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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

SEMIANNUAL TECHNICAL PROGRESS REPORT FOR PERIOD: JULY 1 – DECEMBER 31, 1983

Mechanical Technology Incorporated

March 1984

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
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NOTICE

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MTI Report No. 84ASE369SA5

Semiannual Technical Progress Narrative Report
for Period of July 1 - December 31, 1983

March, 1984

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INTRODUCTION

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

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

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

The Mod I and Mod II engines are experimental versions of the RESD. The Mod I (the first engine design) used existing technologies embodied in USAB's P-40 and P-75 engines that existed at the beginning of the program. The Mod II (the second engine design) was planned based on the RESD, Mod I, and component/technology development improvements made during the course of the program. It is the engine intended to meet the final ASE Program objectives; however, the Mod II was postponed in 1981 due to Government funding cutbacks, making the Mod I the only experimental engine in the program.

During the succeeding years, the Mod I was modified and upgraded, wherever possible, to develop and demonstrate technologies embodied in the RESD. As a result, a "proof-of-concept" logic evolved whereby the upgraded Mod I design emerged as an improved engine system, "proving" specific design concepts and technologies in the RESD that were not in the original Mod I design, but this logic was recognized as having inherent limitations when it came to actual engine hardware, since Mod I hardware was larger and, in some cases, of a fundamentally different design than that of the RESD or Mod II.

The upgraded Mod I, however, incorporated a few new technologies that existed in the RESD, i.e., the use of new iron-based materials in the Hot Engine System in place of costly cobalt-based materials. The upgraded Mod I was designed to operate at 820°C heater head temperature as was the RESD, whereas the Mod I was tested at 720°C. Smaller, lighter designs were incorporated into the upgraded engine to optimize for better fuel economy and reduced weight (the upraded Mod I engine was 100 lbs. lighter than the Mod I engine).

A design review was conducted on an RESD in March, 1981 that provided the focal point for development during fiscal years 1981, 1982, and 1983. This changed in May, 1983 when the RESD was updated to incorporate more advanced designs and reduce the manufacturing cost to a level comparable with spark-ignition and diesel engines. As a result of this update, new concepts and technologies were identified that required immediate attention and development if they were to be ready for the initiation of a Mod II engine program.

The May, 1983 updated RESD has a predicted combined mileage of 41.1 mpg using unleaded gasoline, which is 50% above the projected spark-ignition engine mileage for a 1984 X-body vehicle with a curb weight of 2870 pounds. The significant point was that the manufacturing cost had been reduced more than 25%, while other design parameters such as engine efficiency, weight, power density, and power remained approximately the same. To achieve this important redesign, the RESD configuration was changed considerably from the previous design, i.e., the updated design uses: 1) a V-drive system rather than a U-drive system, as used in the Mod I; 2) an annular heater head rather than the cannister configuration used in the Mod I; and, 3) a simplified control system and auxiliary components. It became apparent that new technologies were being called for through these changes, and that a number of development efforts needed to be initiated. These efforts began during the last six months of 1983, and are documented in this report.

During this semiannual report period, the ASE Program has expanded into developing new technologies for the RESD/Mod II. In order to evaluate and understand the differences between cannister and annular heater head geometry, the P-40R engine, a modified P-40 engine utilizing an annular heater head whereby the regenerator and cooler surround the cylinder in an annular fashion similar in concept to the 1983 updated RESD, was reinstated into the program. The P-40R, fabricated and tested earlier in the program and then discontinued, will provide valuable data on the difference between the Combustion System and Hot Engine System performance of annular and cannister configurations as used on P-40 and Mod I engines. The P-40R still utilizes a U-drive system.

During the last six months of 1983, the P-40R engine hardware was pulled from storage, spare parts were manufactured, and the engine was assembled. Initial tests were conducted to establish baseline performance data from which development changes could be evaluated.

The first test (currently ongoing) will center around an understanding of the radial and axial losses through the partition wall, which separates the expansion cylinder space from the annular regenerator.

