminum Casting
- Cranks
- Connecting Rods.

1981 RESD Cost

$191.07

Fig. 3-5 Engine Drive System (End View)

Total manufacturing cost of an initial production run of the RESD engine in 1981 dollars is projected to be $1,426.37. Using inflation rates of 8.5% in 1982, 8% in 1983, and 8.5% in 1984, the engine would cost $1,813.49 in 1984; the projected cost for an equivalent spark-ignition engine system* in 1984 is $936.85.

*consisting of engine, emission controls, and catalytic converter

Reduced Size RESD Study

Current/projected trends indicate that the size of automobiles will continue to reduce, with subcompact vehicles representing a major segment of the future market. So, in addition to the previous RESD design study (which concentrated on larger vehicles), a study has been initiated to investigate reduced size RESD design concepts. The purpose of this effort was to investigate the applicability of the technology being developed at the 60-kW (80-hp) level to that of lower power engines. Specifically, will the lower power engines require technologies not currently being generated for the larger 60-kW (80-hp) engines? General design goals for this study are shown below in Table 3-3.

TABLE 3-3

REDUCED SIZE RESD GOALS

Initial Production Year	1990-1995
Vehicle Class	Subcompact Front-Wheel Drive
Basic Vehicle Test Weight	2250 lbs
Engine Power	45 kW (60 hp)
Vehicle Acceleration (0-60 mph)	15 seconds
Engine Specific Weight	4.35 lbs/hp
Stirling Power System** Cost	Same as Equivalent Diesel-Power System
Vehicle Fuel Economy	
• Combined Cycle	> 1.3 x SI Engine - 69 mpg
• Urban Cycle	> 1.3 x SI Engine - 59 mpg
• Highway Cycle	> 1.35 x SI Engine - 86 mpg
Exhaust Emissions (Urban Cycle)	
• HC	< 0.41 g/mi
• CO	< 3.4 g/mi
• NO_x	< 0.4 g/mi
• Total Particulates	< 0.2 g/mi
Primary Fuel	Unleaded Gasoline
Multifuel Capability	Yes
Engine Volume	Packaged in current subcompact engine compartment with modified hood line (increased slope)

**includes engine, all auxiliaries, emissions control, hardware, transmission, battery, and cooling system

IV. COMPONENT DEVELOPMENT

The original objective of the Component Development task was to improve performance, producibility (reduced cost), and durability/reliability of the combustion system, heat exchangers, seal systems, drivetrain, control systems, and auxiliaries in support of engine development. Material development was aimed at identification/substantiation of low-cost, nonstrategic alternate materials. This was to be accomplished through the Mod I/Mod II engine designs, as guided by the RESD; however, budget restrictions required a redefinition of these tasks during the last half of 1981, replacing Mod II development with Upgraded Mod I (UMI) development as "proof-of-concept" for the RESD. Further, component development has been reorganized on an engine subsystem basis, with the following areas of emphasis:

- External Heat System Development - combustor, fuel nozzle, igniter, and preheater.
- Hot Engine System Development - heater head and regenerators.
- Materials Development - focus on heater head casting and tube materials.
- Cold Engine System Development - piston ring, main seal and capseal systems, piston domes, and cylinder liner.
- Engine Drive System Development - crankcase, crankshaft, bearings, and connecting rods.
- Control System Development - mean pressure control, combustion control, temperature control, and microprocessor-based electronic control.

Mod I component performance characterization and initiation of UMI component design and testing have been the focal point of current development activities. These tasks were accomplished through substantial technology transfer and accelerating utilization of test facilities in the United States.

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

EXTERNAL HEAT SYSTEM (EHS)

The foremost goal for the EHS is low emissions while maintaining high efficiency for a 20:1 fuel turndown ratio. The design must consider the expected use of alternate fuels, and recognize the significant cost impact of system size/design.

Development activity during the last half of 1981 focused on rig testing, evaluation and improvements of Mod I ignition technique, preparations for UMI combustor testing, a parametric rig study of six alternate combustor configurations, emissions testing of five alternate fuels* in an engine combustor, and analysis of preheater performance/size tradeoffs.

The primary goals during 1982 are to design, fabricate, and test a reduced size/cost EHS (including an upgraded combustor and preheater) requiring the evaluation and selection of a nonair-atomized nozzle with complimentary igniter and blower.

Ignition Testing

The requirement of the combustion system to ignite rapidly is essential to meeting CVS cycle emissions goals. Failure to light within a specified time results in hydrocarbon (HC) emissions that will exceed the goal of 0.41 g/mi. At the current ignition fuel flows/airflows (20-30 g/s of air; 1.0 or more g/s of fuel), ignition must occur within $\sim$.4 seconds (see Figure 4-1) to meet the emissions goal.

The ignition system was tested to first obtain an ignition map for the current Mod I ignition system (fuel/nozzle/igniter

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*unleaded gasoline, shale-oil derived marine-diesel, ET3 (gasohol), ERB (experimental referee-broad-cut aviation fuel), commercial diesel

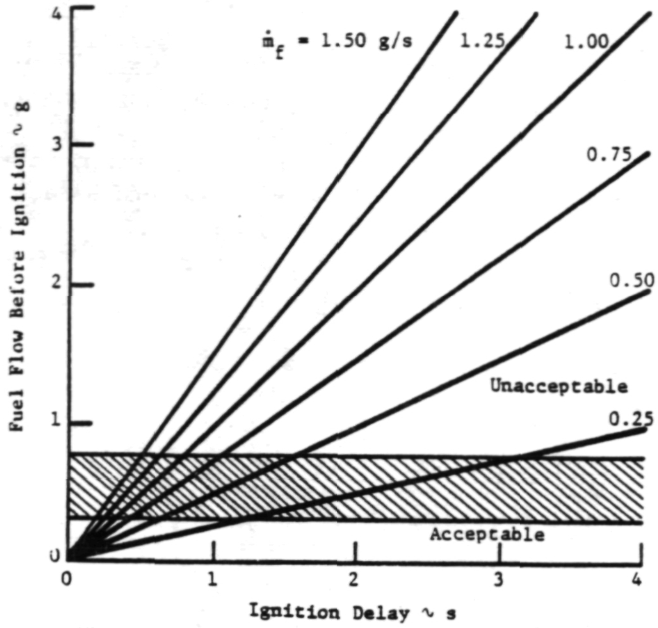


Fig. 4-1 Allowable Ignition Delay of Mod I Engine

combination). This testing determined the affect of combustor airflow, λ $((A/F)/(A/F)_{stoch})$, atomizing air pressure, and combustor bypass position on ignition delay time/blowout. The tests were repeated with three alternate igniters to determine if ignition characteristic improvements could be made. During testing, over 400 data points were run in the Free-Burning Rig (utilizes a Mod I combustor (see Figure 4-2), and simulates the heater head with a perforated screen). A schematic of the test setup is shown in Figure 4-3. Combustor airflow and initial atomizing air pressure is preset. Fuel

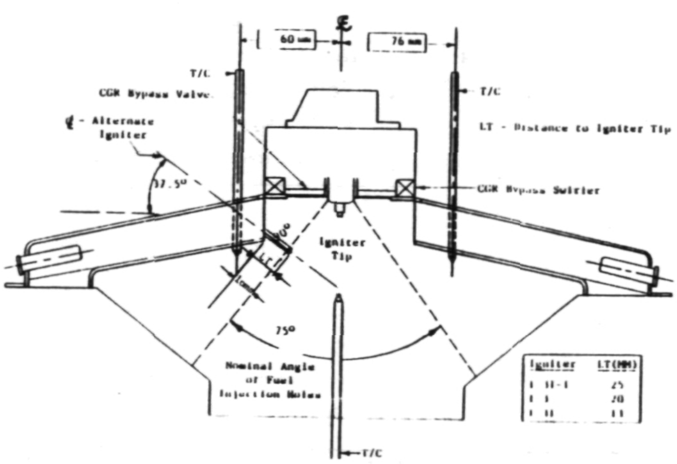


Fig. 4-2 Mod I Combustor Showing Alternate Igniter/Thermocouple Locations

flow is then set with a manual three-way valve in the bypass mode (fuel bypasses the combustor), the spark is initiated, and the fuel is diverted to the combustor.

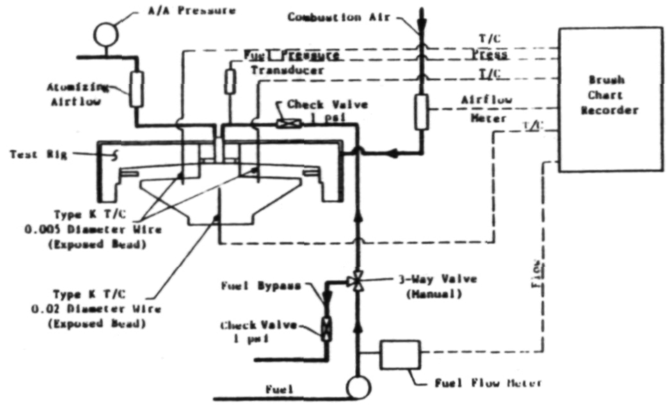


Fig. 4-3 Schematic of Ignition Test Setup

Comparing ignition/blowout limits (Figures 4-4/4-5) of the combustor, one sees that with the bypass closed, blowout limits are very close to the ignition limit; thus, once the combustor ignites, it is difficult to keep it lit when the igniter is turned off*. With the bypass open, ignition/blowout limits widen and, at the lowest atomizing air pressure ratio, are considerably better than those with the bypass closed. The flame also appears stronger and more stable. Optimum airflow for ignition is ~20 g/s, but at a 1.2 atomizing air pressure ratio, ignition times less than 0.2 seconds can be achieved at all airflows tested.

Ignition delay times of all four igniters are compared in Figure 4-6. The comparisons were made for an atomizing air pressure ratio of 1.2 (this gave the best performance for all the igniters). Also shown are the allowable ignition delay times based on the allowable fuel limits of Figure 4-1.

All things considered, adequate ignition delay time can be achieved with the bypass fully open if the atomizing air pressure ratio is reduced (this can be easily

*Igniter life can be greatly improved if it is used only at ignition.

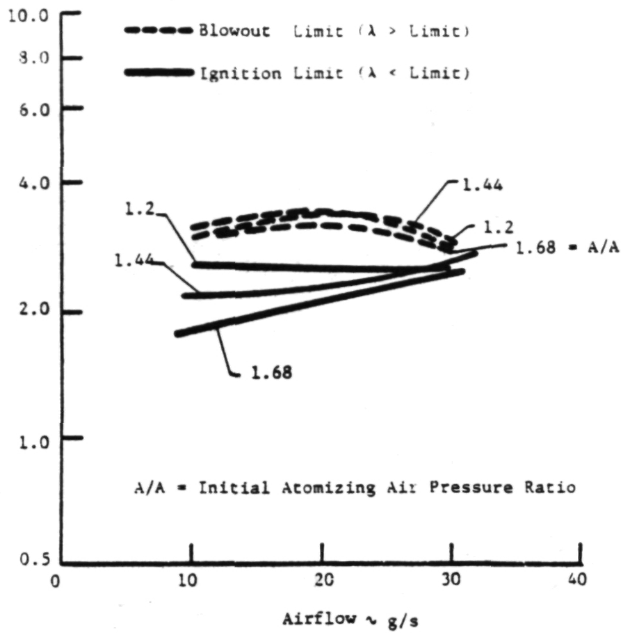


Fig. 4-4 Ignition/Blowout Characteristics of Mod I Combustor (Bypass Closed) USSw Mod I Igniter, Combustor Air Temperature = 12-13°C

accomplished by bypassing some of the atomizing air. In order to have optimal ignition performance in the Mod I engine, the following recommendations are made:

- continue to use the current Mod I igniter location;
- ignite with CGR bypass valve open;
- reduce atomizing air pressure ratio at ignition to 1.2; and,
- set combustor airflow between 20 and 25 g/s at ignition.

The successful completion of these tests provides all the background knowledge needed to design the UMI igniter.

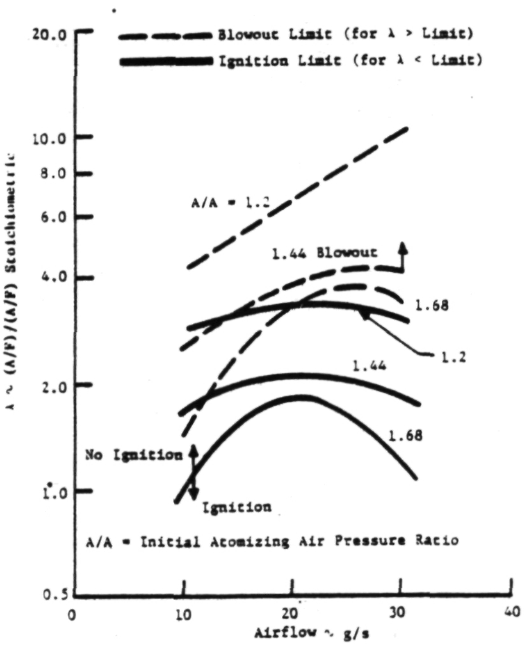


Fig. 4-5 Ignition/Blowout Characteristics of Mod I Combustor (Bypass Open) (USSw) Mod I Igniter, Combustor Air Temperature = 15-20°C

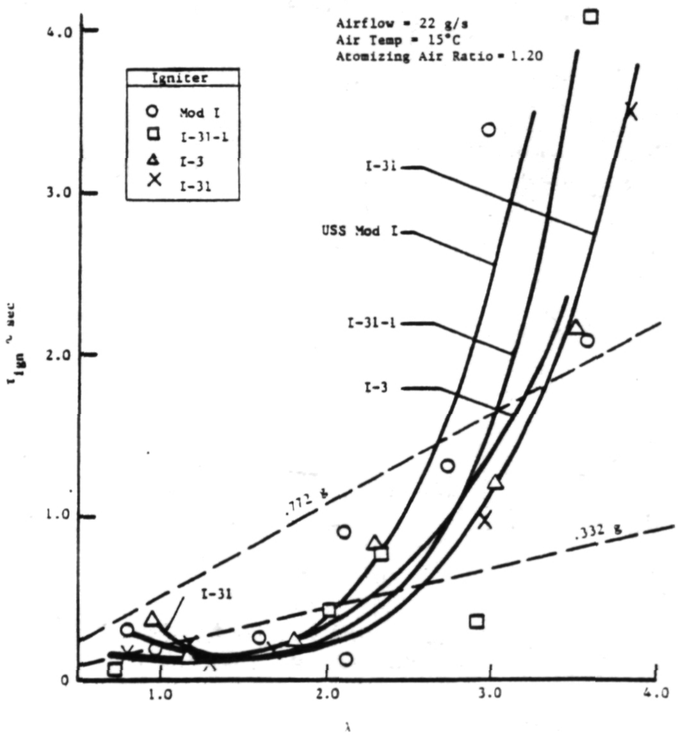


Fig. 4-6 Comparison of Ignition Delay Times of all Four Igniters (---Based on Allowable Fuel Flow Prior to Ignition)

UMI Combustor (Spray Pattern/Calibration)

Preparations for the UMI combustor design included the fabrication and preliminary testing of four alternate nozzles, a parameteric study of six alternate combustor designs, and the completion of a Combustion Performance Rig (except its simulated heater head) and all of its facilities.

The UMI nozzle is nonair-atomized, consistent with the RESD goal of eliminating the atomizing air compressor. Two air-blast and dual-orifice, pressure-atomized fuel nozzle bodies/bypass assemblies were fabricated for commercially available nozzles (were preliminarily tested in a newly constructed spray booth for spray pattern and flow characteristics). Calibration of the most promising nozzle (see Figure 4-7) indicates that it can handle the desired turndown ratio. After further subassembly testing in the Free-Burning Rig, the most efficient nozzle will be combined with a UMI combustor and evaluated in the Performance Rig (see Figure 4-8).

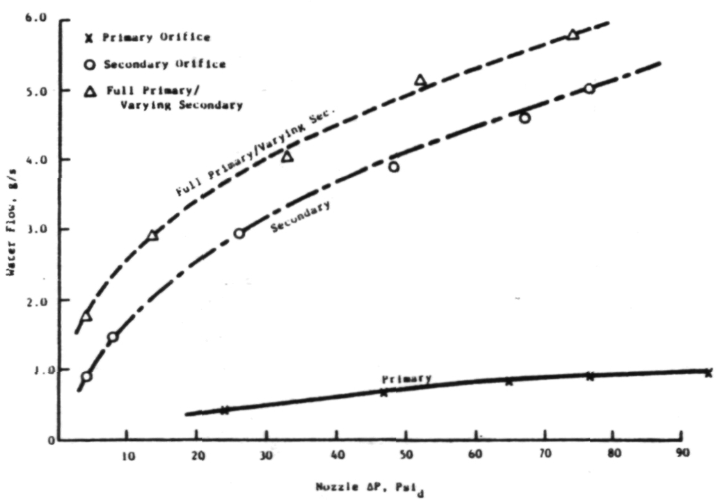


Fig. 4-7 Delavan Pressure-Atomized Fuel Nozzle No. 1 (35814)

A parametric study of emissions (NO_x, HC, CO, and particulates) was completed for six alternate combustor designs having the same dimensional envelope as the Mod I. The NO_x/HC/CO emissions index at 1 g/s (representative of average CVS cycle fuel flow) as a function of λ is shown in

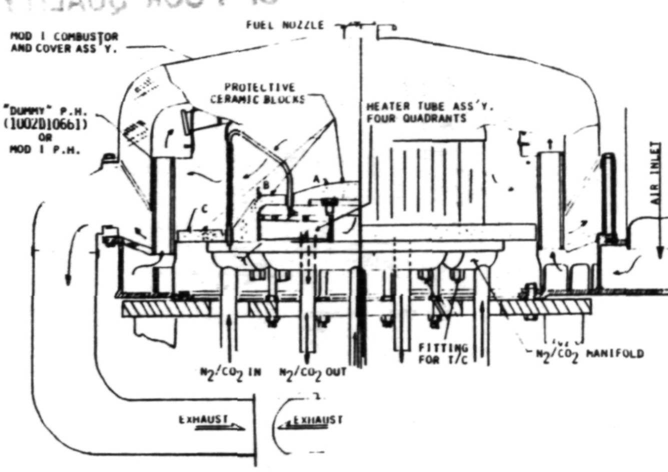


Fig. 4-8 Performance Rig

Figures 4-9a/b/c. Although they show low EI NO_x at 1 g/s (Figure 4-9a), combustors #1 and 2 have high EICO/EIHC at other flow rates, and have been eliminated as potential UMI combustors; thus, the most effective combustor for NO_x reduction, #6, has been selected for further engine tests. While the UMI combustors will be reduced in size, this study has provided a valuable input to their design.

The Combustion Performance Rig simulates the complete EHS combustor, igniter, nozzles, preheater, and heater tubes of the Hot Engine System. Direct comparisons of

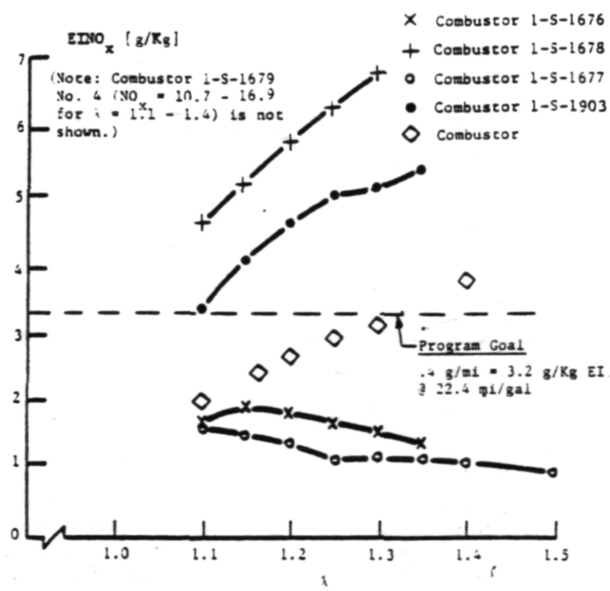


Fig. 4-9a Emission Index NO_x Versus Air Excess Factor

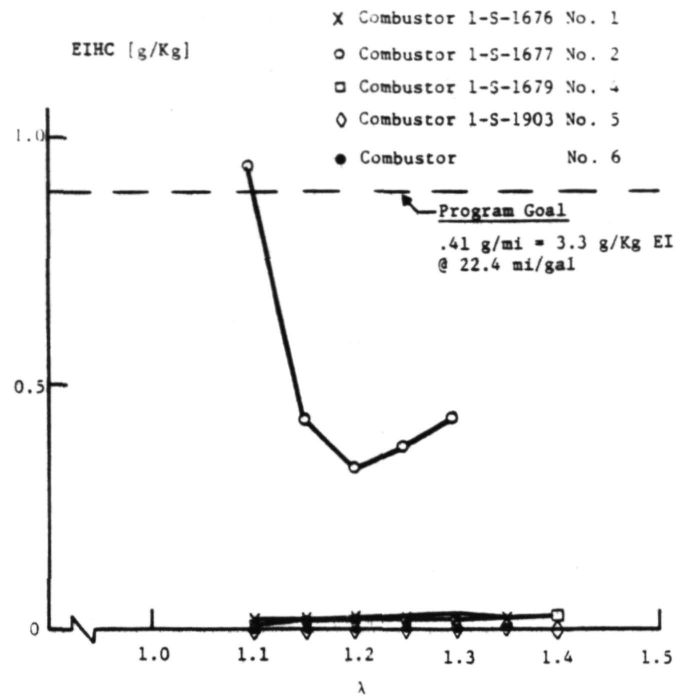


Fig. 4-9b Emission Index HC Versus Air Excess Factor

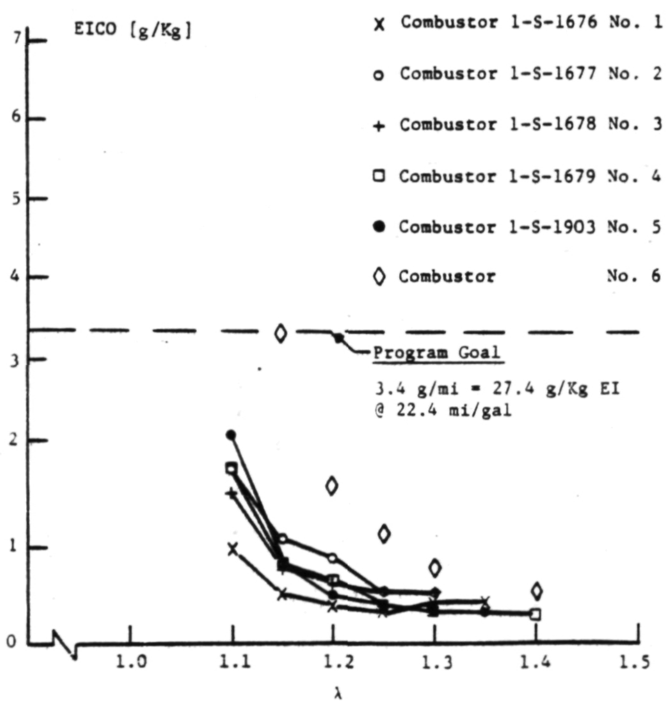


Fig. 4-9c Emission Index CO Versus Air Excess Factor

emissions/performance can be made to evaluated engine data/concepts, and parametric studies and optimizations can be obtained for the full range of fuel flows and engine power conditions. This rig (shown schematically in Figure 4-10), complete except for the heater head, will be the focus of continued UMI combustor and nozzle development activities.

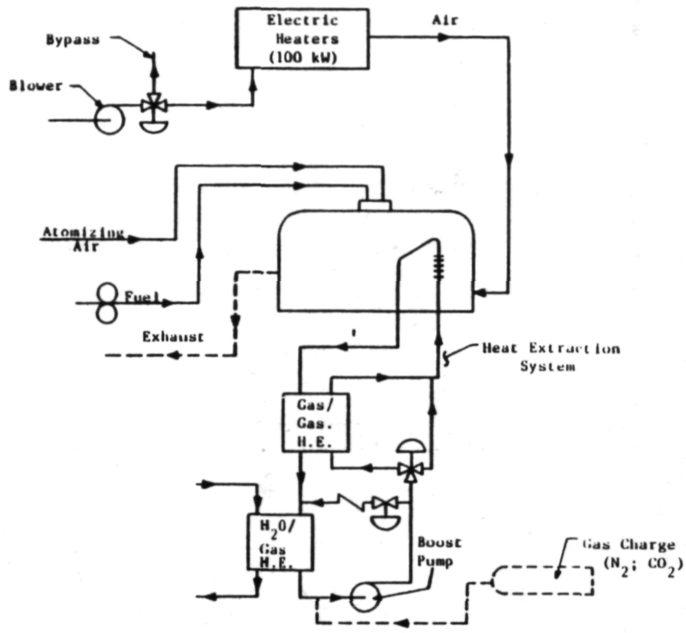


Fig. 4-10 Schematic of Combustion Performance Rig (Showing Fuel, Air, and Heat Extraction Systems)

Alternate Fuels Tests

One of the primary advantages of the Stirling engine is the ability of its external-combustion process to efficiently burn alternate fuels while maintaining low emissions, thereby making it attractive for automotive use; however, a scarcity of quality engine emissions test data caused the necessity of a series of tests with five alternate fuels.* These steady state tests, performed in a back-to-back comparative manner on a P-40 Stirling engine, served to demonstrate the

*see bottom of page 4-1

capabilities of the comprehensive emissions measurement instrumentation. The P-40 combustor used for these tests is representative of the EGR approach that can be used on the Mod I and UMI engines. The emissions sampling port and instrumentation are shown in Figures 4-11 and 12, respectively.

For each fuel, 16 power settings with and without EGR were run. Emissions were sampled from the engine's exhaust with and without EGR, and from the combustor inlet with EGR; thus, the number of measured emissions test points was 261*.

A summary of emission results for CO, NO_x, and HC, with and without EGR, is shown in Figures 4-13, 4-14, and 4-15, respectively. CO and HC emissions meet the calculated fuel economy goal of 22.4 mi/gal; however, NO_x emissions generally fall above .4 g/mi at 22.4 mi/gal (32 g/kg).

*includes 21 points to check repeatability

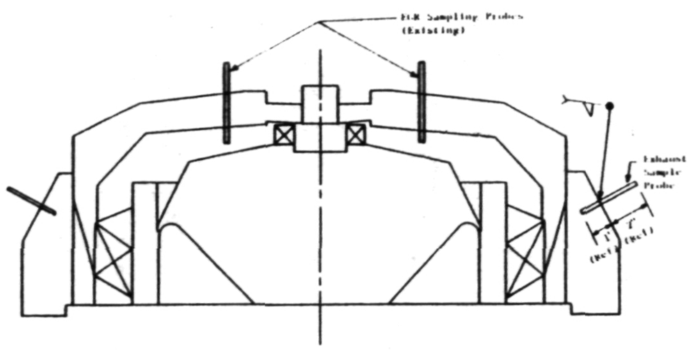
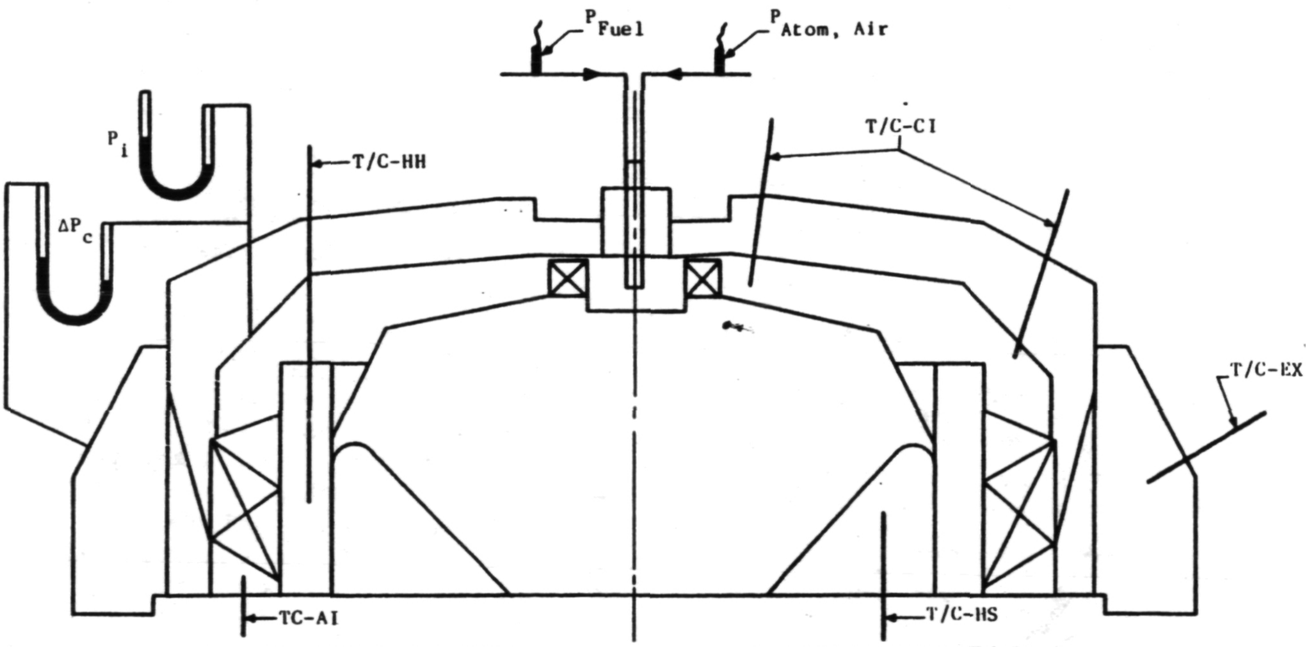


Fig. 4-11 Exhaust/EGR Gas Sampling Probes



T/C Design	Qty.	Description
T/C-CI	8	Combustor Inlet
T/C-AI	4	Air Inlet
T/C-HS	4	Hot Air Space
T/C-HH	4	Heater Head Exit
T/C-EX	4	Exhaust Gas

Pressures
P_i = Combustor Inlet
ΔP_c = Combustor Inlet - Exhaust

Fig. 4-12 Instrumentation Requirements

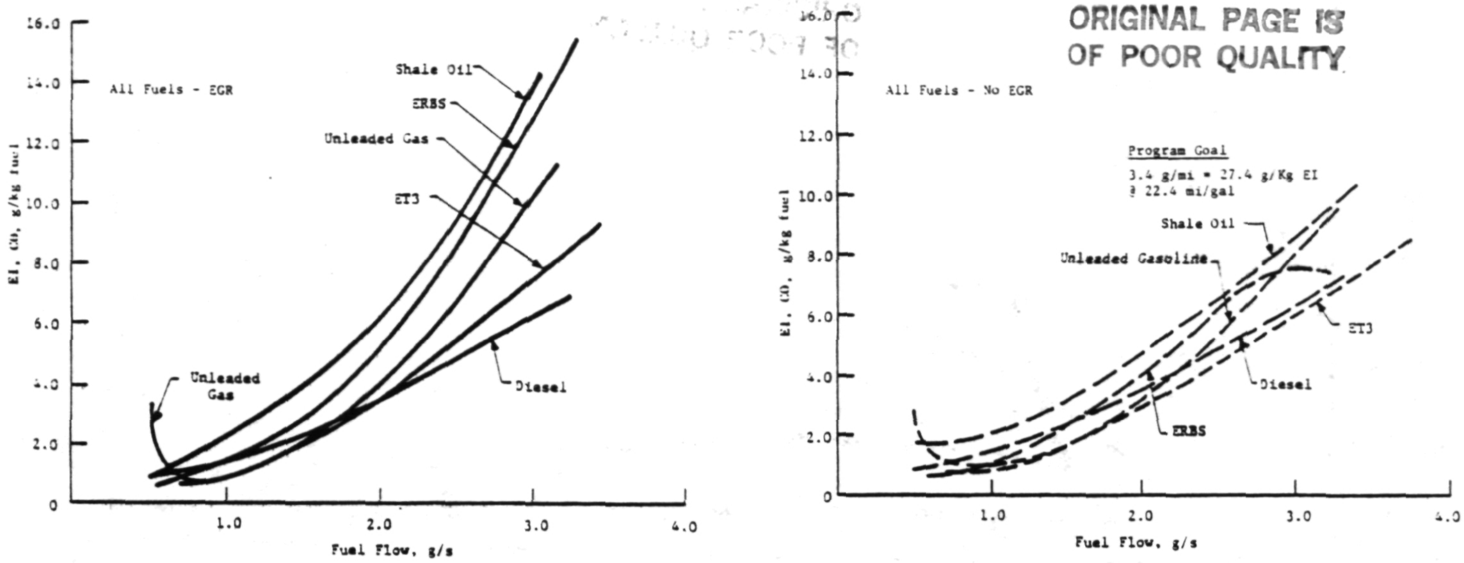


Fig. 4-13 CO Emissions P-40-7 Alternate Fuels Test

The amount of EGR achieved varies from 40 to 20% at fuel flows between 1.0 and 3.8 g/s, respectively. At least 50% EGR is needed over the entire load range to meet the above NO_x goal. Of special interest here is the fact that NO_x emissions with EGR are essentially independent of fuel used (see Figure 4-14 below), illustrating the feasibility of an EGR/CGR combustor that will meet the emissions goal while burning a variety of fuels. The emissions measuring equipment (shown in Figure 2-9 of Section II) used to obtain this data from the P-40 engine has been demonstrated to have the measuring capability shown in Table 4-1.

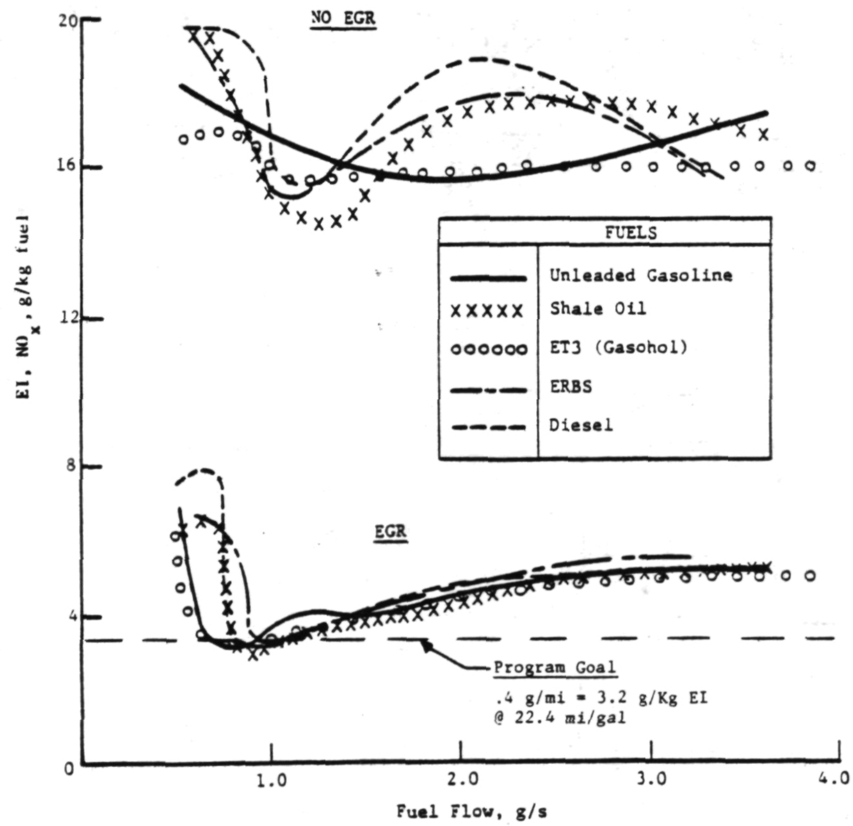


Fig. 4-14 NO_x Emissions P-40-7 Alternate Fuels Test

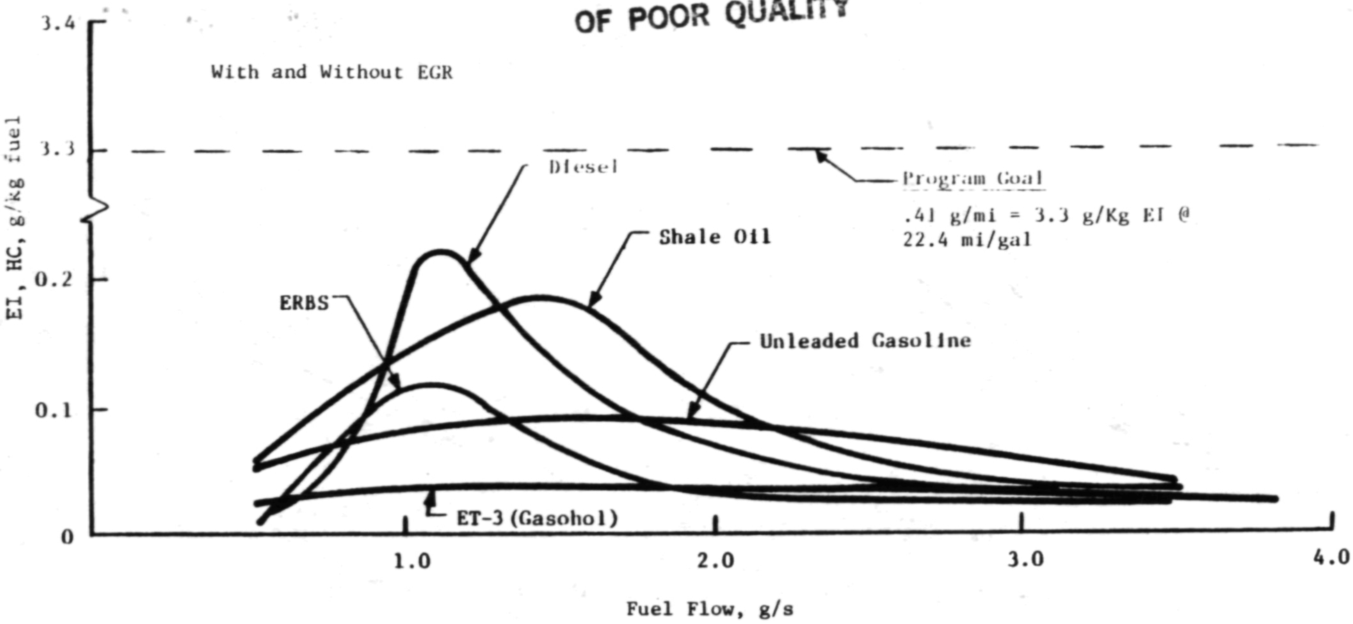


Fig. 4-15 Hydrocarbon Emissions P-40-7 Alternate Fuels Test

TABLE 4-1

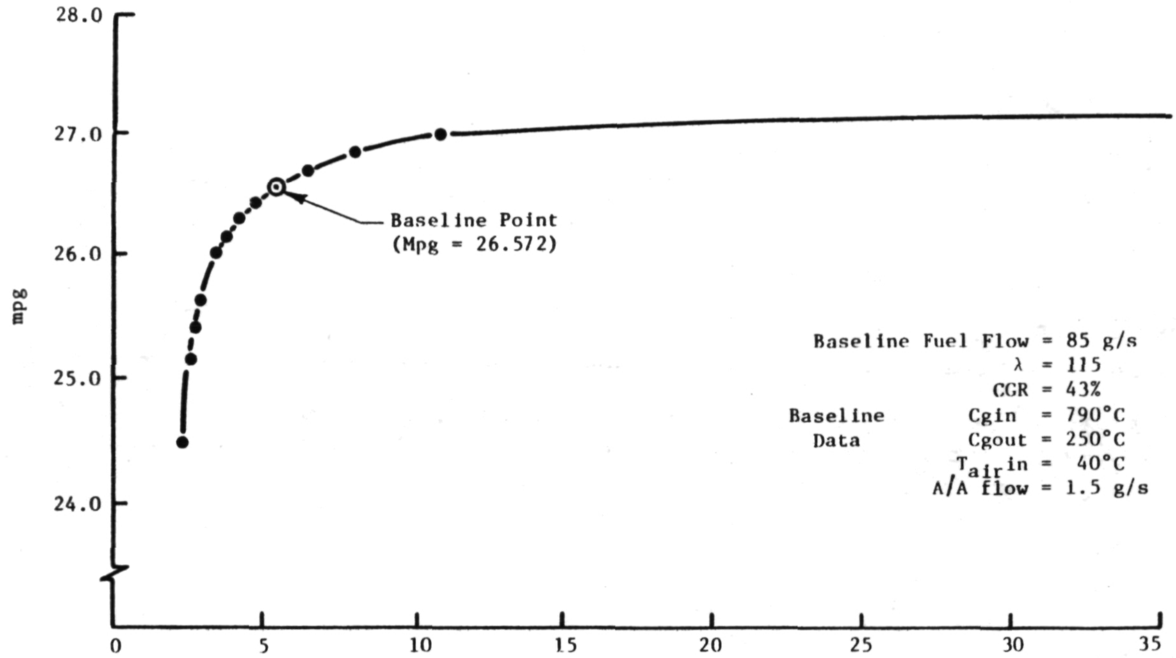
INSTRUMENTATION LISTING/CHARACTERISTICS

Instrument	Constituent	Ranges	Calculated Gas Accuracy
Beckman #955 Chemiluminescent Analyzer	NO/NO_x	10 ppm 25 ppm 100 ppm 250 ppm 1,000 ppm 2,500 ppm 10,000 ppm	 ±2% ±2% ±2%
Varian #1400 Total Hydrocarbon Flame Ionization Analyzer	HC	100 ppm 1,000 ppm	±2% ±2%
Horiba #AIA Lo CO NDIR Analyzer	CO	500 ppm 2,500 ppm 5,000 ppm	±2% ±2% ±2%
Horiba #AIA Hi CO NDIR Analyzer	CO	.5% 1.0%	±2% ±2%
Horiba #AIA CO_2 NDIR Analyzer	CO_2	5%	±2%
Perkin Elmer #154D Gas Chromatograph	CO_2 O_2 N_2		

Upgraded Mod I Preheater

A computer program was written to analyze the affect of preheater NTU (nondimensional heat transfer units) on CVS cycle mpg. This analysis used the average-operating-point approach, and assumed constant flame and heater tube temperatures.

Current Mod I design values for λ, % CGR, and preheater effectiveness at 1 g/s and 4.5 g/s fuel flows were used to obtain the results shown below in Figure 4-16. These results indicate that Mod I preheater NTU is near the maximum; however, due to the significant impact of preheater size on packageability and cost in an automotive



$$\text{Preheater NTU} = \frac{\text{Heat Transfer Coefficient x Heat Transfer Area}}{\text{Minimum Flow Heat Capacity}}$$

Fig. 4-16 MPG Versus Preheater NTU

TABLE 4-2

MOD I PREHEATER EFFECTIVENESS

% of Mod I External Heat System (EHS)	100%	95%	90%	85%
EHS Diameter	26 1/2"	25 1/8"	23 7/8"	22 1/2"
EHS Height	10 1/8"	9 5/8"	9 1/8"	8 5/8"
% Reduction in Preheater H.T. Area	0	21.5%	39.7%	62.2%
Preheater NTU	5.55	4.36	3.35	1.93
MPG	26.57	26.3	25.9	22.5
% Reduction in mpg	0	1.0%	2.5%	15.3%

Note: Analysis assumes that the heater head tube temperature will remain constant; however, with large changes in the size of the preheater, this assumption may not yield an optimum mpg.

application, a tradeoff of performance will be made to attain these objectives. The information in Table 4-2 was generated from the above analysis. To predict the affect of reducing the outer dimensions of the EHS, as can be seen from the results, a 10% reduction of Mod I volume can be accomplished with only a 2.5% reduction in CVS cycle mileage. These will be the size goals for the UMI preheater. Both ceramic and metallic preheaters will be considered for the reduced size design, and a small rig (shown schematically in Figure 4-17) will be fabricated to compare the designs.

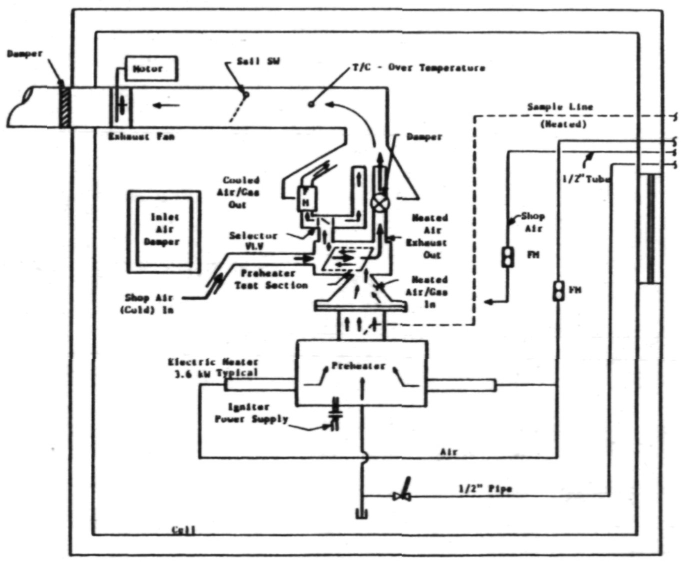


Fig. 4-17 Schematic of Preheater Rig

HOT ENGINE SYSTEM

The primary goal of this task is to provide a low-cost regenerator design with 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 woven high-count stainless mesh.

Activity during this report period was aimed at heat transfer rig testing of alternative regenerator matrix materials, and the selection of an optimum material and design. During the coming year, regenerators will be fabricated (with the selected material) and tested in an engine. An evaluation of heater head performance in both the engine and Combustion Performance Rig will also 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 4-18) is used for these measurements. In this rig, air is blown through a sample test section (see Figure 4-19) 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 and outlet temperatures is shown in Figure 4-20. The Nusselt Number of the sample matrix material can be determined from analysis of the outlet temperature trace.

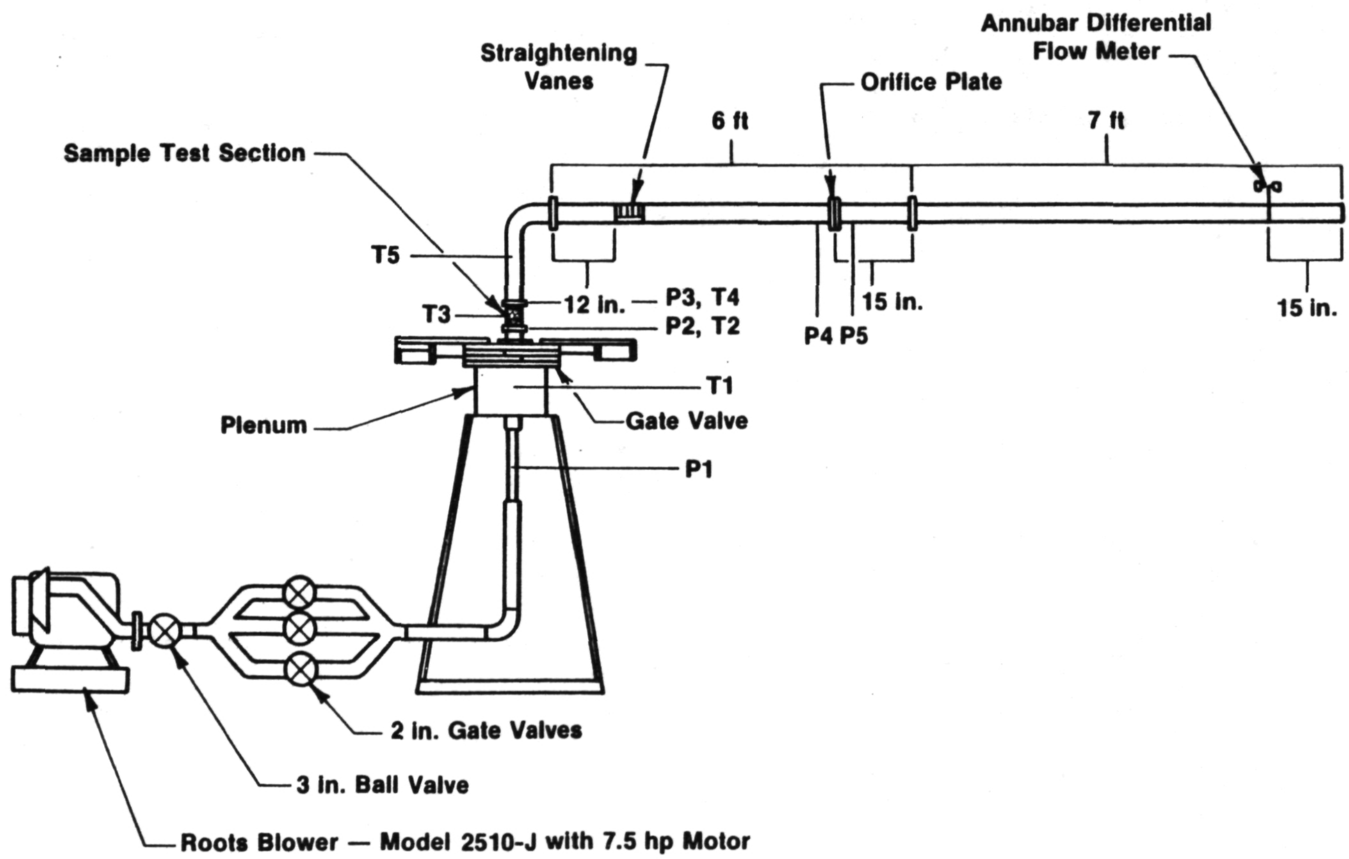


Fig. 4-18 Regenerator Matrix Test Rig

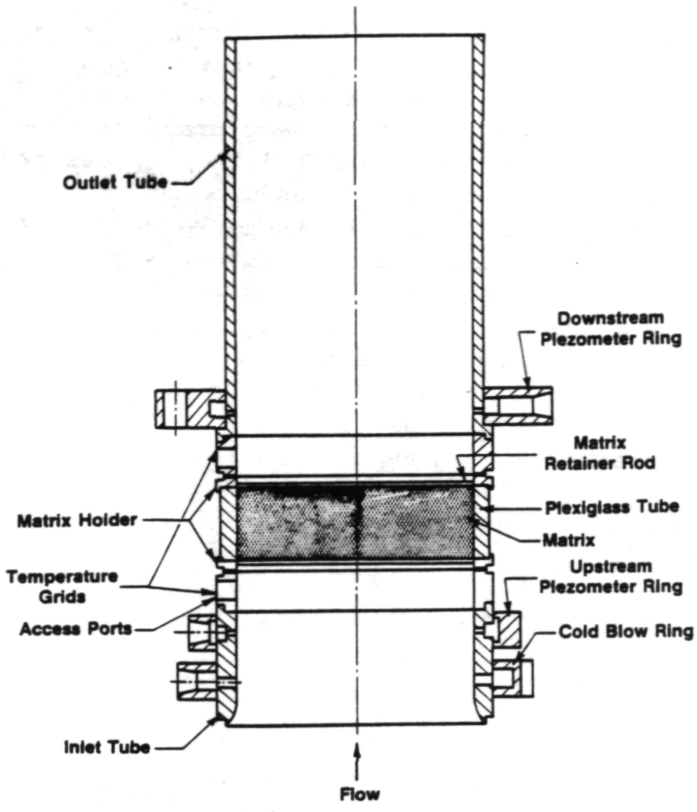


Fig. 4-19 Sample Test Section

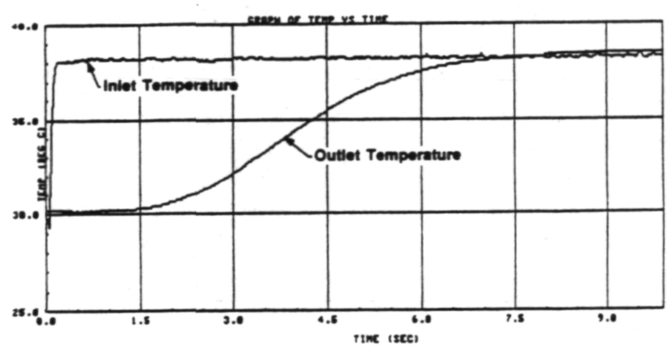


Fig. 4-20 Typical Temperature History

The following material combinations (having several different porosities and thicknesses) were tested:

- 200-mesh, 0.002-inch sintered wire cloth;
- 200-mesh, 0.002-inch unsintered wire cloth;
- 60-mesh, 0.0075 inch sintered wire cloth;
- 60-mesh, 0.0075-inch unsintered wire cloth; and,
- 0.006-inch "Metex" knitted wire cloth.