The second major new technology associated with the updated RESD is the use of a single-shaft V-drive system. Specifically, concerns center around the design concept of using a single-piece, cast-engine block that is more compatible with automotive practice. A preliminary design was completed and reviewed with automotive-industry casting foundries to determine proper specifications for the quality. The technical challenge is whether the proper casting quality can be obtained in a portion of the casting where pressurized working gas must be contained. Since hydrogen permeates metal, this problem represents a significant technical challenge. This manufacturing technology must be developed if the RESD/Mod II is to meet its specific weight goal.

A major program milestone calling for the characterization of the upgraded Mod I engine by September 30, 1983 has been met. An upgraded Mod I engine scheduled for delivery to General Motors under the Industry Test and Evaluation Program (ITEP) was assembled and acceptance tested. This engine will be installed in a Spirit vehicle (also modified and prepared for delivery to General Motors in April, 1984). Hardware for a second engine, scheduled for delivery to John Deere & Co., was also procured during this report period under the ITEP.

Major advances were made during this semiannual report period in determining the influence of crosshead guide clearance on the life of main rod seals. With tight clearances of 20-40 microns, while running in the hot-condition, seal life demonstrations in P-40 engines of greater than 1000 hours were achieved.

At this point in time, six Mod I engines are testing within the program, each with a specific purpose of providing crucial data toward technology development. A total of 4570 hours have been accumulated to date within the Mod I engine test program.

I. SUMMARY

Since the inception of the ASE Program in 1978, 13 Quarterly Technical Progress Reports have been issued under NASA Contract No. DEN3-32, "Automotive Stirling Engine Development Program;" however, reporting was changed to a semiannual format in July, 1981. This report, the fifth Semiannual Technical Progress Report issued under the contract, and covering the period of July 1 to December 31, 1983, includes technical progress only. Although the program has been modified to a proof-of-concept format, the objectives described below still apply to the RESD. The upgraded version of the Mod I engine is not, however, required to demonstrate all these hardware objectives.

Overall Program Objectives

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

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

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

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

Major Task Descriptions

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

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

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

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

The component development tasks, directed at advancing engine technology in terms of durability/reliability, performance, cost, and manufacturability, will include work in the areas of combustion, heat exchangers, materials, seals, engine drivetrain, controls, and auxiliaries.

Task 3 - Technology Familiarization - The existing USAB P-40 Stirling engine will be used as a baseline for familiarization, as a test bed for component/subsystem performance improvement, to evaluate current engine operating conditions and component characteristics, and to define problems associated with vehicle installation.

Three P-40 engines will be built and delivered to the United States' team members; one will be installed in a 1979 AMC Spirit. A fourth P-40 engine will be built and installed in a 1977 Opel sedan for testing in Sweden. The baseline P-40 engines will be tested in dynamometer test cells and in the automobiles. Test facilities will be planned and constructed at MTI to accommodate the engine test program and required technology development.

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

Task 4 - Mod I Engine - A first-generation automotive Stirling engine (the Mod I) will be developed using USAB P-40 and P-75 engine technology as an initial baseline upon which improvements will be made. The prime objective will be to increase power density and overall engine performance.

The Mod I engine will also represent an early experimental version of the RESD, but will be limited by the technology that can be confirmed in the time available. The Mod I need not achieve any specific fuel economy improvements. It will be utilized to verify concepts incorporated in the RESD, and to serve as a stepping stone toward the Mod II engine, thus providing an early indication of the potential to meet the final ASE Program objectives.

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

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

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

Task 5 - Mod II Engine - Postponed.

Task 6 - Prototype Study - Postponed.

Task 7 - Computer Program Development - Analytical tools will be developed that are required to simulate and predict engine performance. This effort will include the development of a computer program specifically tailored to predict

Stirling Engine System steady-state cyclical performance over the complete range of engine operations.

Using data from component, subsystem, and engine system test activities, the program will be continuously improved and verified throughout the course of the program.

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

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

Program Schedule, Status, and Plans

A current schedule of the major milestones for the ASE Program is presented below, and a summary of the accomplishments achieved in the ASE Program during this semiannual report period are presented in the following sections.