The sintered 200-mesh wire cloth, currently used in the Mod I engine, is the baseline to which all other materials must be compared. A comparison of the wire cloth alternatives is shown in Figures 4-21 and 4-22 for Nusselt Number and friction factor, respectively. The unsintered screen has the highest Nusselt Number and lowest friction factor; therefore, it will have the highest efficiency at the tested porosity. While the coarse-mesh wire cloth has a cost advantage over the fine-mesh wire cloth, it did not achieve the program goal of a 70% cost reduction.

Further tests were run (shown in Figures 4-23 and 4-24) on a coarse (0.006") Metex knitted wire material; however, this material proved to be unsatisfactory because of a lower Nusselt Number and higher friction factor. The results were extrapolated to a finer wire size (0.002") Metex material, with the expected performance calculated (shown in Figure 4-25) indicating that only a 1% penalty in efficiency would result. Tests with this material are planned in Program Year 1982, since it can achieve the 70% cost-reduction goal.

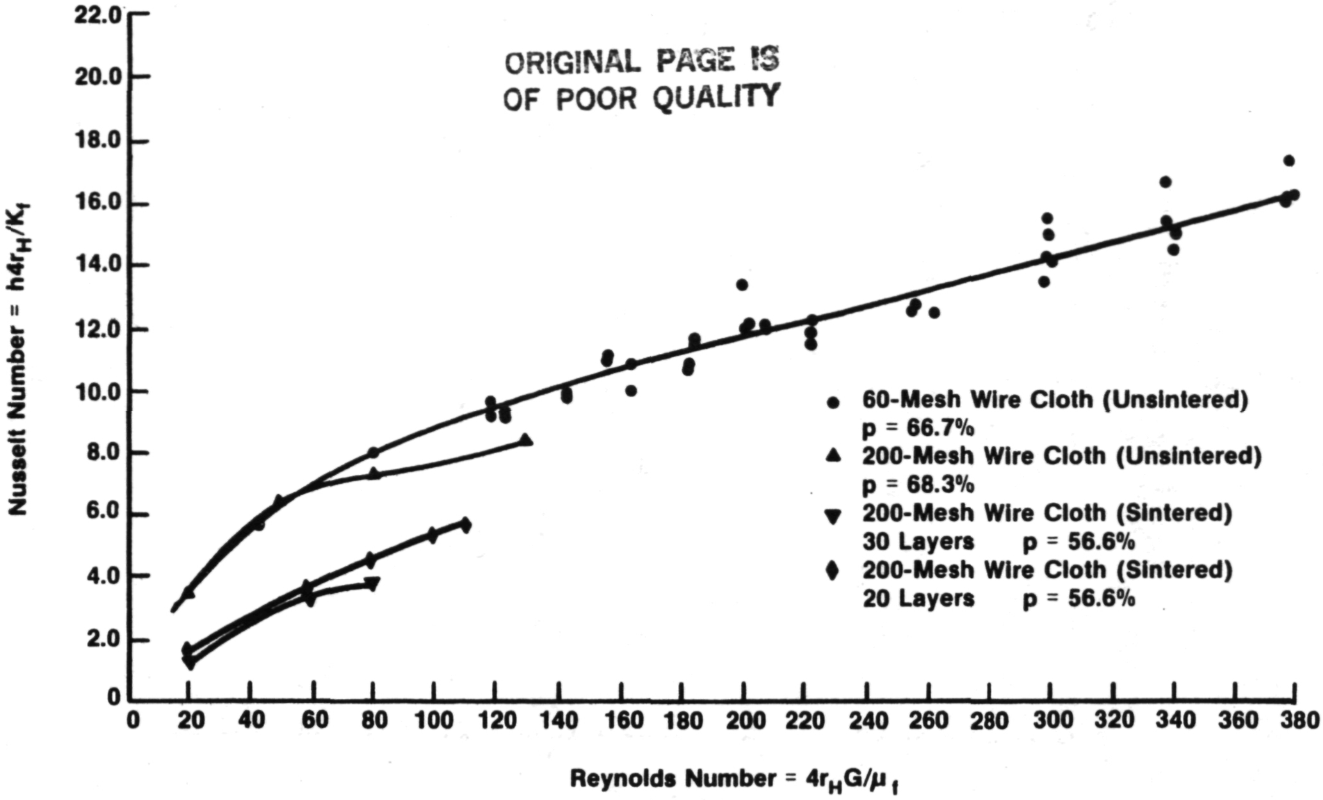


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Fig. 4-21 Unsintered 60-Mesh Test, Nusselt Number Versus Reynolds Number

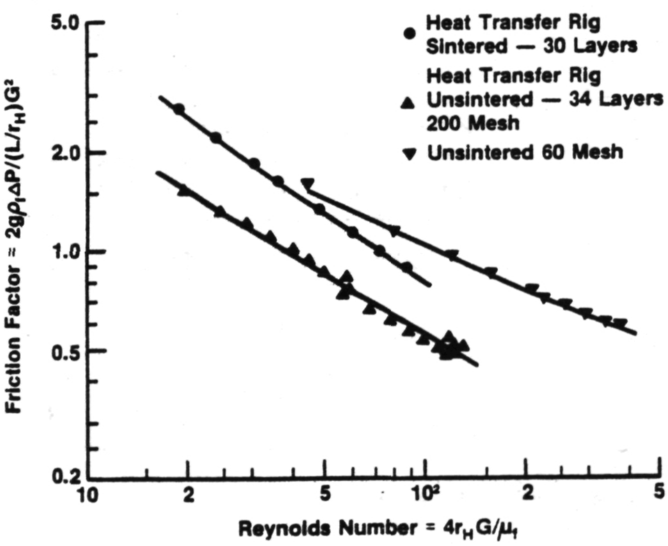

Fig. 4-22 Unsintered 60-Mesh Test, Friction Factor Versus Reynolds Number

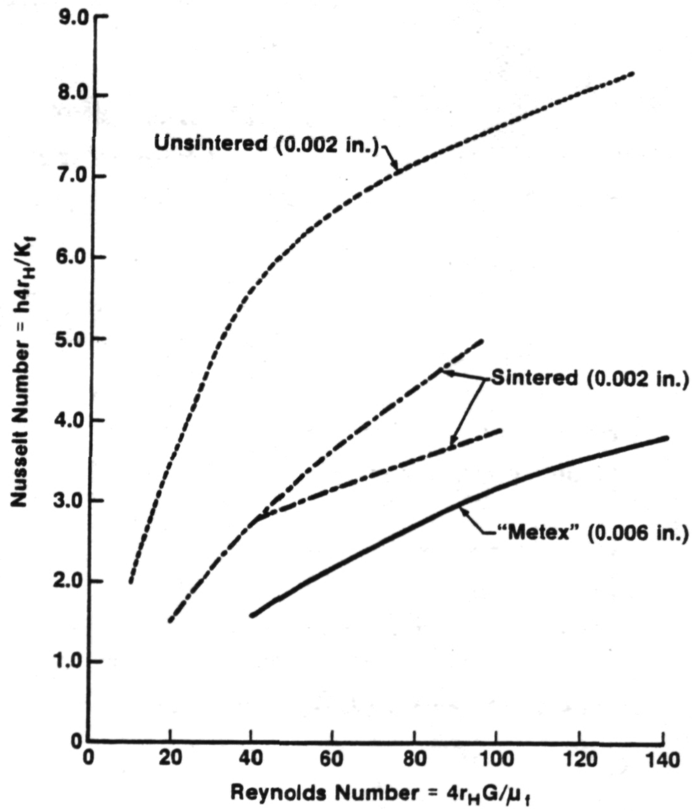

Fig. 4-23 Regenerator Matrix Test, Nusselt Number Versus Reynolds Number

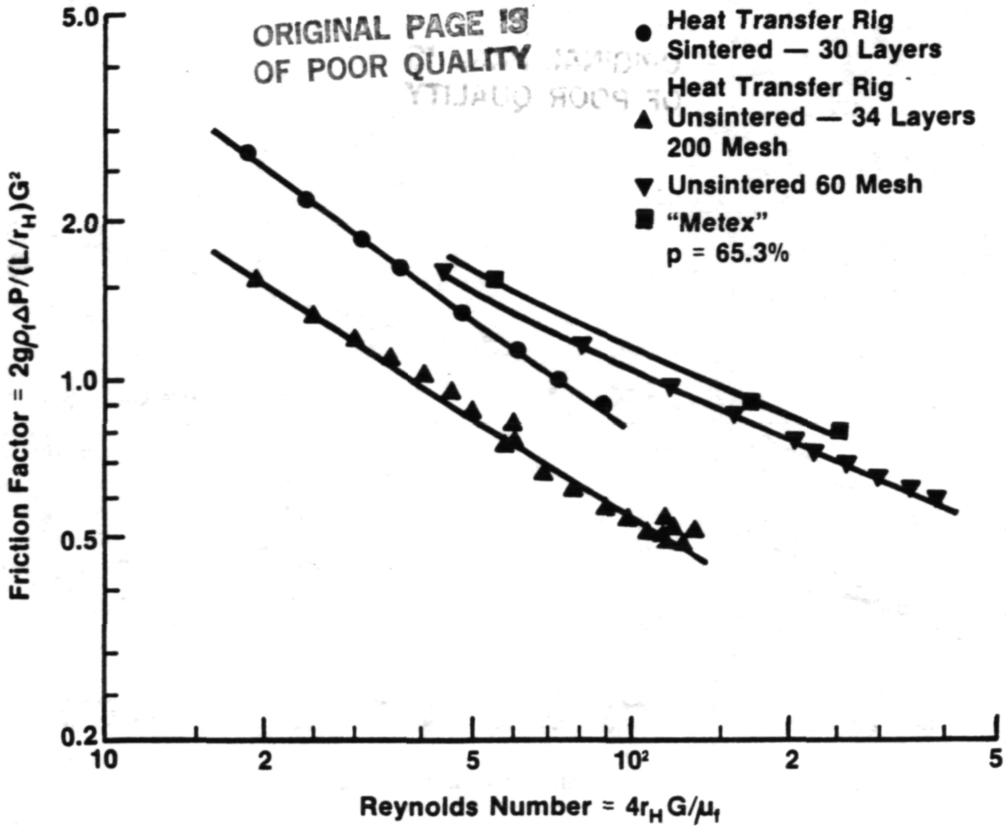

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Fig. 4-24 Sintered/Unsintered Test, Friction Factor Versus Reynolds Number

Data Applies to the Mod I Engine at the Average Operation Point (5 MPa at 2000 rpm)

	Sintered Regenerator Wire Dia. = 0.002 in. Porosity = 57%	Unsintered Regenerator Wire Dia. = 0.002 in. Porosity = 68%	"Metex" Regenerator Wire Dia. = 0.006 in. Porosity = 65%	"Metex" Regenerator Wire Dia. = 0.002 in. Porosity = 57%
Indicated Power (kW)	16.2	16.3	16.5	16.2
Indicated Efficiency (%)	47.0	49.3	40.8	46.4
Estimated Change in mpg (%)	0	+5	-13*	-1

Due to high losses in this regenerator, the maximum power point (15 MPa at 4000 rpm) would not be attainable.

813051

Fig. 4-25 Regenerator Performance Effect on Stirling Engine Performance

MATERIALS DEVELOPMENT

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

Recent development efforts have focused on the testing of three specimens of alternative casting materials and five alternative tubing materials, the fabrication of two heater heads from these materials, and heater head permeation testing.

During 1982, the alternative-material heater heads will be tested on an engine in a simulated duty cycle, fatigue tests of the casting materials will be performed in a hydrogen environment, and the materials for an upgraded low-cost heater head will be selected.

Design Properties Tests

The three alternative iron-base casting alloys (XF-818, SAF-11, CRM-6D) and the current cobalt-base Mod I material* (HS-31) have been subjected to room-temperature tensile tests (over a range of operating temperatures in air) in order to determine their design properties. The results, shown in Figures 4-26 through 4-29, indicate that all the materials will be acceptable for use in a low-cost design since they exceed design stress requirements and are ∿30-40% of the cost of HS-31.

The low-cost tubing materials, Sandvik 12RN72, Sanicro 31H and 32, Inconel 625, CG-27, and the current cobalt-containing Mod I material, Multimet (N155), have been under test for creep rupture strength for as long as 2800 hours. These tests were performed under internal gas pressures up to 44 MPa, and temperatures to 850°C.

*see composition in Table 4-3

TABLE 4-3

COMPOSITION OF ALLOYS USED IN TESTING PROGRAM

	XF-818 (Climax Molybdenum)	XF-818 (Bulten-Kanthal)	CRM-6D (Bulten-Kanthal)	SAF-11 (Bulten-Kanthal)	HS-31 (Bulten-Kanthal)
C	0.22	0.22	1.11	0.64	0.54
Si	0.30	0.34	0.5	0.62	0.90
Mn	0.20	0.09	4.6	0.61	0.4
Nb+Ta	0.46	0.53	1.00	-	-
S	-	0.024	-	-	0.02
Cr	18.2	18.9	21.5	23.0	25.6
Ni	18.3	17.1	5.4	16.6	10.4
Mo	7.4	8.1	1.03	-	-
B	0.76	0.66	0.006	0.45	-
N	0.094	0.12	-	-	-
CO	-	-	-	-	BALANCE
W	-		1.02	12.8	7.9
P	-		-	-	0.03
FE	BALANCE	BALANCE	BALANCE	BALANCE	0.5

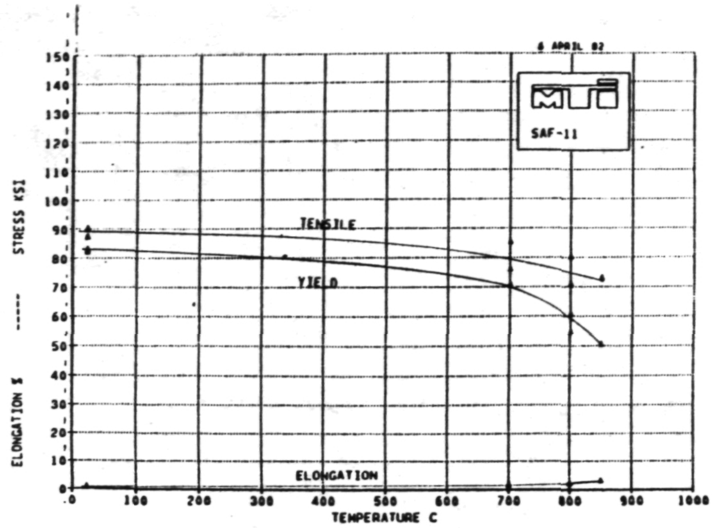

Fig. 4-26 Tensile Test, HS-31, Room Temperature to 850°C

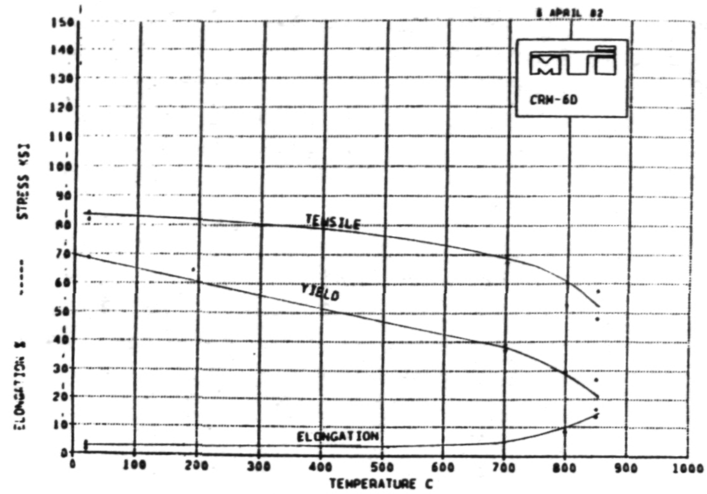

Fig. 4-27 Tensile Test, XF-818, Room Temperature to 850°C

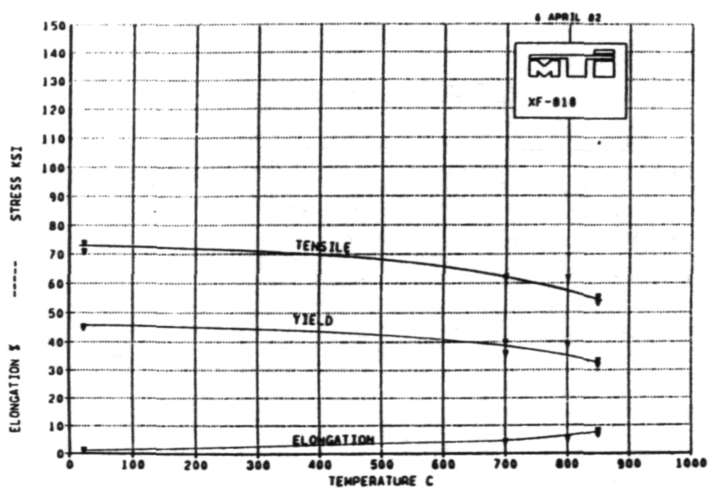

Fig. 4-28 Tensile Test, CRM-6D, Room Temperature to 850°C

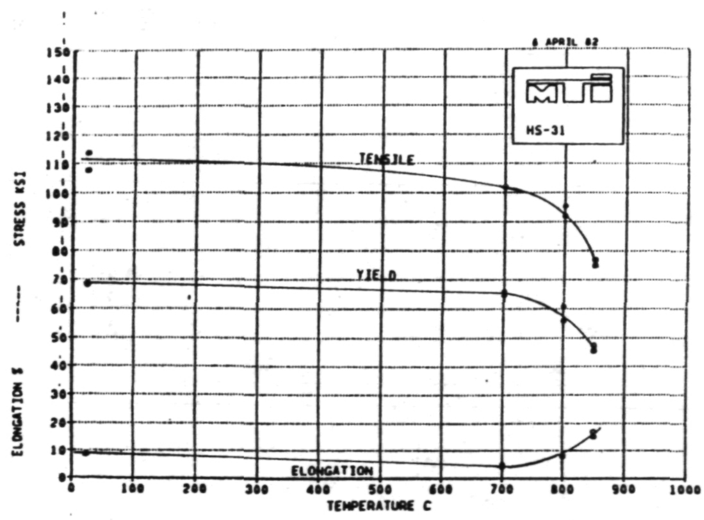

Fig. 4-29 Tensile Test, SAF-11, Room Temperature to 850°C

The results for the three most promising materials (CG-27, Inconel 625, Sandvik 12RN72) are shown in Figures 4-30 through 4-32. In all instances, the desired design stress of 28 MPa (4060 psi) at 3500 hours was exceeded. The cost of these three materials was ∿50% or less of the Multimet baseline material due to the substantial reduction in critical alloy content.

Casting Material Fatigue Tests

A series of fatigue tests were performed on HS-31 and the three alternative casting materials to determine their endurance under cyclic loading conditions (results are shown in Figures 4-33 through 4-36). Tests were run under fully reversed (R = -1) and tension-tension (R = .5) conditions at 800°C. In all cases, the tests exceeded the requirements for the Mod I rainflow analysis*; thus, the materials are considered viable candidates for low-cost regenerator housings and cylinder heads. The final selection will be based on the results of fatigue testing, high-temperature P-40 (HTP-40)* testing, and hydrogen atmosphere fatigue testing.

*summation of the number of stress cycles at the different stress ranges/time intervals encountered in the driving cycle.

High-Temperature Engine Tests

One of the RESD goals is to demonstrate that the Stirling engine can run for six months/6000 miles without a working gas recharge, and with only a 10% reduction in engine power (translates to average standing leakage rate of 0.06 NL/hr, and average operating leakage (seals plus permeation) of 1.2 NL/hr). In order to demonstrate the capability of reaching this goal, a test series was conducted on the HTP-40 using hydrogen and 1% CO_2 doped hydrogen. Preliminary results of a measured stationary leakage of 0.048 NL/hr, and an asymptotic operating leakage of 1.8 NL/hr, indicate that the RESD goal could be met.

*tested at an average heater head temperature of 820°C, rather than at a base engine temperature of 720°C

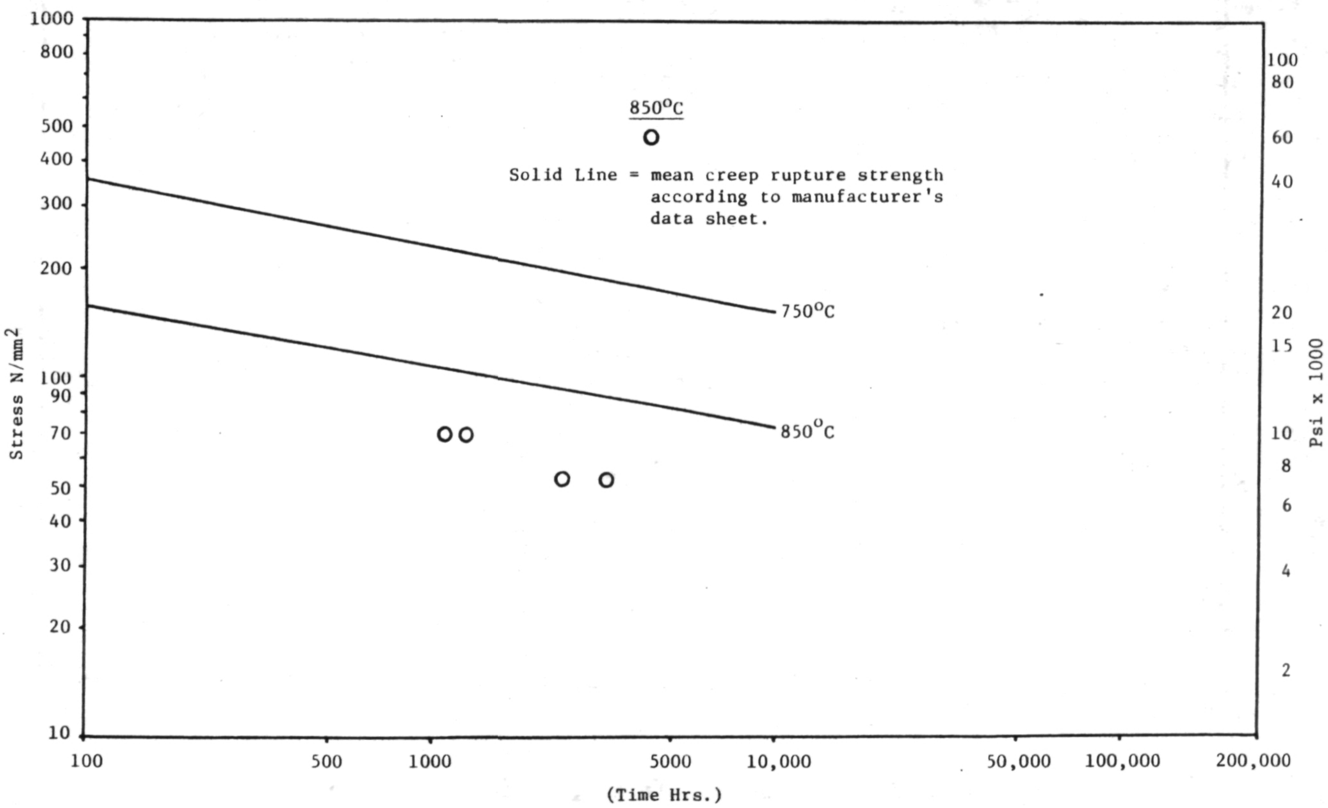

Fig. 4-30 Creep Rupture Testing of CG-27

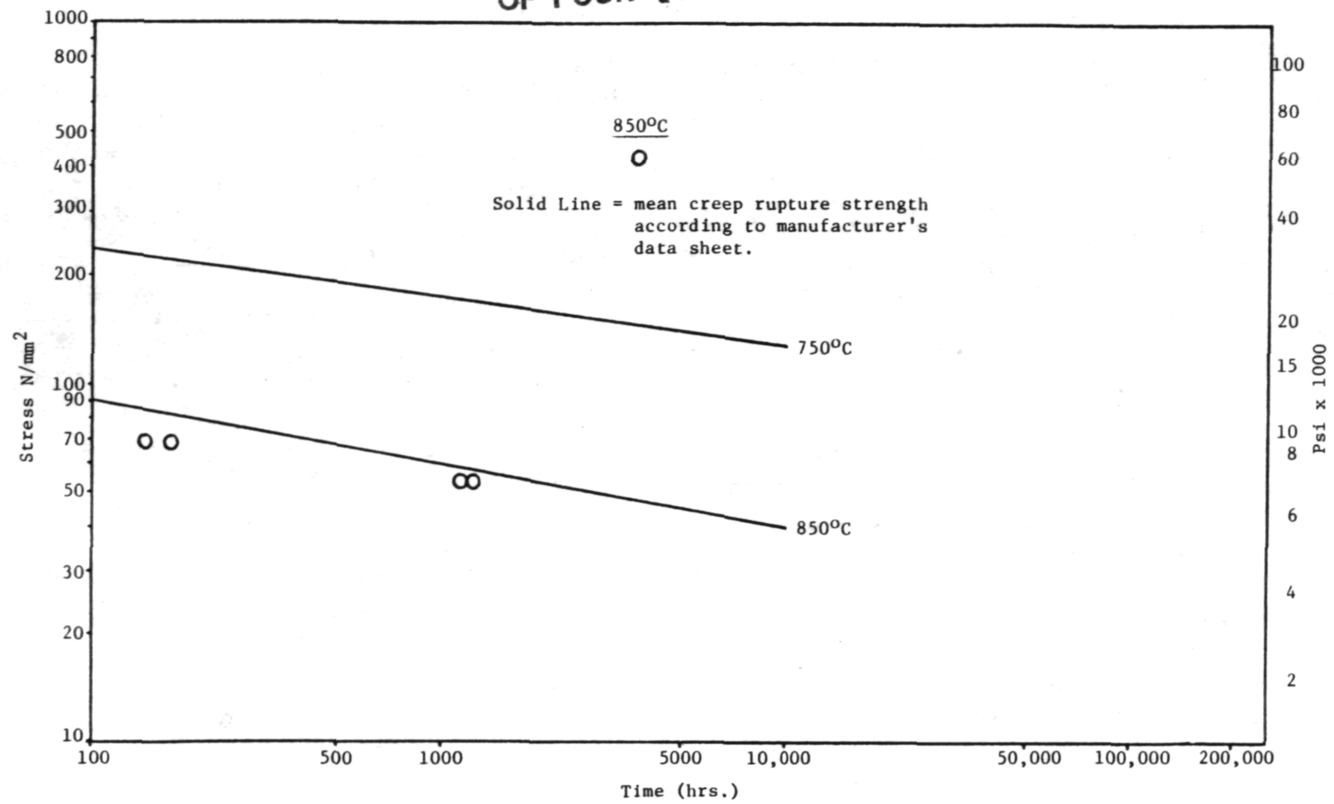

Fig. 4-31 Creep Rupture Testing of IN 625

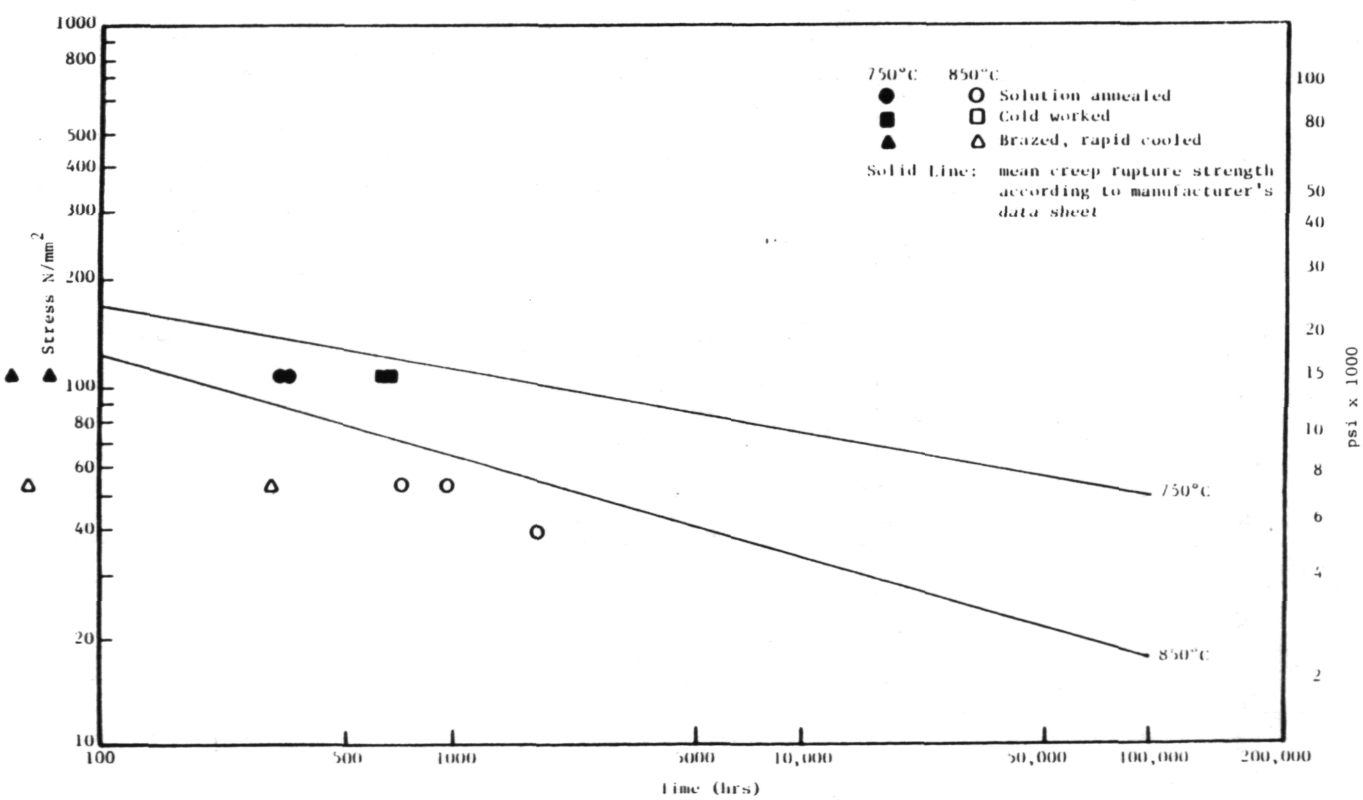

Fig. 4-32 Creep Rupture Testing of Sandvik 12RN72

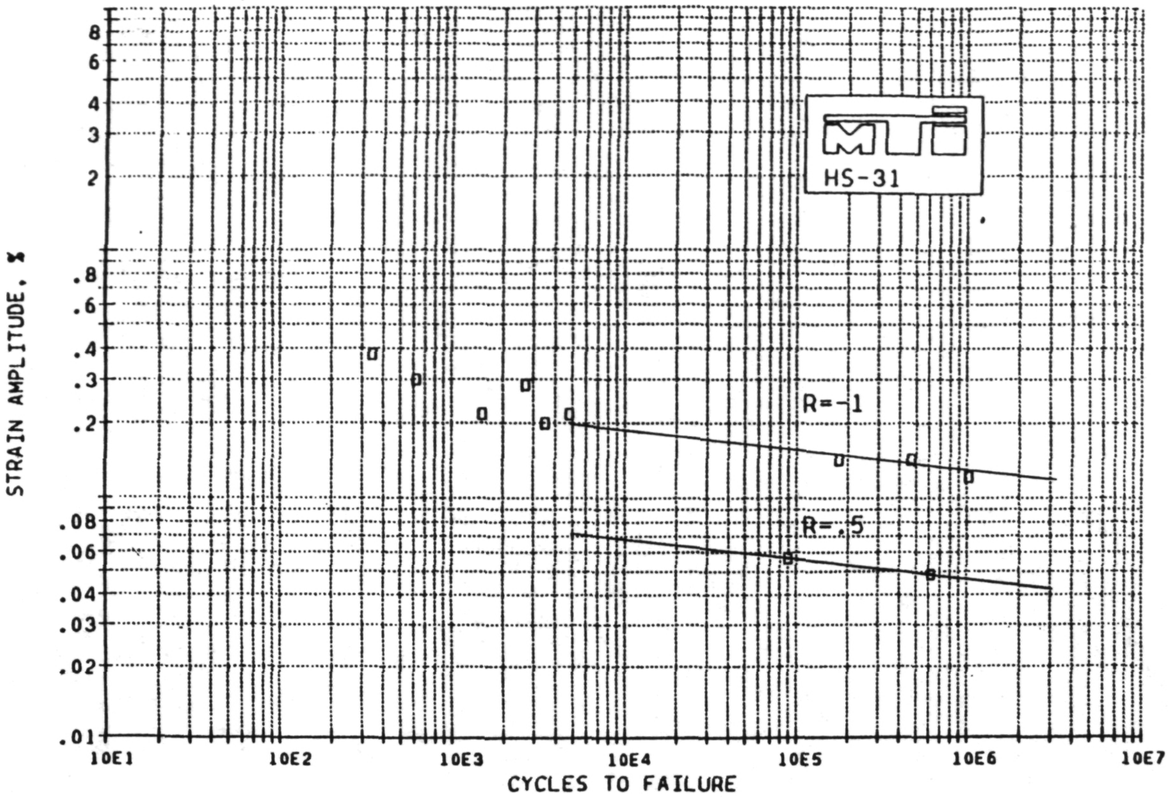

Figure 4-33 Fatigue Tests, 800°C, Air, Baseline Casting Alloy

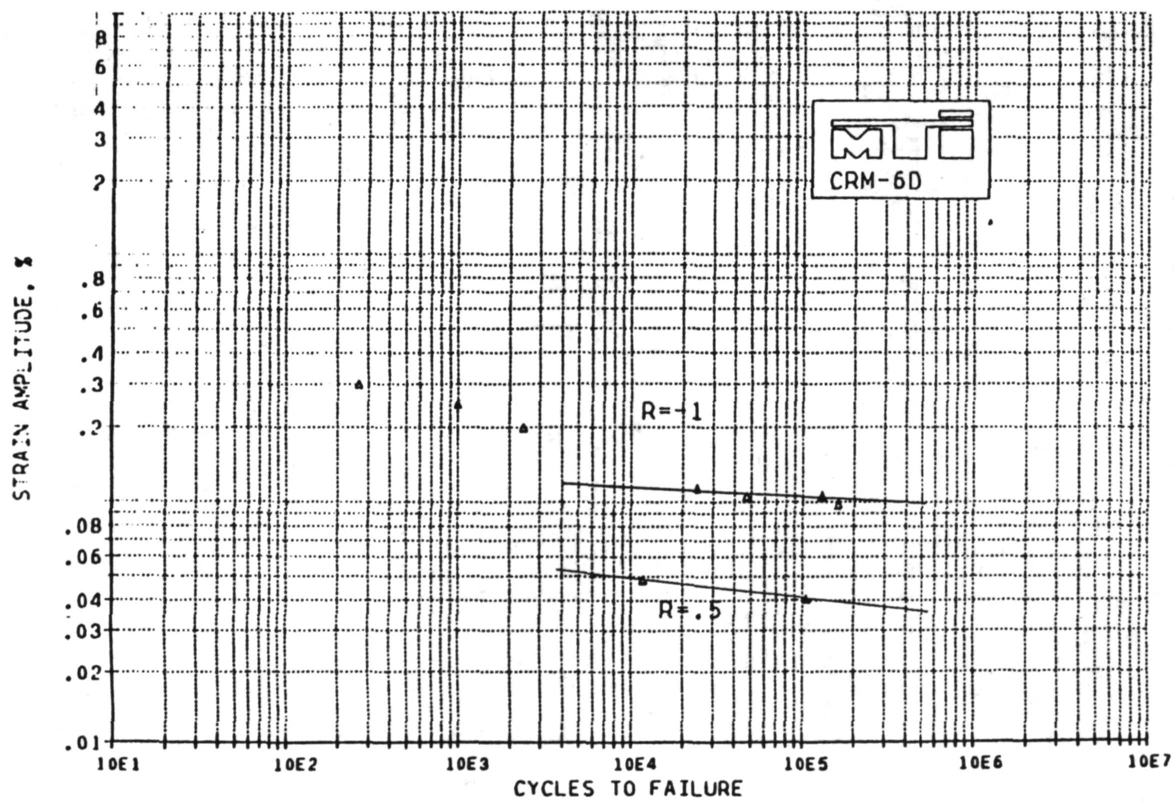

Fig. 4-34 Fatigue Tests, 800°C, Air, Low-Cost Casting Alloys

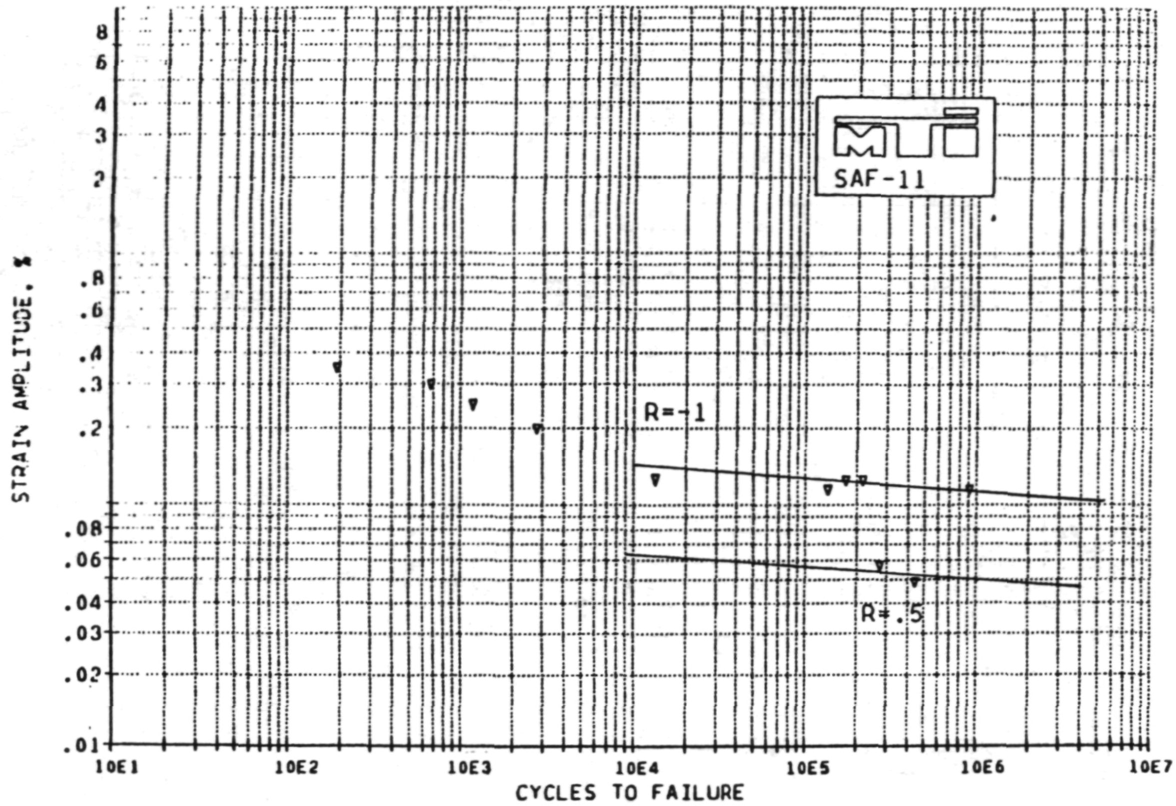

Fig. 4-35 Fatigue Tests, 800°C, Air, Low-Cost Casting Alloys

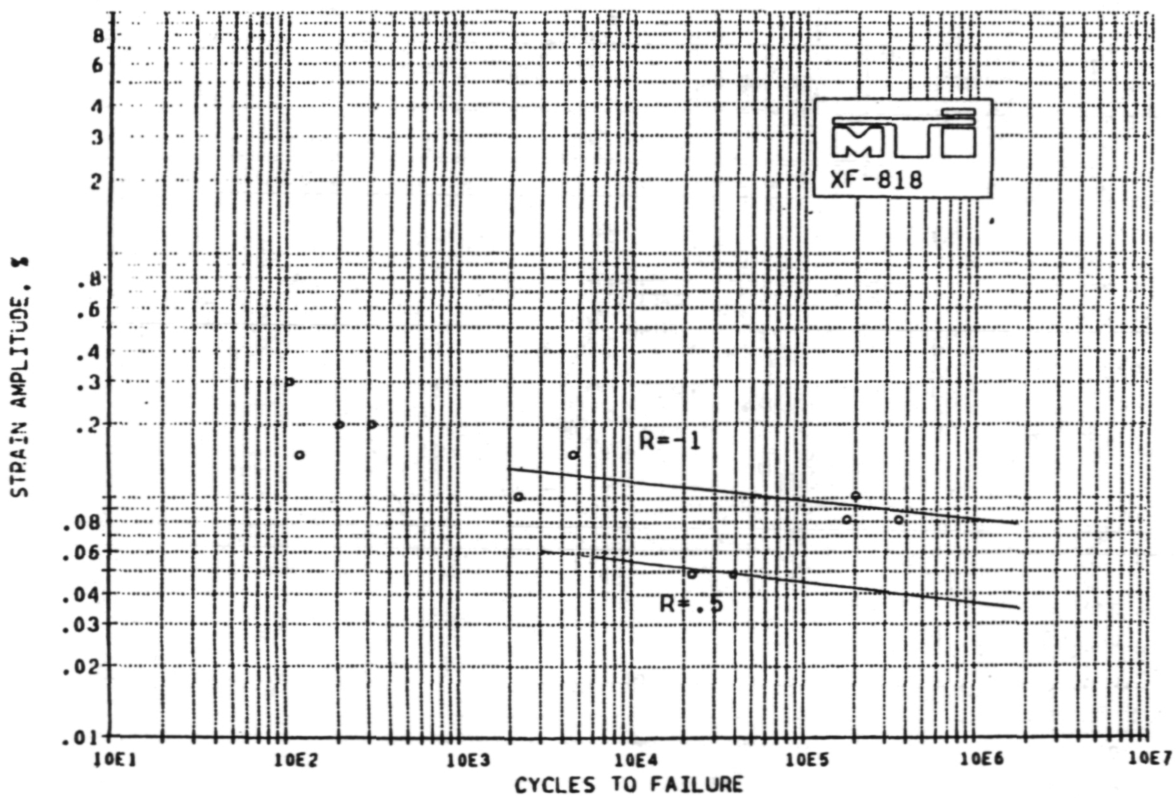

Fig. 4-36 Fatigue Tests, 800°C, Air, Low-Cost Casting Alloys

In preparation for HTP-40 engine testing during 1982, eight heater head quadrants have been manufactured with the combinations of tube and casting materials shown in Table 4-4. These tests will be used to compare the tube and casting alloys under a simulated automotive duty cycle.

TABLE 4-4

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

COLD ENGINE SYSTEM (CES) DEVELOPMENT

The focus of CES activity is the development of reciprocating seals having low friction and wear when subjected to an automotive duty cycle. Effort early in this reporting period was directed toward the screening rig evaluation of both friction and wear for candidate seal materials identified earlier in the program. Promising materials were then compared to the Mod I baseline design in a single-cylinder piston ring test rig under a simulated automotive-style test cycle. Further, an analytical study was performed to determine the distribution of seal and drive component friction losses as an aid in directing development efforts.

The goals for PY 1982 are to complete piston ring tests of the promising materials, evaluate two alternative piston ring designs, evaluate two potential main seal designs capable of operating without a cap seal, and select the UMI design. Both the Mod I motoring rig and engine tests will be performed as a final check of the acceptability of these designs.

Seal Material Screening Tests

The Mod I engine design contains three different types of reciprocating seals: piston rings, cap seals, and main seals (shown schematically in Figure 4-37). The piston rings separate the gas cycle volumes above and below the pistons. Leakage of gas across the piston rings reduces the power output from the engine. The cap seal on the piston rod separates the gas cycle below the piston from the volume between the cap seal and the main seal, where the pressure is maintained at a constant level. Leakage of gas across the cap seal is equivalent to increasing the dead volume in the engine and reducing power output. The main seal performs two functions: 1) prevents leakage of gas from the engine into the crankcase; and 2) prevents the crankcase oil from entering the engine. With a vented crankcase, any gas

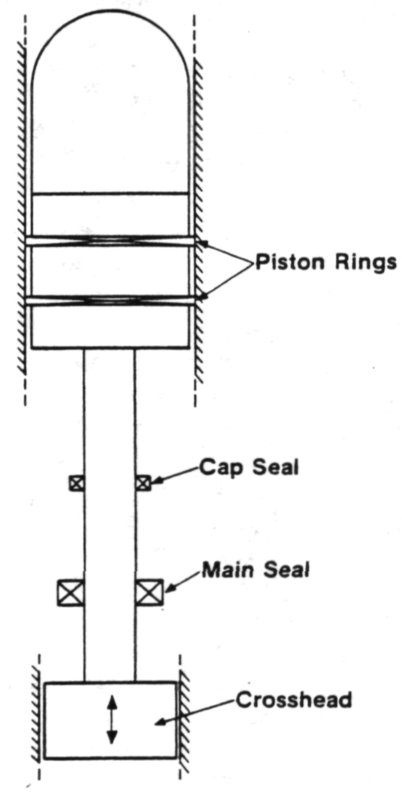

Fig. 4-37 Reciprocating Seals

that passes the main seal cannot be recovered, directly affecting the recharge time (an important factor in automotive application). Oil that enters the gas cycle contaminates the engine (particularly the regenerators) and leads to a deterioration in engine and seal performance.

For automotive application, the seals must be capable of reliable, effective operation over a long period of time, and friction in the seals must be minimized to achieve high overall engine efficiency. Although the reciprocating seals are located in the cold part of the engine, the environment and conditions under which they operate are exacting. The choice of seal materials and designs is critical to meeting the overall requirements of effective, low-friction, long-lived seals.

In the current Mod I engine design, the piston rings and cap seals are made from Rulon LD, a PTFE-based* material. Selection of this material was based on USSw's experience with different materials in their engine development programs.