Component and Technology Development

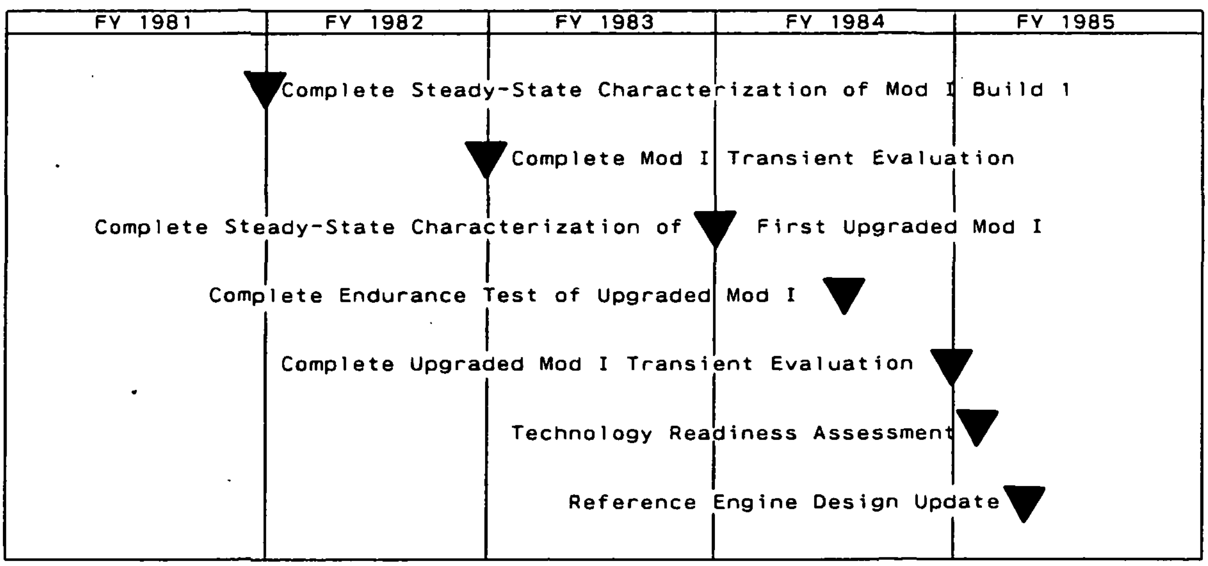
MAIN SEAL DEVELOPMENT

Although manufacturing drawings showed that there was no significant differences in the clearances and, therefore, the motion of the rods in the P-40 and Mod I engines, the P-40 engines were demonstrating seal lives of $\sim$1000 hours, whereas the Mod I was demonstrating a seal life of less than 100 hours and was very erratic, varying from 100 hours to less than 5 hours. An analysis into the design differences of the two engines was conducted, revealing that the crosshead/guide clearances in the Mod I in the hot-operating condition were twice that of the P-40.

The main rod seals are designed to act as seals, not as guides for the piston rod. To minimize any static radial loading, the seals are mounted on separate conical seats, allowing them to move laterally and align with the piston rod during assembly. The seals are effectively clamped in the axial position by the seal loading spring so no movement of the seats will occur during engine operation.

Stirling Proof-of-Concept Program

Milestones



The reaction of the piston rod during engine operation is a function of the crosshead/guide clearance, piston/liner clearance, and the concentricity of the crosshead/guide and cylinder liner. Manufacturing tolerances and assembly procedures must be controlled to minimize the radial dynamic loads on the seal that adversely affect their performance.

Mod I crosshead/guides were made from aluminum, whereas those in the P-40 were steel. Although the clearances are the same at room temperature, the operating clearance in the Mod I increases due to the differential expansion ratio of aluminum to steel. The P-40 crosshead/guide clearance is specified as 20-50 microns, whereas the Mod I clearance is 30-60 microns at a room temperature of 20°C. Since oil temperature increases to 85°C during operation, the P-40 clearance remains constant, whereas the Mod I clearance grows to 80-120 microns.