The screening program was evolved to identify alternative and potentially superior seal materials. In this program, a wide range of materials has been evaluated using data derived from two test rigs. One rig was designed to measure wear of seal materials under reciprocating motion, while the other was a pin-on-disc-type machine in which friction measurements were made under unidirectional sliding conditions. The two rigs are shown in Figures 4-38 and 4-39. For the basic screening tests, the wear measurements were made over a 100-hour period at a speed of 1200 rpm (typical average engine speed), and the specimens were loaded to 0.69 MPa (100 psi), which is within the PV limit for Rulon LD under those conditions. After studies of published literature, manufacturers' data, and MTI experience in relevant applications, 31 materials were initially selected for evaluation, including Rulon LD. The summary wear and friction data for the best materials tested are presented in Figure 4-40, together with the data for the best low-friction/low-wear materials from the original list (see Table 4-5). These results are only valid for the particular conditions adopted for the tests; however, they do suggest that materials having a significantly lower wear than that of Rulon LD are available. This lower wear may also be accompanied by reduced friction.

The Koppers K30W35 material contains approximately 35% fine copper wire in short lengths that is oriented in a particular direction. The test data indicate that the orientation of the wires relative to the sliding surface is important. The specimens tested with the wires normal to the plane of sliding gave the lowest wear rate of all the materials tested, and had the smallest coefficient of friction (μ). Test samples with the wires parallel to the plane of sliding gave approximately three times the wear rate and a slightly higher coefficient of friction. This behavior is probably the result of differences in heat conduction properties of the material in directions parallel and normal to the wires, but this conclusion has not been verified. If this heat conduction difference does affect the behavior, maintaining the preferred orientation of the wires relative to the sliding surface may not be possible in an actual reciprocating seal application.

All friction and wear data were derived under dry sliding conditions and, as such, are mainly relevant to piston ring/cap seal applications. With the Mod I engine design, oil will always be present on the piston rod in the section between the crosshead and the main seal. In practice, the main seal must be nearly 100% efficient in preventing the oil from entering the engine. Inevitably, however, some trace of oil will be carried into the main seal contact region on the surface of the rod. To investigate the affects of this oil on the seal wear, the reciprocating wear test rig was slightly modified. Felt wicks were introduced adjacent to the test specimens, contacting the reciprocating

*polytetrafluorethylene

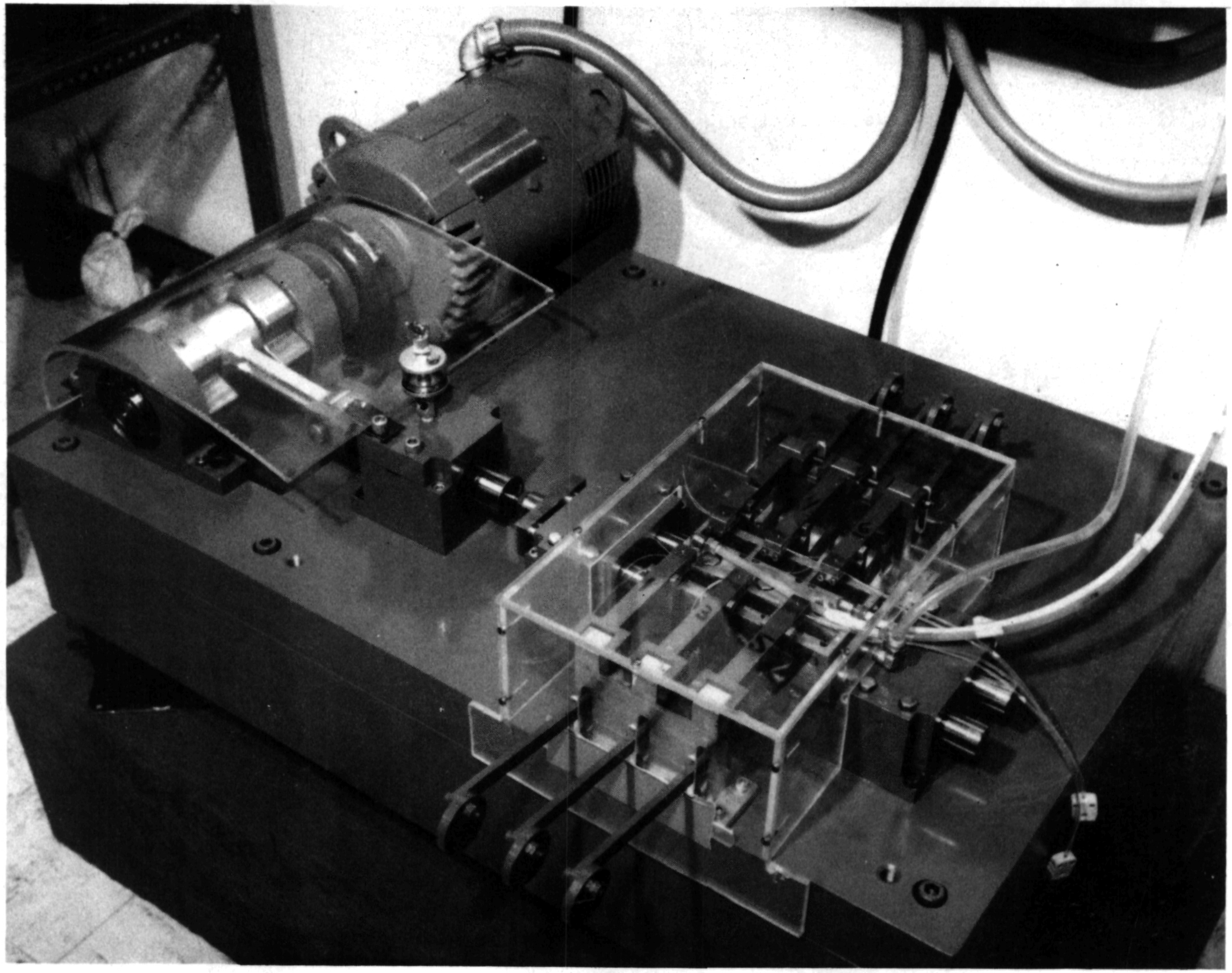

Fig. 4-38 Reciprocating Wear Test Rig

MTI-20035

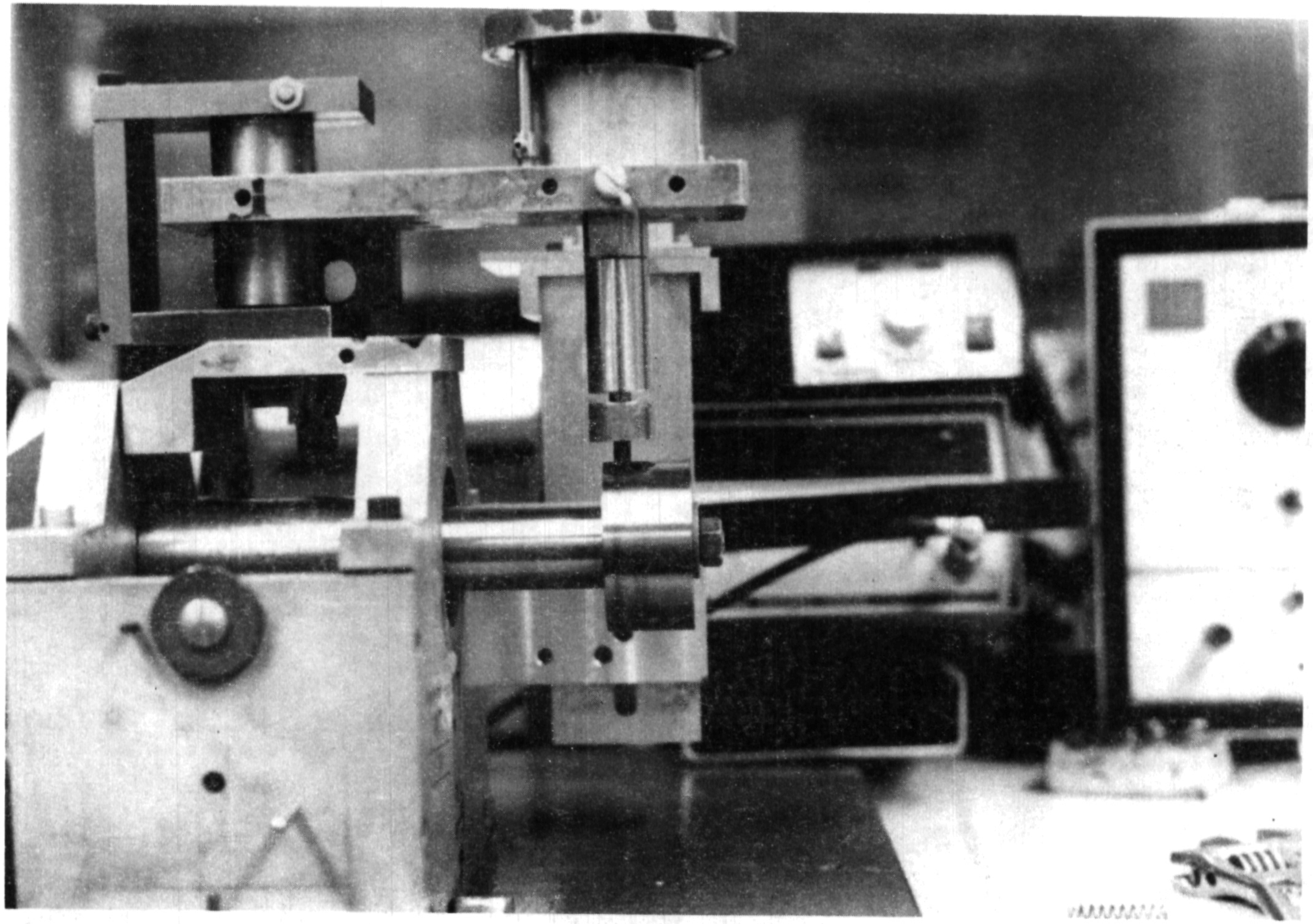

Fig. 4-39 Unidirectional Sliding Friction Tester

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TABLE 4-5

MATERIALS FOR SEAL SCREENING TESTS

No.	Manufacturer	Trade Name	Base	Additives
1	Dixon	Rulon LD	PTFE*	GLF and other fillers
3	Dixon	Rulon E	PTFE	GLF and other fillers
13A	Dixon	TFE-GL-HL800-2	PTFE	10% GF/15% CGA
14A	Dixon	TFE-GF-HL800-2	PTFE	10% GF/10% CGA
32	Dupont	Vespel SP-211	PI**	-
33A	Koppers	K30W35 (wires normal)	PTFE	35% Copper Wire/ 10% Glass Filler
33B	Koppers	K30W35 (wires parallel)	PTFE	35% Copper Wire/ 10% Glass Filler
37	Simrit	PTFE 551	PTFE	-
40	Rogers	RT/Duriod 4300	PTFE	Glass Fiber

*PTFE - Polytetrafluoroethylene
**PI - Polyimide

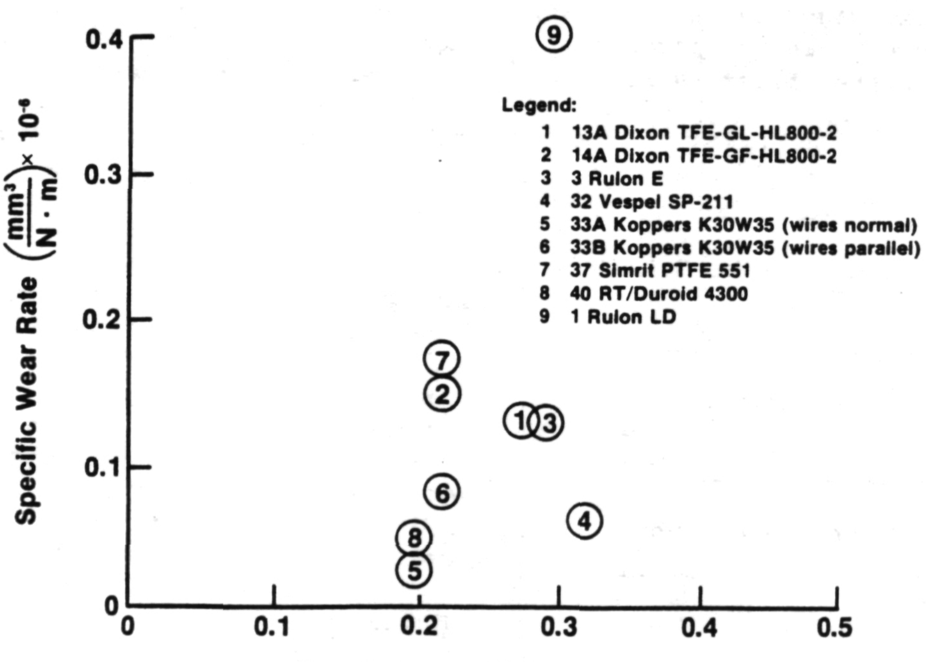

Fig. 4-40 Specific Wear Rate Versus Coefficient of Friction

surface. By loading the wicks with oil, a thin film of oil was introduced and maintained on the sliding surface. Three different seal materials were used in the tests: Rulon E, Dixon TFE-GL-HL-800-2, and Koppers K30W35 (wires normal). The load and speed were the same as in the screening tests (0.69 MPa and 1200 rpm).

The average wear rates for the three materials measured over 100 hours are given in Table 4-6 together with the values measured under dry conditions. All three materials exhibited a substantial increase in wear rate with the lubricant present. Examination of the steel coupons showed that the presence of the oil prevented the formation of a transfer film. Visual evidence also showed increased wear of the coupons.

To establish the full affect of the lubricant, a second series of tests was carried out in which the specimens were run-in under dry conditions for 10 hours to establish a transfer film before the oil was applied. In the presence of the oil, the transfer films degenerated rapidly; the wear rates were then comparable to those measured in the first series of tests.

TABLE 4-6

EFFECT OF LUBRICANT ON WEAR RATE

	Specific Wear Rate $(mm^3 \cdot N^{-1}\, m^{-1}) \times 10^{-6}$	
Material	Unlubricated	Lubricated
Koppers K30W35	0.029	0.426
Rulon E	0.130	0.635
Dixon TFE-GL-HL-800-2	0.130	0.635

The friction and wear measurements carried out in this program were aimed at providing a quick, simple evaluation of seal materials as a basis for the design and development of effective, long-life, low-friction, reciprocating seals. In the ongoing seal development program, selected materials will be evaluated in the form of actual seals. Several other significant results of these tests are:

- no particular seal wear advantages are to be gained from surface coatings; and,
- with both coated and uncoated surfaces, a similar initial run-in period of rapid wear as a transfer film was established, followed by a low, steady state wear level.

Piston Ring Rig Testing

A test rig was designed and fabricated for the primary evaluation of reciprocating seals under simulated engine conditions. The overall configuration of the rig is shown in Figure 4-41. The rig accommodates different seal test heads which are mounted on the crankcase/crosshead unit, which provides the reciprocating drive for the test heads. Three piezoelectric load cells located between the test head and the crankcase unit are used to measure the dynamic seal friction forces.

At this stage in the program, testing has been confined to piston rings, but heads are also available for cap seal and main seal testing. The piston ring test head is shown schematically in Figure 4-42. The piston houses four piston rings which are tested simultaneously. Three Rulon LD rider rings, symmetrically disposed, centralize the piston and prevent metallic contact between the piston and cylinder. Thermocouples in the cylinder liner allow the temperature to be monitored at the midstroke position of two of the test rings. A cooling water jacket surrounding the test head facilitates heat dissipation and temperature control. High-pressure gas (helium/hydrogen) is introduced between the upper and lower pairs of rings.

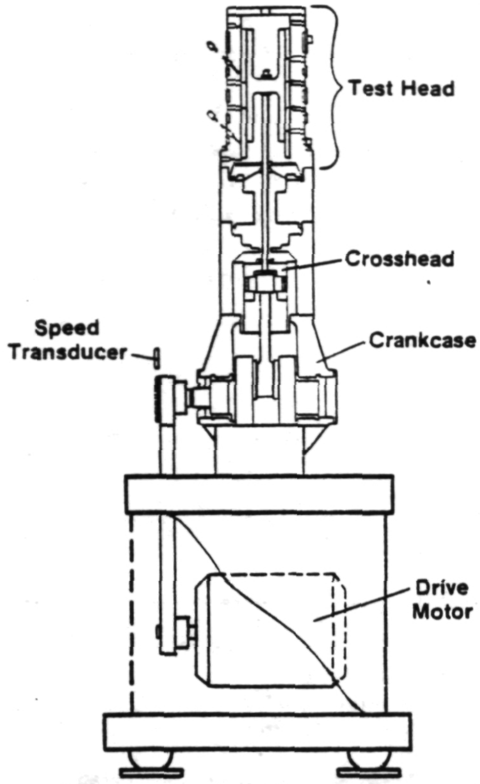

Fig. 4-41 Exploratory Seals Test Rig

Gas that leaks across the rings passes into sections that are vented into the atmosphere. The ports in these vented sections are also used to measure gas leak rate across each ring. To measure the leak rate for individual rings, nitrogen is introduced into the vented, downstream area of the ring at a known flow rate. The gas mixture (nitrogen plus helium/hydrogen) issuing from the exhaust port is passed into a mass spectrometer leak detector that has been calibrated to indicate the proportion of hydrogen or helium in the mixture. A flexible diaphragm above the crosshead maintains an oil-free environment in the piston ring test section.

To generate baseline data, initial testing was carried out on the proposed Mod I piston ring design, which is a continuous Rulon LD ring design manufactured to give an initial interference between the ring and cylinder wall. Additional ring loading is provided by the high-pressure gas which has access to the inner surface of the piston ring, as shown in Figure 4-43.

For initial evaluation, the rings were tested with helium at 1, 2, 3, and 4 MPa, and at speeds of 1000, 1500, and 2000 rpm. Test duration was two hours for each pressure/speed combination. The rig was

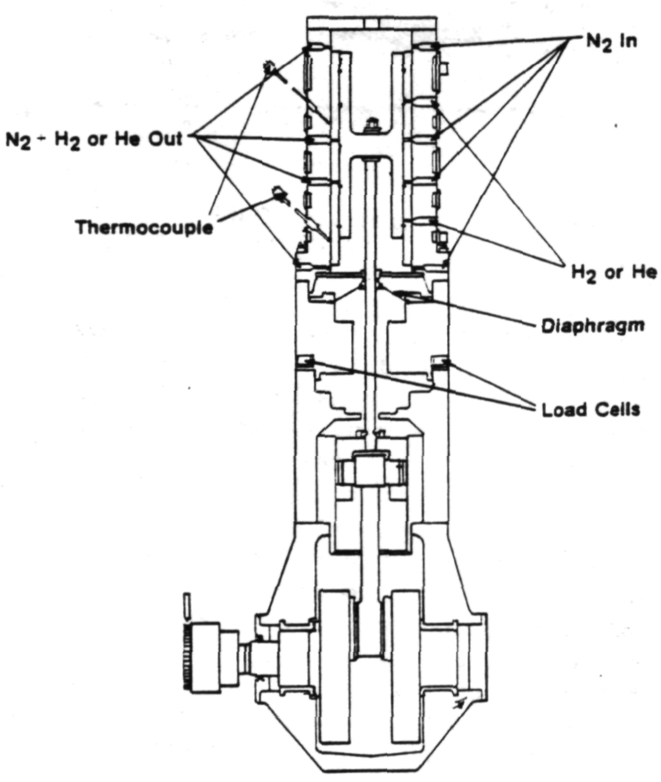

Fig. 4-42 Piston Ring Test Head

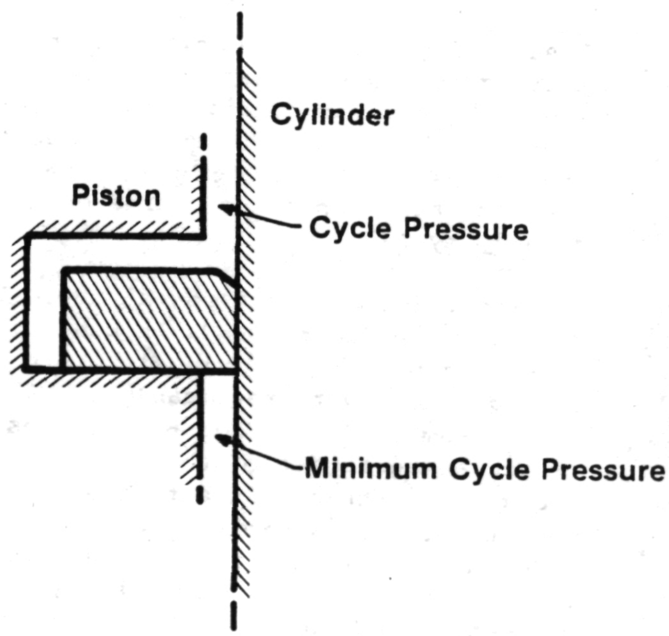

Fig. 4-43 Solid Piston Ring Design

stopped between tests and allowed to cool before starting the next test. Once the piston rings had sealed at the start of each test, the seals were maintained throughout the test, i.e., no complete failure of a ring was experienced during operation in an actual test. Life of the rings appeared to be determined by the number of cold starts. After completing a number of tests, an initial ring seal could not be achieved, and further testing had to be abandoned. Average piston ring life was 24 hours; wear during this period averaged 25% by weight. No obvious, progressive deterioration was apparent in the rings' performance during testing. Leak rates were low even before the rings failed to seal at start-up. The inability of the rings to seal was probably due to a combination of wear, cold flow, and contraction of the rings, resulting in large ring/cylinder clearances under cold conditions. Large leakage paths then prevented the buildup of pressure to expand the rings and produce a seal.

If testing had been carried out in a continuous manner, piston ring life would probably have increased; however, intermittent operation is more representative of the conditions to which the rings are subjected in an automotive application.

Piston ring life data derived from rig testing is a useful parameter for comparing different ring designs/materials; however, the data cannot be translated into the engine life of the rings since the rig doesn't fully reproduce engine environment or duty cycle.

Leakage/friction data for a typical test are shown in Figure 4-44. In general, leak rates across an individual piston ring during a test were reasonably constant and of the same order. From test to test, the relative magnitudes of the individual leak rates were not consistent; the ring with the highest leak rate in one test might have the lowest leak rate in a subsequent test. Apparently, the quality of the seal was determined at start-up, and did not change significantly during the test.

Leakage/friction data are summarized in Figures 4-45/4-46. Test data were measured with different sets of piston rings at various stages in their life which, coupled with inconsistent sealing of individual rings, accounts for the scatter; however, during their useful life, the rings gave leak rates that were acceptable in terms of their affect on engine performance.

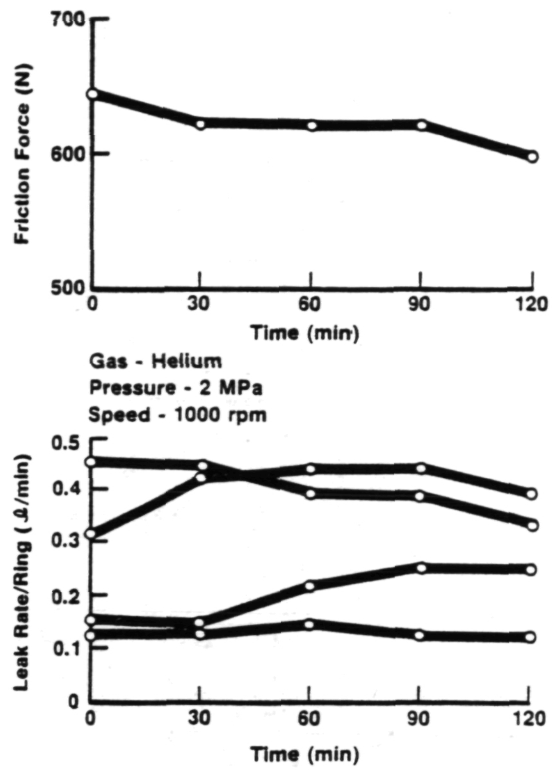

Fig. 4-44 Typical Test Data for Rulon LD Solid Piston Rings

For the tests described here, the cylinder liners were nodular cast iron with a 0.8 m (32 microinch) CLA finish, consistent with the engine design at that time. At the end of the test series, the liner was noticeably worn in the areas where the rings had been operating. To rectify this situation for further testing, the cylinder was honed. Following the honing, the surface finish could not be measured in the assembled condition, but it would certainly be better than 0.4 m (16 in.) CLA. To assess the effects of honing, repeat tests were conducted using helium (results are shown in Table 4-7). A comparison with Figures 4-45 and 46 shows that leakage and friction are significantly lower with the honed cylinder. In addition, average ring life was increased to 29 hours, and average wear was only 5% by weight. Life and wear values are based on limited experimental measurements, but a definite indication that honing is advantageous in all respects is apparent.

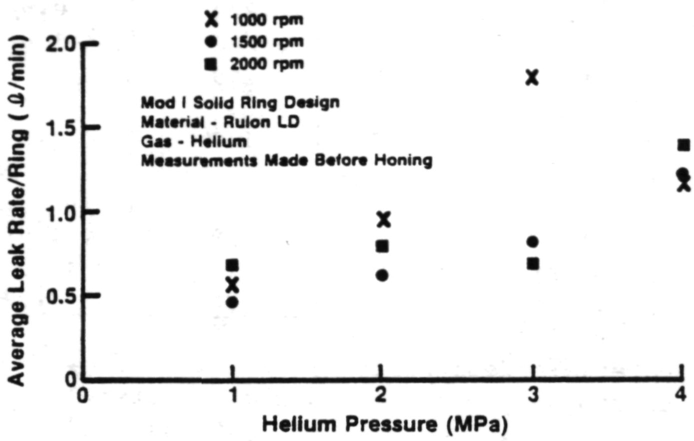

Fig. 4-45 Piston Ring Leakage

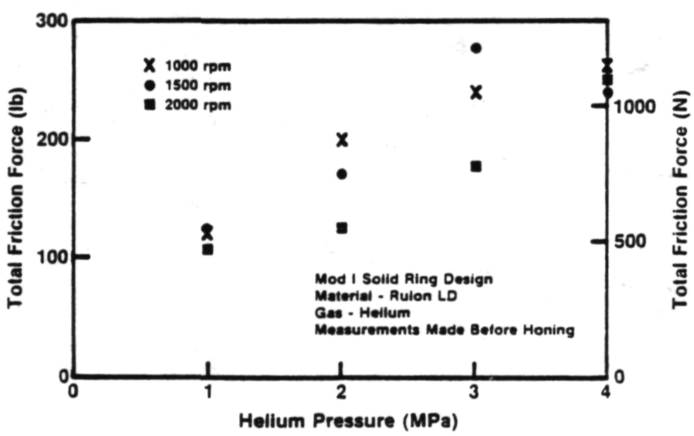

Fig. 4-46 Piston Ring Friction

To establish the influence of the working gas, additional tests were carried out using hydrogen. The leakage and friction data are summarized in Table 4-7. Compared with the data for helium, the leakage and friction values have less scatter; the friction forces are of the same order, but the hydrogen leak rate is generally lower than the helium leak rate. With hydrogen, the average ring life was 42 hours, and over this period, the average wear was 7% by weight. These limited data show that rings may perform better with hydrogen, and certainly not worse.

Two of the materials identified in the material screening tests as having lower wear and friction (Dixon TFE-GL and GF) were also tested in the piston ring test rig. The results are compared with the Rulon LD material in Table 4-7 for the Mod I solid design. While the friction measurements are consistent with the material screening results, the wear rates do not agree. While no clear experimental evidence was obtained, it is believed that reduced life may be due to the differences of the thermal properties of the materials. Rig testing can never reproduce the thermal operating conditions in an engine, but it is a valuable tool for primary evaluation of piston rings prior to further detailed study in the Mod I motoring rig and engines. As an example, the difficulty associated with achieving an initial ring seal demonstrated in the rig tests was also encountered in the prototype engine test, and a modified split/solid ring design had to be adopted to overcome this problem. The results of these tests are also included in Table 4-7.

The most promising design, however, is embodied in an improved design known as the H-ring, on which a patent application has been filed. This design uses pressure-balance piston rings whose friction force and leakage is independent of pressure, as shown in Figures 4-47 and 48; however, while the reduction in friction is significant, an increase in leakage also occurs. A modified H-ring, having higher contact pressure, has also been tested; it has increased friction but lower leakage. The results are also summarized in Table 4-7. Studies will be performed to determine the optimum tradeoff between leakage and friction. The wear results are quite encouraging to date.

TABLE 4-7

PISTON RING/WEAR DATA

Mod I Solid Ring Design

Design	Ring Material	Gas	Average Life (hrs)	Average Wear (%)	Est. Max. Friction Force (#)	Maximum Leakage (l/min)
Mod I Solid	Rulon LD*	H_e	24	25	280	< 1.0
Mod I Solid	Rulon LD	H_e	29	5	200	< 0.5
Mod I Solid	Rulon LD	H_2	42	7	200	< 1.0
Mod I Solid	Dixon TFE-GL-HL-800-2	H_2	15	7	210	< 1.0
Mod I Solid	Dixon TFE-GF-HL-800-2	H_2	46	7	180	< 0.5
Mod I Split/ Solid	Rulon LD	H_2	19	10%	100	< 1.0
"H"	Rulon LD	H_2	> 68**	7%	25	10.0
Modified "H"	Rulon LD	H_2	>110**	7%	60	8.0

*Results obtained before honing, all other materials/designs tested after honing.
**Rings were still functioning and starting up properly when test was terminated.

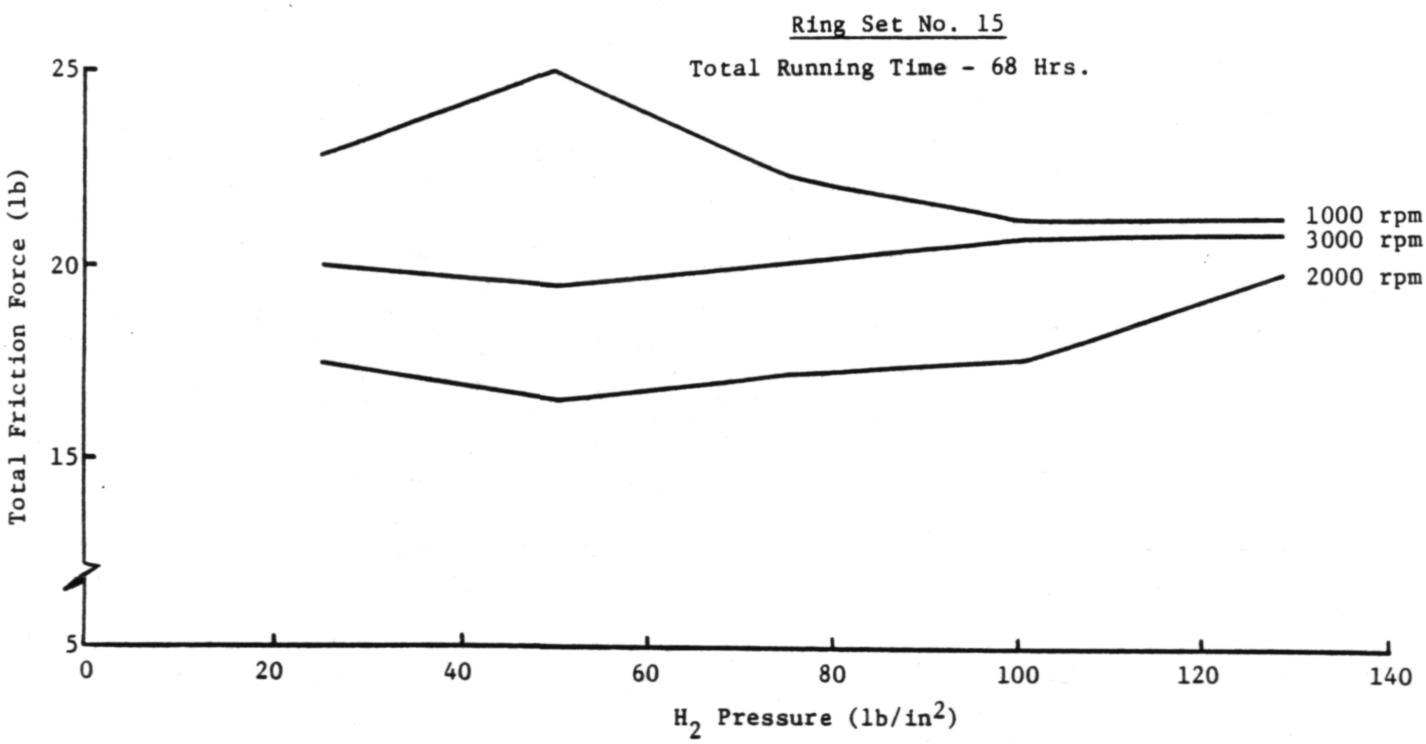

Fig. 4-47 H-Type Piston Ring Friction

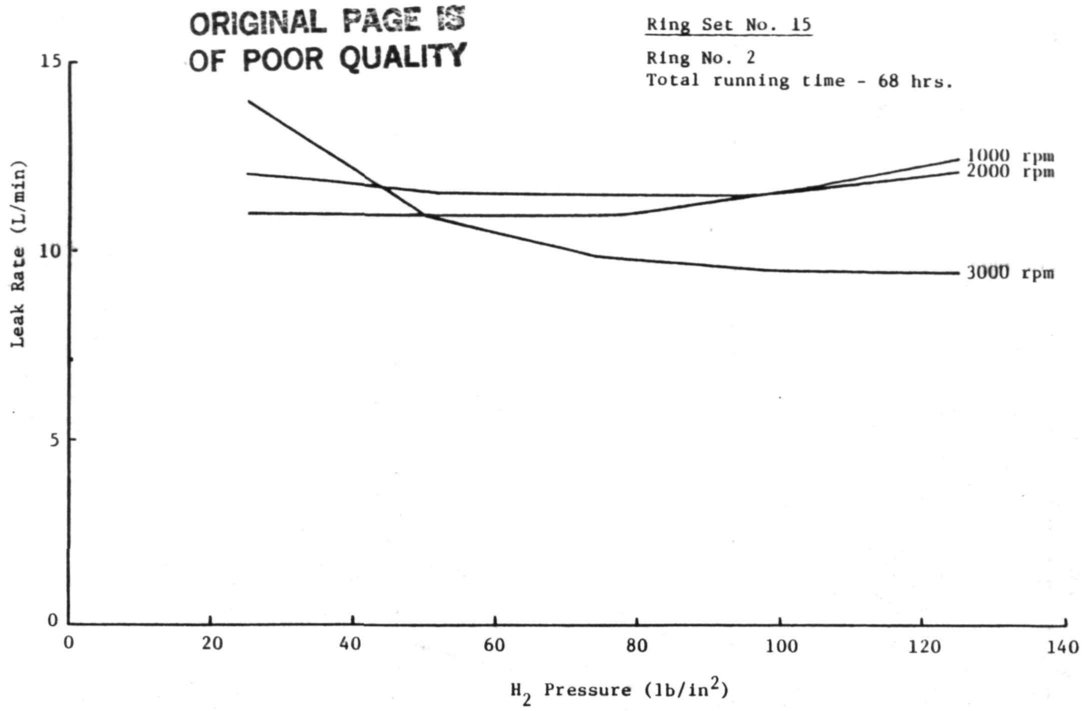

Fig. 4-48 H-Type Piston Ring Leakage

Seal/Drive Component Friction Distribution

An analytical study of the mechanical friction loss of the seal and drive components of the Mod I engine was conducted as an aid in the identification of components that need further development. These losses, subdivided into bearings, gears, crosshead, and seals, are summarized below.

Bearings

In computing the power losses in the bearings, the affect of loading was taken into account by computing the losses for a range of eccentricities that would be representative of what might occur for light and heavy duty. The affects of speed and clearance variations were also taken into account. An example of the type of analysis performed for each of the bearing types is that for the main bearings.

Main Bearings - For a given clearance, the power loss increases with speed and eccentricity. With the maximum diametral clearance, the power loss under heavy load conditions ($\varepsilon = 0.9$) is approximately twice that for a light load ($\varepsilon = 0.5$). For the average clearance, the affect of eccentricity is slightly less pronounced; however with a smaller clearance, there is an increase in power loss of 20-30% at all speed/eccentricity conditions (see Figure 4-49). For the worst case of $\varepsilon = 0.9$, C_d (diametrical clearance) = C_{dav}, the total power loss for the six main bearings is approximately 1.5 kW at 4000 rpm.

The main bearings are in the form of half shells located in a housing, and the manufacturing drawing calls for an assembled diametral clearance of 0-0.040 mm. If the assembled clearance is zero, the run-in period will be very critical, and a significant amount of wear may take place. As the engine warms up, this may become less critical due to the differential expansion of the steel crankshaft and the aluminum bearing housing. The overall affect of the expansion is difficult to establish with any certainty. A 50°C rise in temperature can increase the diametral clearance by 0.02 mm, tending to lower the power loss, but at cold start-up conditions, the power loss can be higher than predicted.

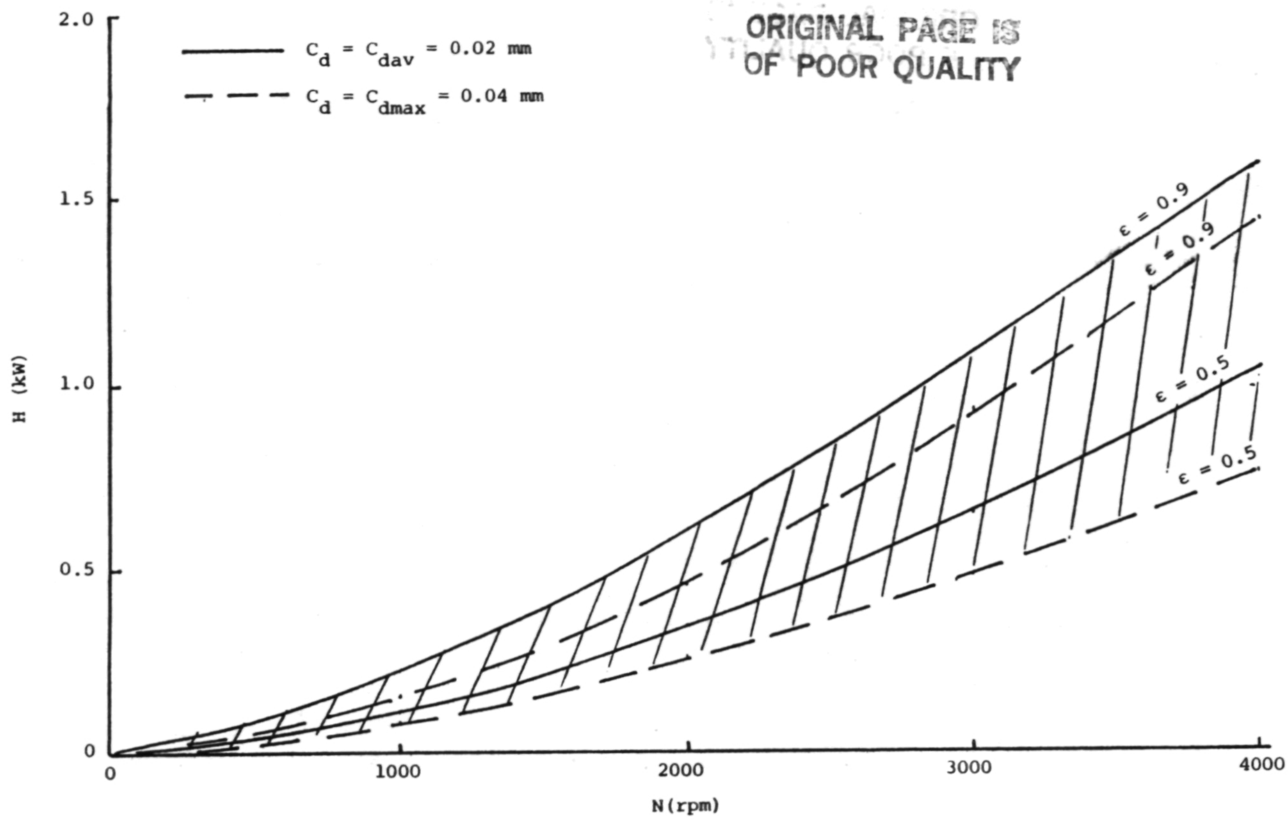

Fig. 4-49 Total Power Loss for Six Main Bearings

Gear Losses

Shearing of the lubricant film between the gear teeth was found to be the major source of power loss in the drive gears, increasing with speed and pressure (torque), as shown in Figure 4-50.

In the engine, drive torque from each crankshaft varies in a cyclic manner, but gear losses were computed assuming a constant torque equal to maximum crankshaft torque; therefore, one might assume that predicted losses will be an overestimate. This is probably not the case. Computations were based on high-quality gear (with negligible misalignment) data; the losses are typically 1% of the transmitted power. In the engine, it is unlikely that gear quality/alignment meet the assumed standards, and gear efficiency will be lower; therefore, predicted gear losses are probably underestimated.

Crosshead Losses

The range of power loss that can occur in the crossheads is shown in Figure 4-51. The diametral clearance can lie anywhere in the range of 0.03 to 0.06 mm; however, this has only a slight affect on the power loss. The increased shear rate at the smaller clearances gives a higher oil film temperature, and the reduction in viscosity offsets the affects of the smaller clearance. The curves for concentric operation represent the minimum power loss that will always occur and, for this condition, the power loss for C_d = 0.03 and 0.06 mm is almost identical. At 4000 rpm, the minimum power loss for the four crossheads is approximately 0.2 kW (.27 hp); with high eccentricity, the power loss can be as high as 0.6 kW (.80 hp).

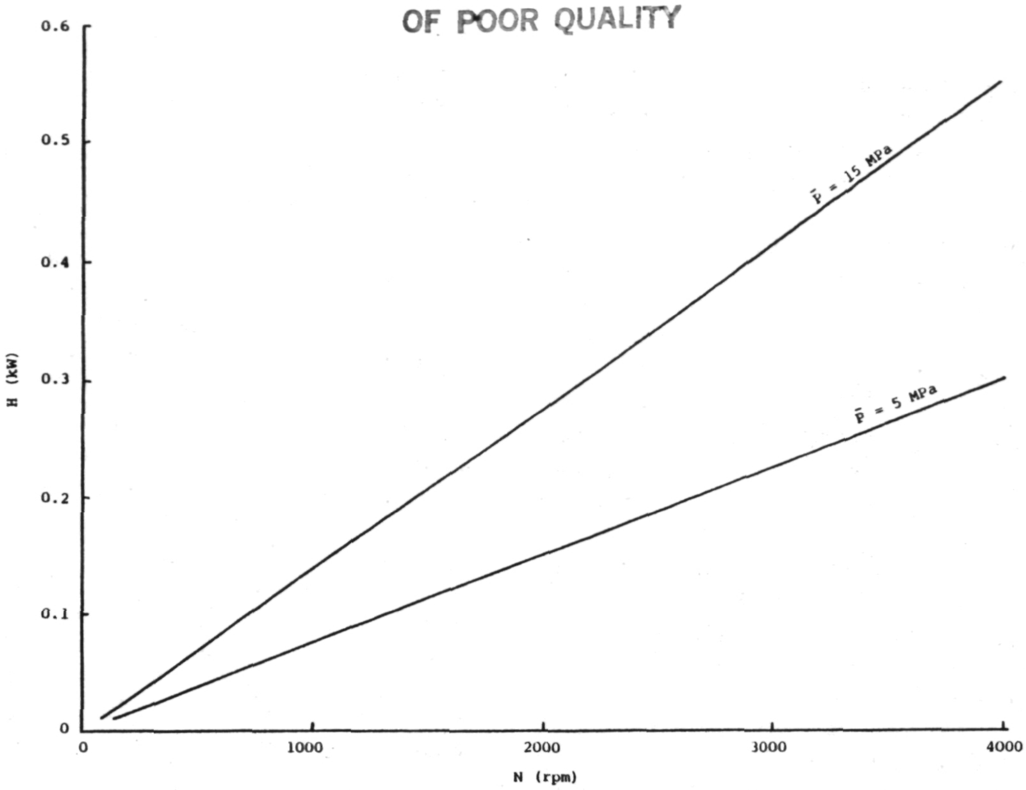

Fig. 4-50 Total Power Loss for Two Gear Meshes

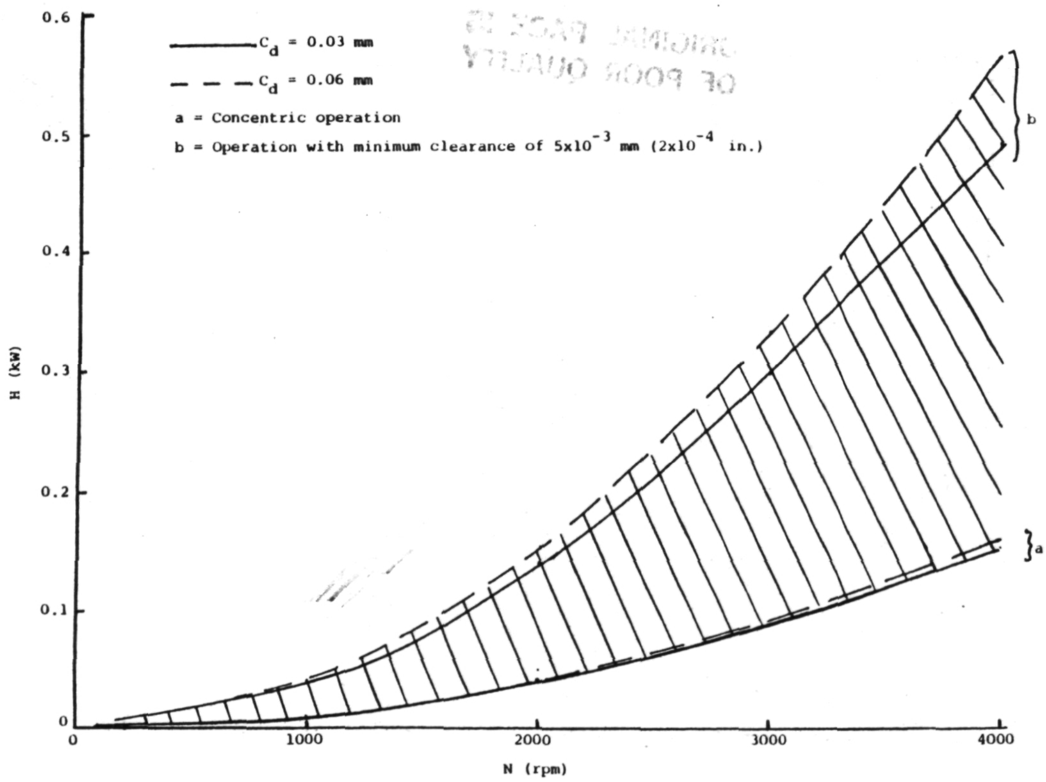

Fig. 4-51 Total Power Loss for Four Crossheads

Seal Losses

With the assumptions made in the analysis, the power loss for each seal is proportional to speed and mean pressure, as shown in Figures 4-52 through 4-54.

In predicting the power losses in the piston rings and cap seals, the coefficient of friction was assumed to be 0.2. From the evidence available for the seal materials, this value is about the minimum to expect under dry, sliding conditions. If the other assumptions made in the analysis are valid, the losses predicted for the piston rings and cap seals should not be overestimated; however, if there is a gas film established between the seal and mating surface during part of the cycle, the power loss will be reduced. With the reciprocating motion and cyclic changes in pressure loading, there is a distinct possibility of the seals being displaced or deflected in their housings; this could also influence the power loss.

The material used for the PL seal is a PTFE composite similar to that used for piston rings and cap seals, but with different fillers; therefore, this material is expected to have a dry, sliding coefficient of friction of the same order as that for the other seals, i.e., 0.2 minimum. With $\mu = 0.2$, N = 4000 rpm, and $\overline{P}^* =$ 15 MPa, the theoretical power loss in one PL seal will be ∿2.5 kW; under these conditions, the seals will degenerate very rapidly. Obviously, the effective coefficient of friction in the PL seal must be substantially less than 0.2. It can only be assumed that the reduction in friction is the result of the presence of an oil film on the piston rod from the coolant spray. Taking this into account, a value of $\mu = 0.05$ was used in computing the PL seal losses. This value is consistent with unpublished friction measurements made at NV Philips on PL seals, and is typical of the coefficients of friction which occur with boundary lubrication.