To investigate this relationship further, P-40 engine No. 4, configured with steel crosshead guides, was assembled with a different clearance in each cycle. The clearances were set at 40, 82, 100, and 120 microns. The engine was tested according to an accelerated duty cycle directed at causing early failure. After 270 test hours, the seal areas were inspected, and direct correlation was found between crosshead/guide clearance and oil migration into the working cycle.

The cycle with the largest crosshead/guide clearance had oil throughout the entire seal housing and in the working gas cycle. Oil was also present on the surfaces of the piston base and piston rings. This condition was observed to progressively improve for the remaining cycles as the crosshead/guide clearance became smaller. In the cycle with the tightest clearance (40 microns), no oil was present in the seal housing cavity; thus, the working gas cycle was perfectly clean.

In order to generate additional information on the effect of crosshead/guide clearance, two additional life tests are currently underway.

Mod I engine No. 7* has been fit with steel crosshead/guides with clearances of 40, 43, 45, and 51 microns, and is being tested according to the accelerated duty cycle. At this point in time, more than 400 hours of operation have been obtained with no indications of seal failure.

P-40 engine No. 9 has been configured with tight and loose clearances (18, 41, 52, and 65 microns), and is being tested according to the accelerated duty cycle. To date, the engine has completed more than 400 hours of testing with no indication of oil leakage.

HIGH-TEMPERATURE ENGINE TESTING

The RESD, and therefore the Mod II, were designed to operate at a heater tube metal temperature of 820°C. The Mod I was designed for a temperature level of 720°C; as a result, Mod I engine durability testing to date has only demonstrated life at the lower temperature. Although the upgraded Mod I was designed for 820°C, only limited tests were accomplished on Mod I engines No. 5 and 6.

In order to evaluate the three alternative castings (iron-based XF-818, CRM-6D, and SAF-11) developed for the ASE program heater heads at the elevated operating temperature of 820°C, the high-temperature P-40 engine (HTP-40) has been used. Eight heater head quadrants composed of different casting/tube materials were fabricated (Figure 1-1). These quadrants were fabricated for the test to provide adequate spares in case of failures. The engine was initially configured with four quadrants, each containing different casting materials so that the data sample would not favor any

*All engine numbers refer to a new engine numbering system described in Section IV.

one casting material. This objective had to be compromised later in the test program as failures were experienced.

The baseline casting material (cobalt-based HS-31) was used in USAB's P-40 engines and in the Mod I engine. The upgraded Mod I and RESD/Mod II were designed to use XF-818 casting material. Although the objective for the HTP-40 test program was to evaluate and compare the alternative casting materials, upon review of Figure 1-1, it is apparent that the quadrants were also fit with alternative tube materials.

The cobalt-based baseline tube material N-155 was used in the P-40 and Mod I engines. Several alternative tube materials are under evaluation, although the upgraded Mod I and RESD/Mod II were designed to use seamless CG-27. In fact, quadrants 1, 2, 5, and 7 were retrofit with N-155 tubes when it was realized, after testing well into the program, that creep-induced rupture of weaker tubes was precluding an accurate determination of the fatigue characteristics of the heater head castings.

The HTP-40 test program was comprised of three phases, all at a heater head metal temperature of 820°C. Phase I consisted of 1000 hours with the working gas mean pressure cycling between 4 and 7 MPa in a cycle time of 90 seconds. The purpose of this phase was to accumulate combustion environmental exposure data. Phase II, consisting of 500 hours with the gas pressure cycling between 4 and 9 MPa in a cycle time of 90 seconds, was intended to initiate fatigue damage in the casting materials, and some creep damage to the tubes. Phase III, designed to induce failure in the castings, had no time limit imposed with the gas pressure cycling between 4 and 15 MPa in a cycle time of 38 seconds. Testing is currently continuing, and a summary of the test results are presented in Table 1-1. The test program is currently in Phase III and, so far, the SAF-11 casting material in quadrant 8 has experienced a failure. The materials selected for the RESD/Mod II design - XF-818 casting and CG-27 tubes - continue to endure the test.