*mean working gas pressure

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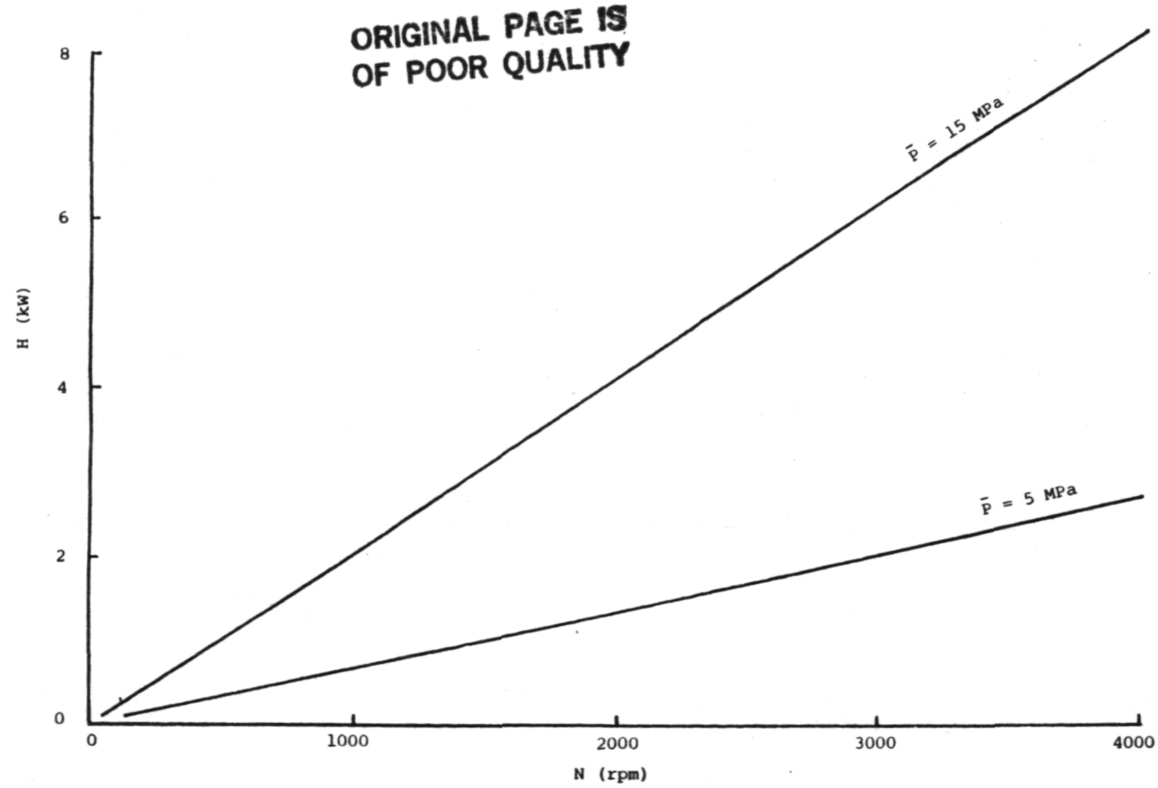

Fig. 4-52 Power Loss for Eight Piston Rings

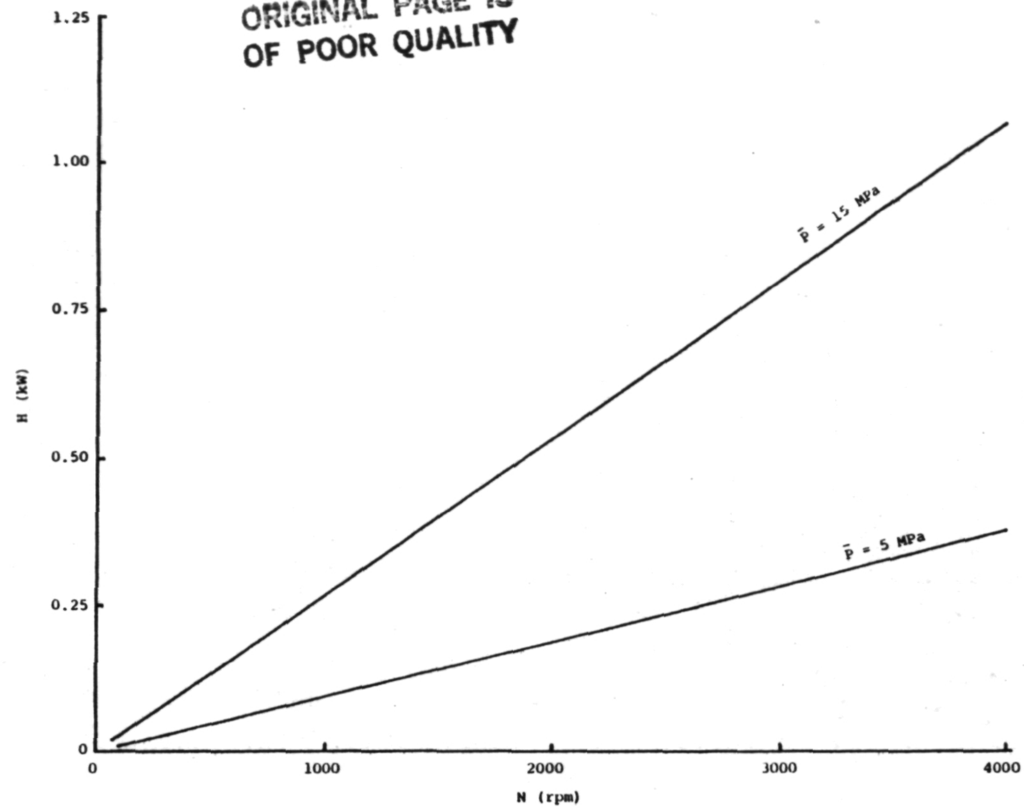

Fig. 4-53 Power Loss for Four Cap Seals

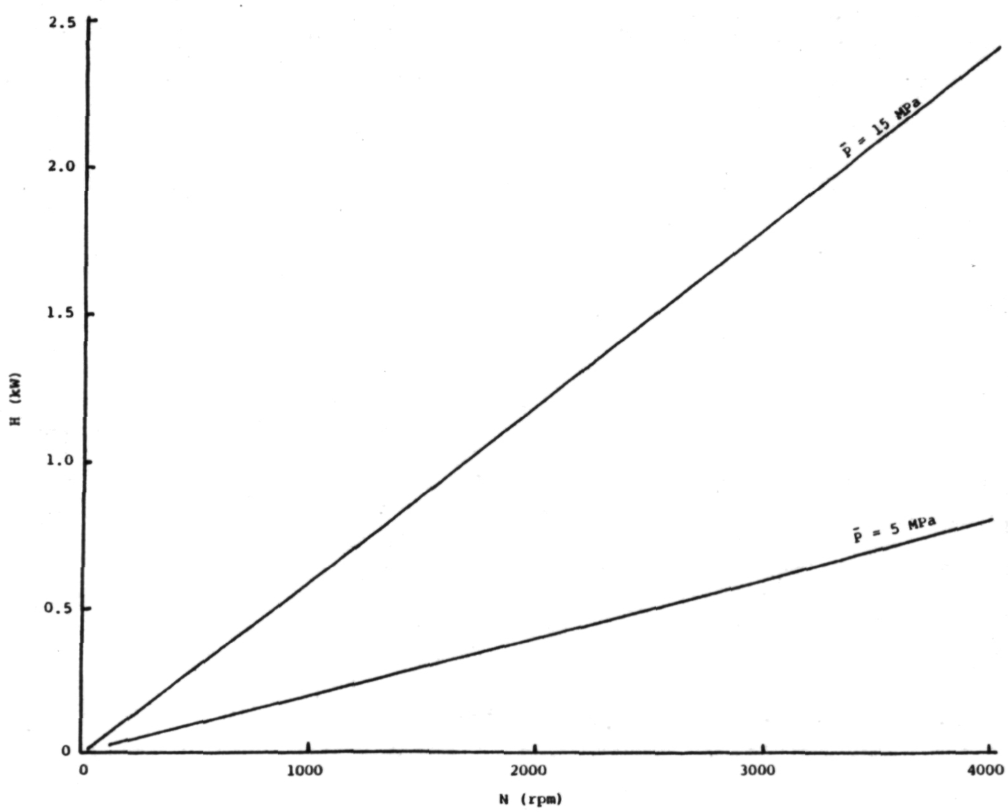

Fig. 4-54 Power Loss for Four Main Seals

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Total Losses

The losses in the bearings and crossheads are compared in Figures 4-55 for average loss conditions. At 4000 rpm, the total losses in bearings and crossheads varies from ∿2 to 4.5 kW (∿2.7 to 6.0 hp), with an average of about 3 kW (4 hp). Since the computations took into account the effects of the tolerances in the individual components, the average values are probably most representative of the total losses in the engine. In comparing the relative magnitudes of the different losses, it must be noted that the component losses presented are for groups of bearings, not for individual bearings. It can be concluded that the main bearings are the main source of loss, followed by crossheads, crankshaft thrust bearings, and crankpin bearings, which give approximately equal contributions. These "high-loss" sources should receive primary attention if any significant reduction in total bearing/crosshead losses is to be achieved, but a more detailed dynamic study of bearing loads/operating conditions will be required to determine whether this is possible.

Total seal losses are presented in Figure 4-56 for mean pressures of 15 MPa. As illustrated, the piston rings are the major source of power loss in the seals. The accuracy of the predicted seal losses is rather suspect, but the relative magnitudes of the seal losses should be of the right order.

Overall losses in bearings, crossheads, gears, and seals are detailed in Table 4-8 and summarized in Figure 4-57 (also gives total losses for mean pressures of 15 MPa). In each case, the average values of bearings/crosshead losses were used for reasons previously discussed.

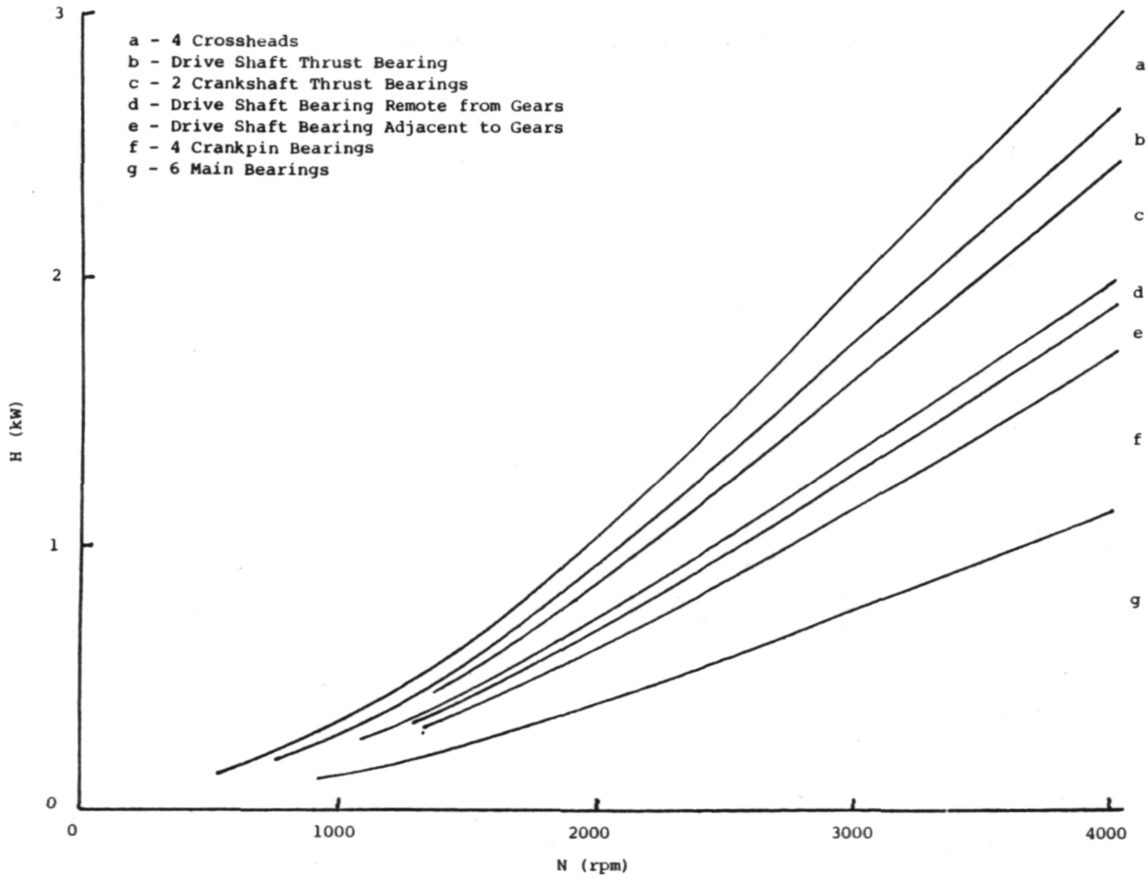

Fig. 4-55 Average Power Loss in Bearings/Crossheads

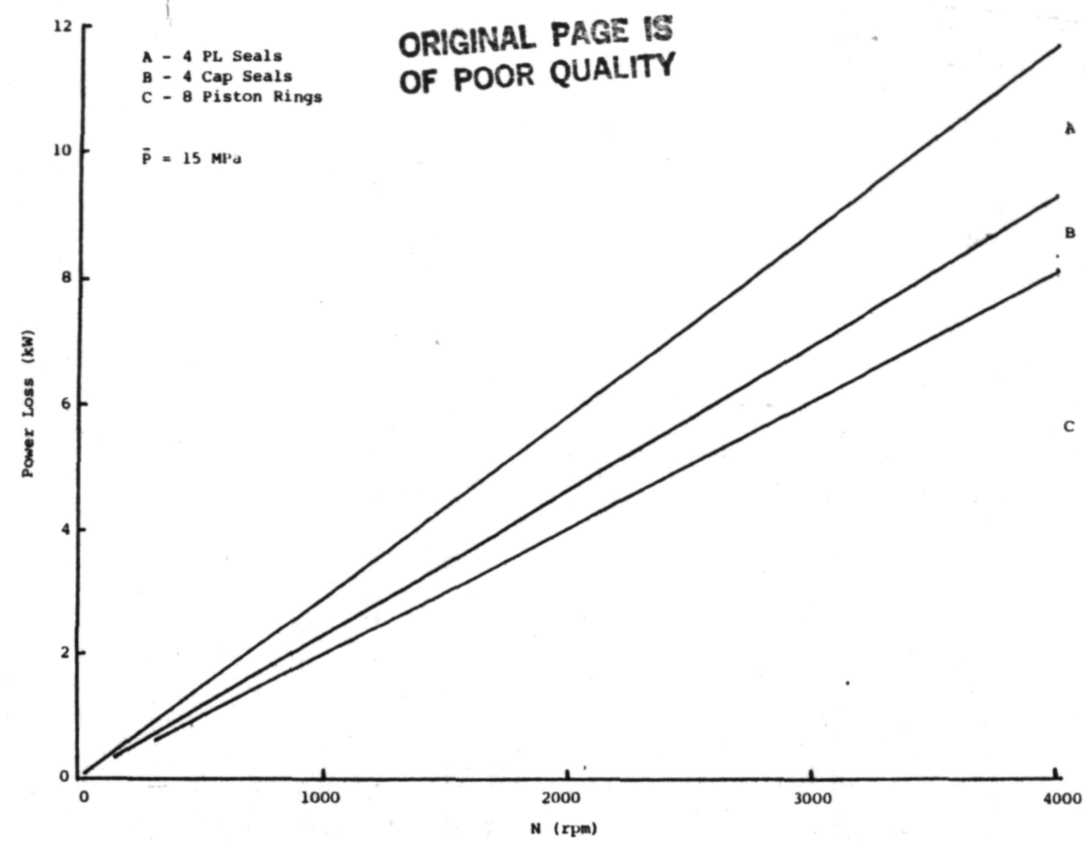

Fig. 4-56 Total Seal Loss

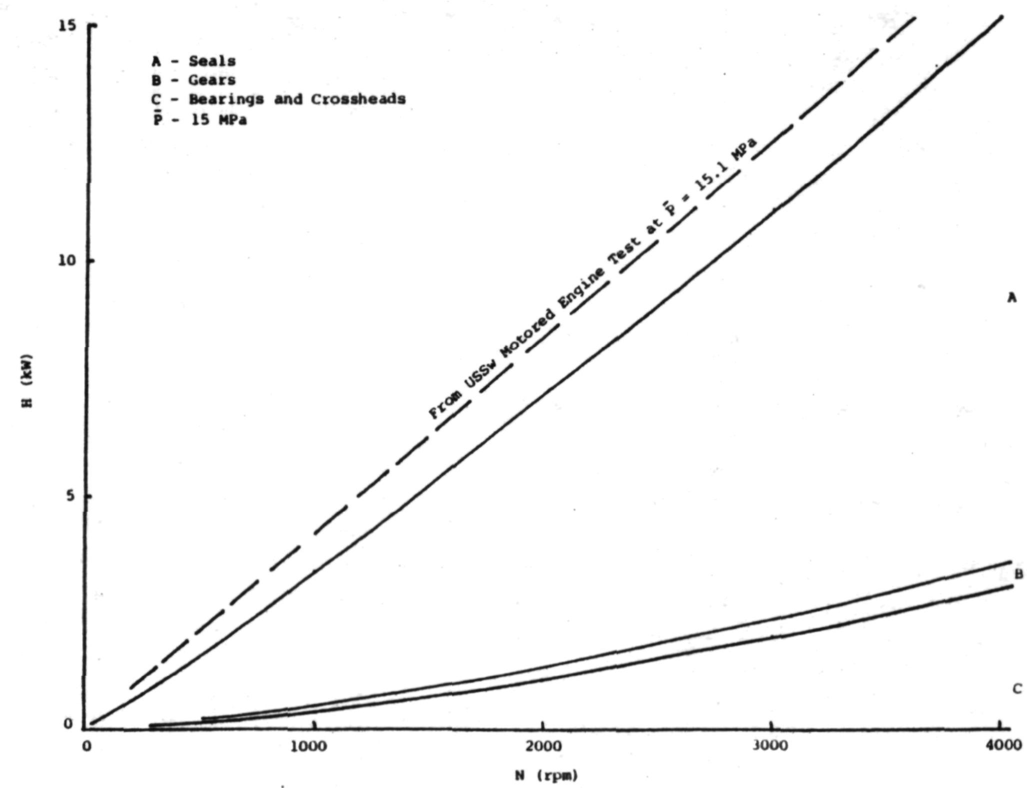

Fig. 4-57 Total Power Loss

TABLE 4-8

SUMMARY OF POWER LOSSES

	Power Loss (kW)											
Speed (rpm)	1000			2000			3000			4000		
$\overline{P}$ (MPa)	5	10	15	5	10	15	5	10	15	5	10	15
Piston Rings	0.68	1.36	2.03	1.36	2.71	4.07	2.03	4.07	6.10	2.71	5.42	8.14
Cap Seals	0.09	0.19	0.28	0.19	0.37	0.56	0.28	0.56	0.84	0.37	0.75	1.12
PL Seals	0.2	0.4	0.6	0.4	0.8	1.2	0.6	1.2	1.8	0.8	1.6	2.4
Σ Seals	0.97	1.95	2.91	1.95	3.88	5.83	2.91	5.83	8.74	3.88	7.77	11.66
Gears	0.07	0.11	0.14	0.15	0.21	0.28	0.22	0.32	0.41	0.30	0.43	0.55
Σ Seals + Gears	1.04	2.06	3.05	2.10	4.09	6.11	3.13	6.15	9.15	4.18	8.20	12.21
Bearings and Crossheads	0.18	-	0.52	0.59	-	1.51	1.14	-	2.82	1.66	-	4.25
Σ Seals + Gears +	1.22	-	-	2.69	-	-	4.47	-	-	5.84	-	-
Bearings + Crossheads Max.	-	-	3.57	-	-	7.62	-	-	11.97	-	-	16.46

At $\overline{P}$ = 5 MPa, the losses in the seals, and the average bearing/crosshead losses are approximately equal, whereas the gear losses are small in comparison. At 4000 rpm, the total system losses are approximately 7 kW (9.4 hp). At $\overline{P}$ = 15 MPa, the seal losses become the dominant component. At 4000 rpm and 15 MPa, the total system loss is approximately 15 kW (20 hp).

Figure 4-57 also includes power loss data from motored engine tests. These data were not direct measurements, but were derived from the measured drive power by deducting losses associated with the gas system which were predicted theoretically. At $\overline{P}$ = 15 MPa, the total predicted power loss is slightly less than that derived from the motored engine tests at $\overline{P}$ = 15.1 MPa, with the largest difference being approximately 1.5 kW (2 hp).

The predicted gear losses are probably optimistically low. Good gear alignment must be maintained to minimize these losses.

With the seals making a major contribution to total loss over the complete range of operating conditions, any reduction in seal friction will be directly reflected in overall engine efficiency. From these test results, the following conclusions can be drawn:

- the reciprocating seals are a major source of power loss in the engine and are dominant at high mean pressures;
- the piston rings are the major source of seal power loss;
- depending on operating conditions, bearing and crosshead losses can can account for up to 50% of total losses (these losses <u>might</u> be reduced by closer clearance control and improved design of the main bearings, crankpin bearings, and crossheads; in the combined main bearing/thrust bearing, oil temperature rise can be excessive); and,

- the predicted gear losses are small compared with other losses. (Actual gear losses may be higher depending on the quality of the gears and their alignment.)

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 in this task began during this report period, focusing on the installation and start-up of a motored Mod I drive system with a dummy heater head, the determination of the total mechanical component (bearings and seals) losses, and the preliminary design of a rolling element bottom end.

In 1982, effort will be concentrated on the fabrication, testing, and selection of a reduced friction drive, and the evaluation of new seal/piston ring designs.

Reduced Friction Drive

A Mod I engine drive with a dummy heater head was 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 133-4000 rpm at the EDS input shaft. Engine shaft speed and drive shaft speed are monitored with speed transducers. The speed signals will be compared to prevent safety clutch slippage. A torque transducer mounted in the driveline provides continuous torque input reading, 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 4-58. 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.

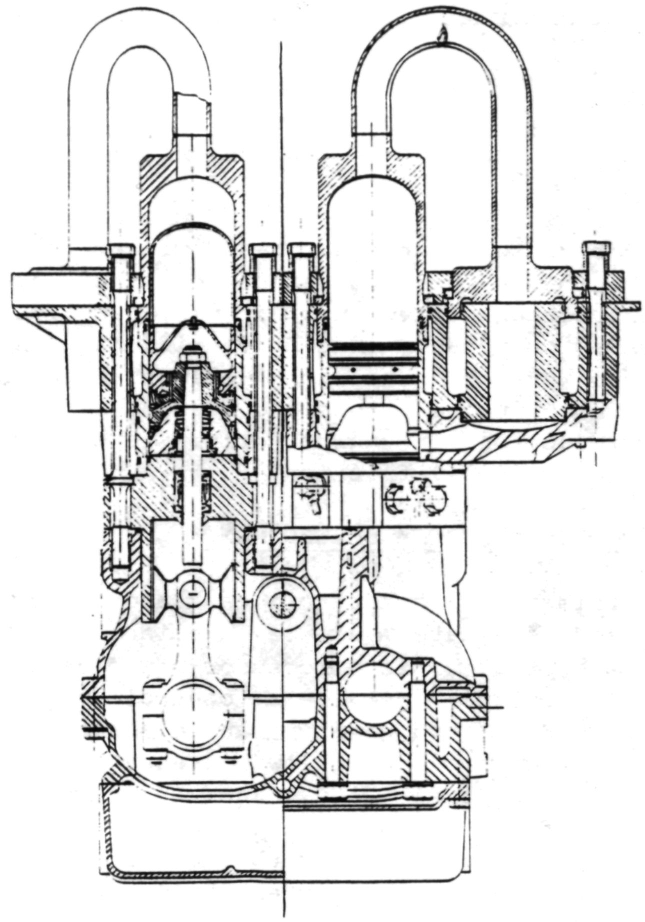

Fig. 4-58 Cross Section of Mod I EDS Motoring Unit

Baseline engine drive friction (seals and mechanical losses) was determined in both helium and hydrogen over a range of gas pressures; the results (Figure 4-59) compare favorably with USSw data at 9 MPa. In all cases, the purpose of using helium instead of hydrogen as a working gas is to increase the drive power requirements due to increased pumping losses.

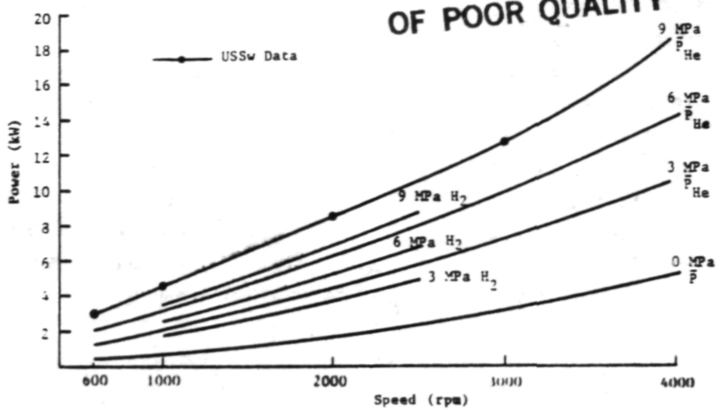

Fig. 4-59 Mod I EDS Baseline Drive and Seal Friction Losses

The results of these tests will be used as a basis for evaluation of the reduced friction drive, now in its preliminary design phase. This drive will make use of split-race, rolling element bearings, which project to yield a 1.3-kW (1.7-hp) reduction in engine drive (exclusive of seal losses) friction.

CONTROL SYSTEM DEVELOPMENT

Primary goals for this task are to develop and evaluate a simple, reliable, driver-acceptable, microprocessor-based electronic control; provide a control model for the EHS; and 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 activity focused on comparative testing of analog/digital controls on a test-cell P-40 engine under simulated engine transients, development/verification of an External Heat System Transient Reponse (EHSTR) Code, and initiation of the electronic air/fuel development.

In 1982, activity will focus on the fabrication, testing, and eventual selection of a prototype air/fuel control, upgrade of the EHSTR code to a Mod I configuration, upgrade of digital control software, and transient characterization of combustion control hardware.

Microprocessor Control

The Mod I Electronic Control Unit is a prototype digital system featuring printed circuit card and cage construction, ten modular boards, and an integral power supply. The microprocessor control architecture is shown in Figure 4-60; the electronics control hardware is shown in Figure 4-61. The system, based on Texas Instruments' TMS 9900 microprocessor (a 16-bit unit), provides:

- 3-Mc clock frequency;
- 8K RAM;
- 32K EPROM; and,
- A/D and D/A converters (32 single-ended signals).

All control functions are contained in an assembly-language modular software package. The various subroutines are called by interrupts in 10-ms intervals; total cycle time is 50 ms.

The system incorporates a 5-inch CRT monitor to be used during the development phase for adjustments in control algorithms. The monitor (equipped with an integral video generator and an ASC II keyboard) provides six displays that can be called from a menu: four parameter displays, a fault test display, and a change parameter display.

Control System Tests

In order to evaluate the performance of the Mod I Digital Electronic Control System with the control logic of subsystems, a sequence of tests was run on a P-40 Stirling engine to:

- obtain/compare baseline Analog and Digital Electronic Control response data with automotive requirements;
- verify the EHSTR code; and,
- evaluate sensitivity of engine response to changes in temperature/mean pressure control (MPC) set points and gains.

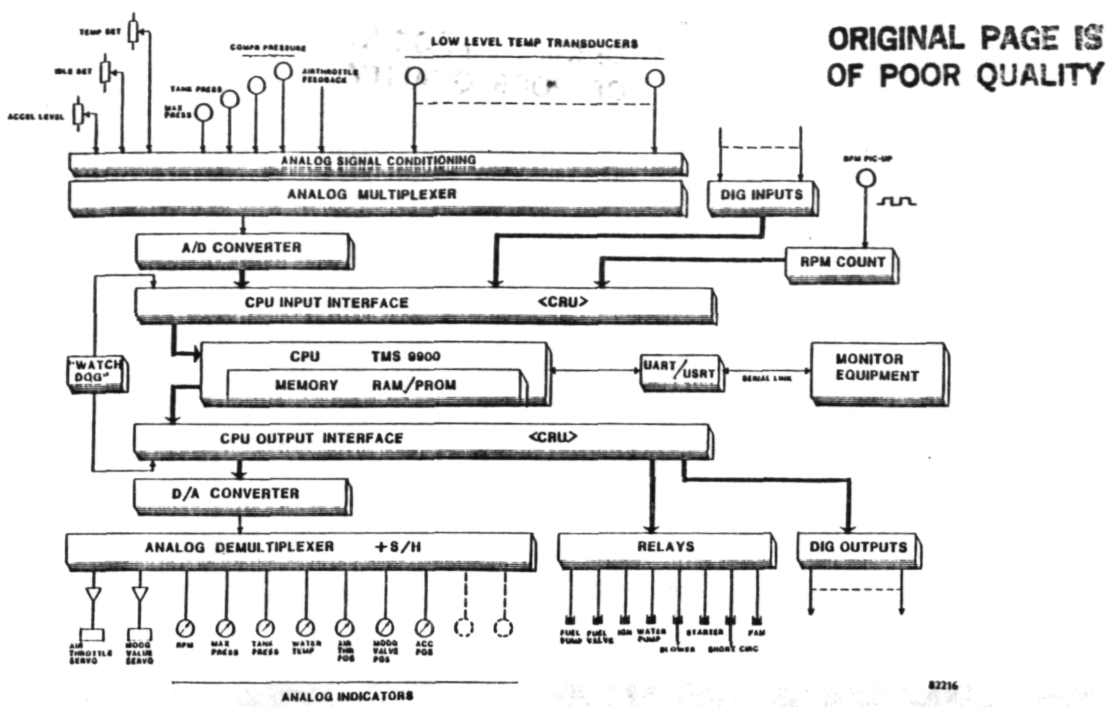

Fig. 4-60 Microprocessor Control Architecture

Actual engine tests were run on a dynamometer operating in both the maximum speed limit and constant-speed modes. Visicorder traces were made of pertinent control parameters.

The up-power trace for the microprocessor control testing provides a clear picture of the overall engine torque response process. As shown in Figure 4-62, demanded pressure (P_D) increases rapidly to change the MPC valve position (X_c) when the accelerator is depressed. Once the valve traverses through the dead band and gas begins to be supplied, $\hat{P}$ (transducer) rapidly increases with $\hat{P}$ (control) a few microprocessor cycles later. Once the desired 190-Bar pressure (P_D) is attained, valve moves into the dead band again and $\hat{P}$ overshoots slightly due to the time lag for $\hat{P}_{supply}$ and software filtering.

The transient response requirement for the Mod I control system, established on the basis of comparable IC engine response, states that there must be an increase in engine torque within 0.3 seconds after the accelerator is depressed, and that 90% of full torque must be attained 0.5 seconds later. As shown in Figure 4-63, a comparison of measured torque data with the desired response for both analog and digital controls indicates reasonable agreement and, in fact, suggests that an operator would not be able to detect the difference in acceleration between an IC engine and a Stirling engine. Conclusions from these tests are that the:

- Digital Control System operated satisfactorily as a prototype design;
- Digital and Analog Control Systems essentially exhibited the same stable operating characteristics at all steady state power operating points tested on the engine power performance map; and,
- prototype Digital Control System provides a more versatile, improved means of altering engine control constants and functions for optimization and engine guarding.

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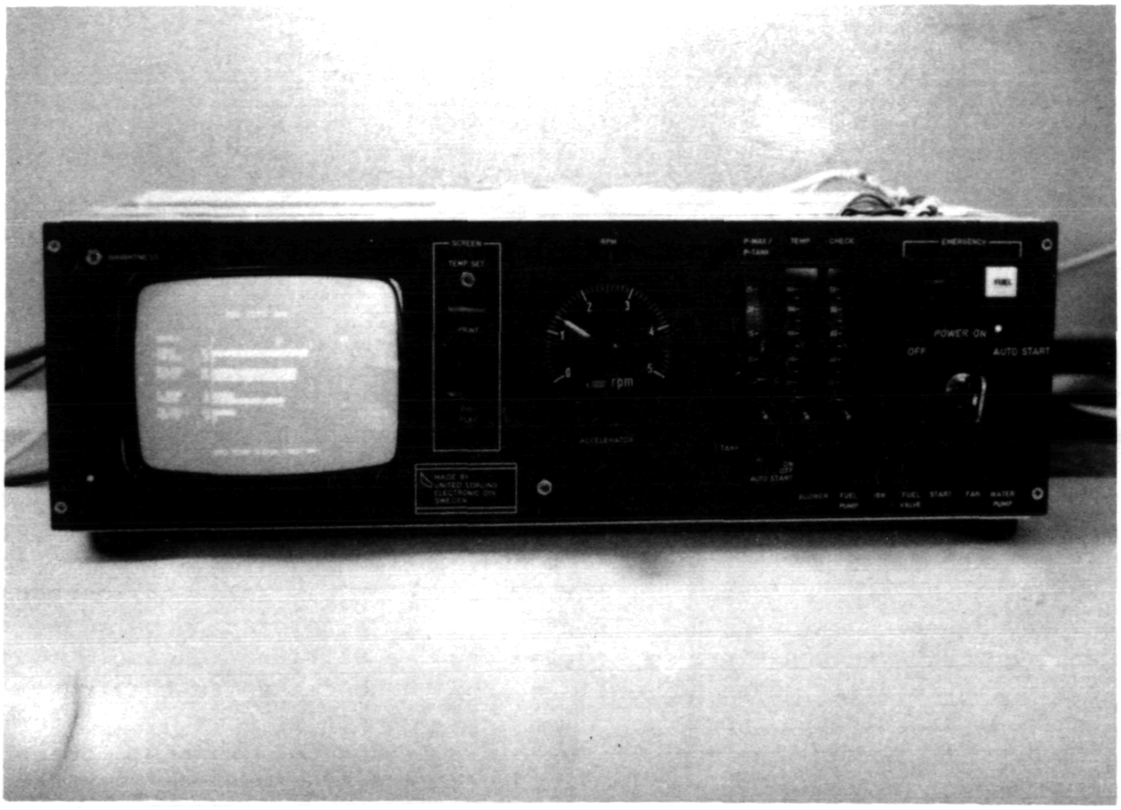

Fig. 4-61 Electronic Control Hardware

MTI-20737

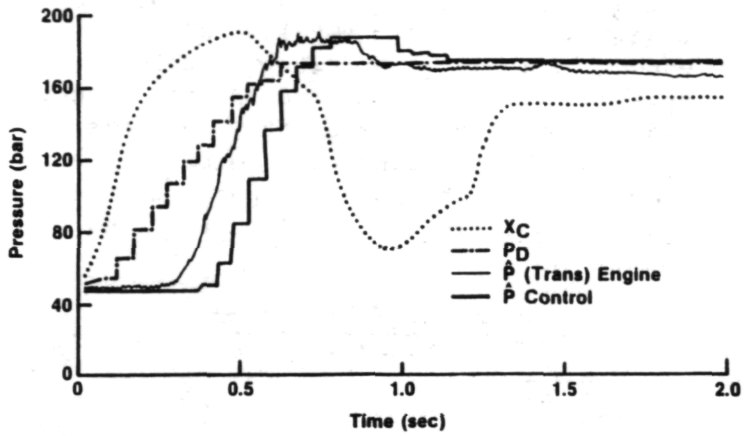

Fig. 4-62 Digital MPC Response

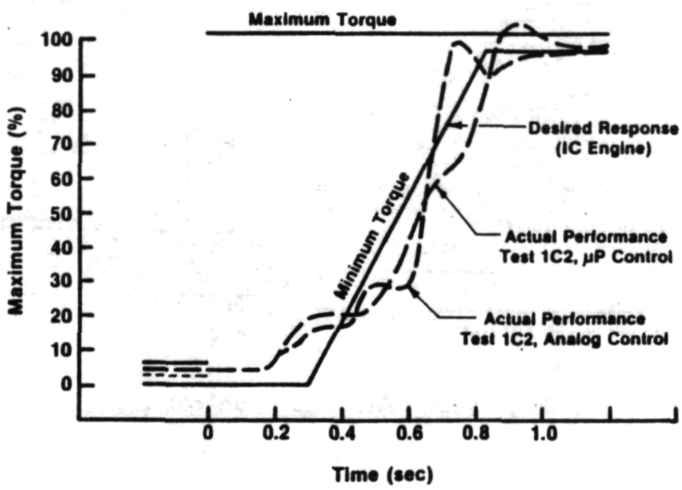

Fig. 4-63 Transient Specification Torque Envelope

EHSTR Code

The transient response of the EHS was examined using a simplistic model that included the blower, preheater, combustor, heater tubes, engine, control system transfer functions, and load (see Figure 4-64). Long-term transients (∿10 seconds) of the temperature control system were of interest rather than the short-term transients (∿1 seconds) of the MPC.

In order to develop and validate the model, Visicorder traces were obtained of engine temperatures as a function of time with impressed step changes in engine control parameters. A typical test examined the engine's response to an instantaneous up-power transient (see Figure 4-65). These results show the agreement between test data and model predictions for air throttle position, hydrogen temperature, and first- and second-row thermocouples. This overall agreement demonstrates that the model is adequate for predicting control system response, and can be used to determine air/fuel control strategy for the Mod I engine.

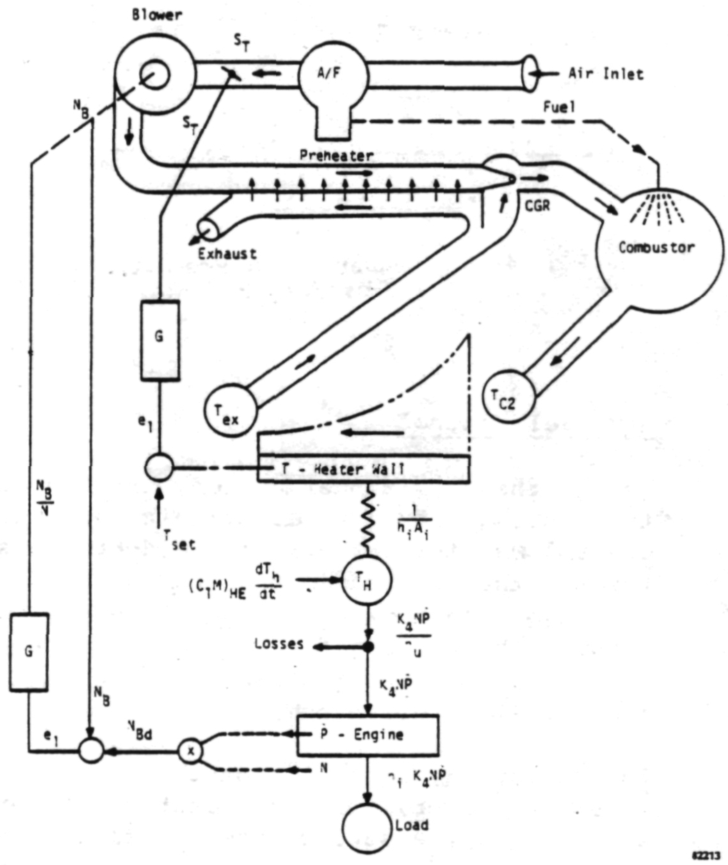

Fig. 4-64 EHS Schematic

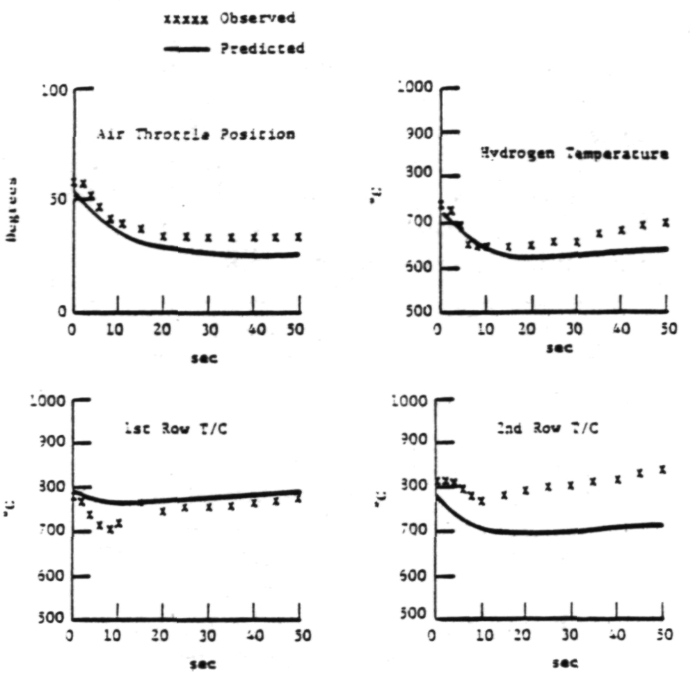

Fig. 4-65 Comparison of Predicted and Observed EHS Response

Air/Fuel Control

During this semiannual period, the conceptual design of the electronic air/fuel control was initiated. Figure 4-66 illustrates the control logic, and identifies the major sensing/controling elements, including the airflow transducer, air temperature sensor, air pressure transducer, and a fuel metering pump.

In operation, signals from the air pressure transducer, air temperature sensor, and airflow transducer enter a flow meter module that linearizes and compensates the flow transducer signal for different air temperatures and pressures. The resulting signal, which is proportional to air mass flow ($\dot{m}_{AIR}$), is converted to the desired fuel flow on an air/fuel ratio (λ) map. The shape of this curve depends on the demand for excess air.

The desired fuel flow is then converted to a desired pump speed (DPS) which, in turn, enters a fuel metering pump drive. By sensing the actual pump speed via a speed transducer, the drive exactly controls the speed of the metering pump motor. Due to the design of the metering pump, this fuel pump characteristic is a linear curve.

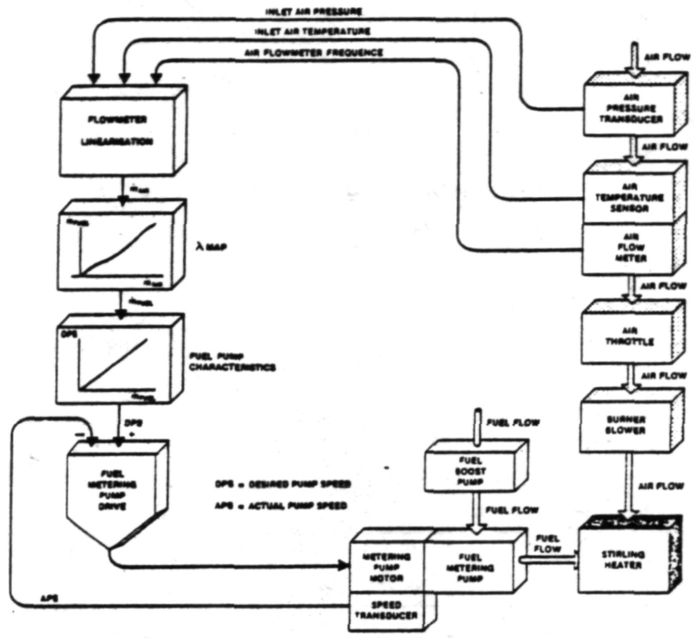

Fig. 4-66 Air/Fuel Control Logic

V. COMPUTER CODE DEVELOPMENT

A computer program specifically tailored to predict SES steady state cyclical performance over the complete range of engine operations has been developed. Using data from component, subsystem, and engine system test activities, the program will be continuously improved and verified throughout the course of the ASE Program. The steady state performance code was delivered prior to this report period; however, validation of the code was initiated during this time frame.

Code Validation

Validation of the steady state code is being accomplished by comparing results from this code to results obtained from a similar existing (but less detailed) code developed and proven by USSw. An example of the correlation established is shown in Figure 5-1 for the conditions noted. Performance for various levels of a friction/heat transfer enhancement factor for the code is indicated. This particular factor models the augmentation of steady state friction and heat transfer resulting from periodic flow effects. Further evaluation is planned during the first half of 1982.

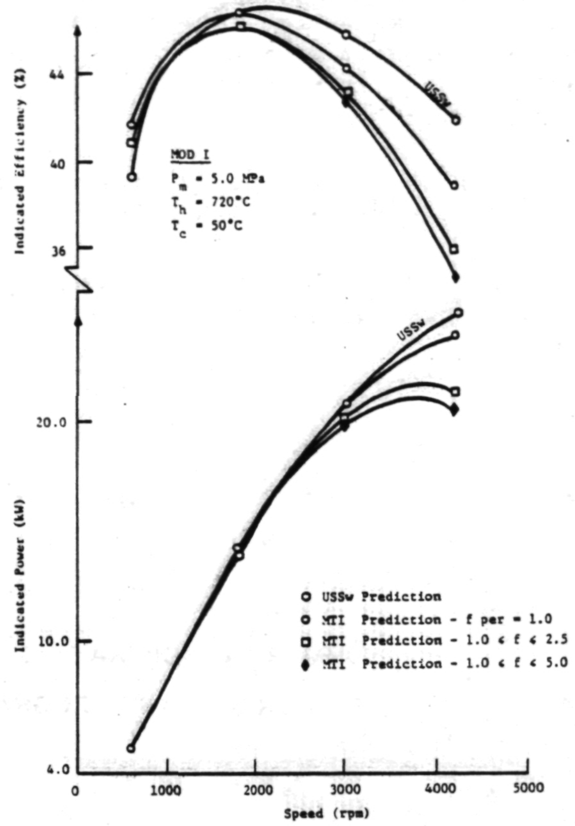

Fig. 5-1 Mod I Engine Performance (USSw/MTI Code Comparison)

VI. ENGINE OPERATING HISTORY

Table 6-1 is a summary of the operating times and mean time between failures for all ASE Program engines as of December 31, 1981. The primary usage of each engine is as follows:

- ASE 40-1: Engine Dynamometer Testing at NASA;
- ASE 40-4: High-Temperature (820°C) Endurance Testing at USSw;
- ASE 40-7: Engine Dynamometer Testing at MTI:
- ASE 40-8: Vehicle Testing at AMG;
- ASE 40-12: Vehicle Demonstrations at MTI;
- ASE Mod I Engine #1: Dynamometer Testing at USSw; and,
- ASE Mod I Engine #2: Dynamometer Testing at USSw.

TABLE 6-1

SUMMARY OF OPERATING TIMES AND MEAN TIME BETWEEN FAILURES FOR ALL ASE PROGRAM ENGINES AS OF DECEMBER 31, 1981

Engine		Operation Time	Mean Operating Time Between Failure
ASE 40-1	(NASA)	261.8[1]	6.23
ASE 40-4	(USSw)	6540.9[1]	97.63
ASE 40-7	(MTI)	417.1[1]	11.29
ASE 40-8	(Spirit (AMG))	293.7[1]	3.53
ASE 40-12	(Concord (MTI))	182.6[1]	14.05
ASE Mod I #1	(USSw)	336.6[2]	168.30
ASE Mod I #3	(USSw)	85.8[2]	42.90

[1]hours after completion of acceptance test
[2]total hours

PUBLISHED REPORTS

The following reports were issued during this semiannual report period:

81ASE198TP33: "Acceptance Test of Mod I Stirling Engine System"

81ASE204ER5: "Users Manual for STENSY Interim #1"

79ASE54TS1: "Vehicle Definition for Reference Engine System Design"

81ASE200MT37: "Monthly Technical Progress Report for April, 1981

81ASE214MF34: "533P for June, 1981"

81ASE201TP34: Test plan entitled "Analog and Digital Control System Testing of ASE 40-7"

81ASE205ER26: "A Computer Code for the Prediction of Bottom End Hoses in a Stirling Engine"

81ASE206MT38: "Monthly Technical Progress Report for May, 1981

81ASE207ER27: "Friction and Wear of Polymeric Composites for Reciprocating Stirling Engine Seals: Part III, Further Screening Tests"

81ASE215MT39: "Monthly Technical Progress Report for June, 1981

81ASE218TP35: "Vehicle Control Transient Test Plan"

81ASE220MT40: "Monthly Technical Progress Report for July, 1981

81ASE190TP29: "Fatigue Testing of Heater Alloys (by MTI)"

81ASE219MF40: "533P for July, 1981"

81ASE196ER18: "MTI Variable-stroke Concept Study"

81ASE216ER28: "Experiments Showing Capability of Iron/Nickel Base Alloy"

81ASE217ER29: "ASE P-40-7 at MTI"

81ASE237ER32: "USSw Acceptance Test Report for Mod I SES No. 1"

81ASE227MT41: "Monthly Technical Progress Report for August, 1981"

81ASE228MF41: "533P for August, 1981"

81ASE234MT42: Monthly Technical Progress Report for September, 1981.

81ASE232QF11: "533Q for July/August/September, 1981"

81ASE239MT43: Monthly Technical Progress Report for October, 1981.

81ASE233ER31: "Air Mass Flow Evaluation - 1981 Lerma with Mod I Mockup from AMG"

81ASE235MF42: 533P for September, 1981"

81ASE231WP7: "Work Plan for Program Year 1982

CCM Papers

81ASE221PR14: "Seals Development"

81ASE222PR15: "Mod I - U.S.A. Build"

81ASE223PR16: "Regenerator Development"

81ASE224PR17: "Mod I Testing"

81ASE225PR18: "ASE Program Status"

81ASE226PR19: "Controls Development"

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15. Supplementary Notes

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NASA Project Manager - William K. Tabata, Transportation Propulsion Division, NASA/Lewis Research Center, 21000 Brookpark Road, Cleveland, Ohio, 44135

16. Abstract

This is the first semiannual technical progress report prepared under the Automotive Stirling Engine Development Program; it covers the fourteenth and fifteenth quarters of activity after award of the contract. Quarterly technical progress reports reported program activities from the first quarter through the thirteenth quarter; thereafter, reporting was changed to a semiannual format.

This report summarizes activities performed on Mod I engine testing and test results, progress in manufacturing, assembling and testing of a Mod I engine in the United States, P-40 Stirling engine dynamometer and multifuels testing, analog/digital controls system testing, Stirling reference engine manufacturing and reduced size studies, components and subsystems, computer code development activities. The overall program philosophy is outlined, and data and results are presented.

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