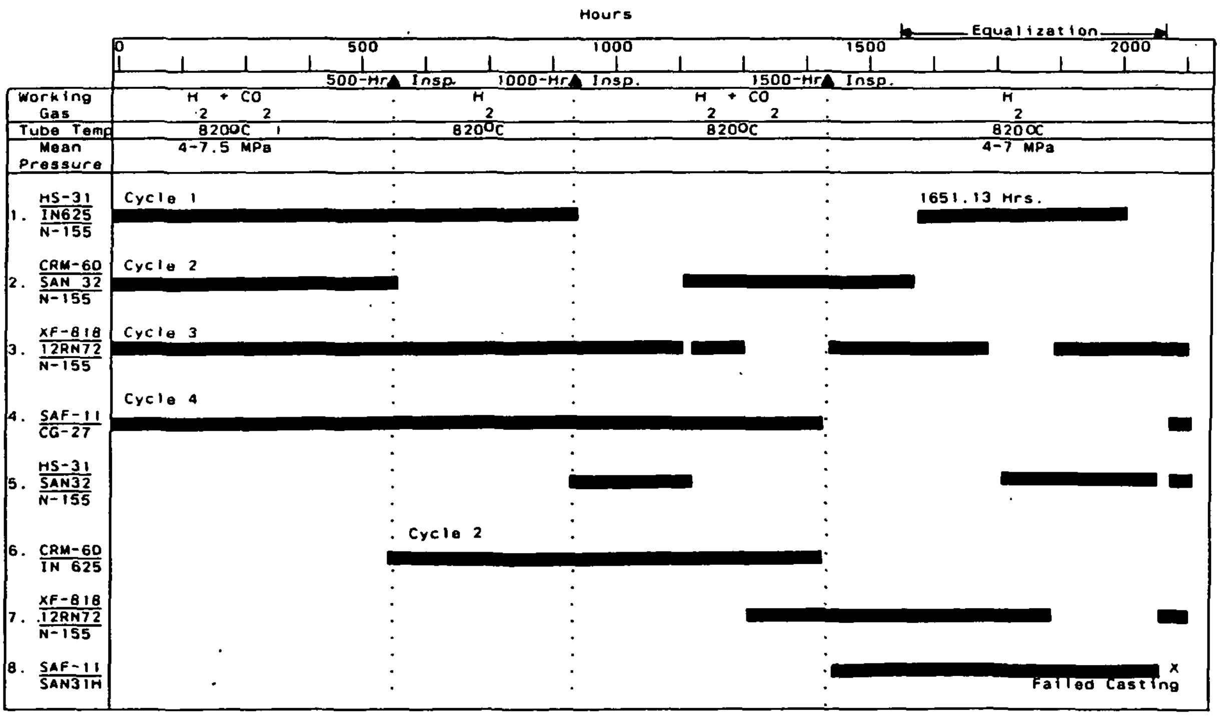


Fig. 1-1 HTP-40 Log Test Hours

Table 1-1
HTP-40 Quadrant Engine Experience Summary

			Time Accumulated During Test Period				
			Phase I	Phase II		Phase III	
Quad-rant	Casting Material	Tube Material	First 1000 Engine Hours (4-7 MPa)	Next 500 Engine Hours (4-9 MPa)	4-9 Equalization (509 Engine Hours)	4-15 Phase	Status
1	HS-31	*Inconel 625* N-155	940.14	--	-- 295	- 37.76	OK OK
2	CRM-6D	Sanicro 32 N-155	568.28	343.48	45.9 64.6	- 4.46	OK
3	XF-818	12RN72 N-155	940.14	241.63	-- 288.48	- 38.48	OK
4	SAF-11	CG-27 3-Tube N-155	940.14	513.38	-- --	- -	OK
5	HS-31	Sanicro 32 N-155	--	169.9	-- 242.2	- 34.02	OK
6	CRM-6D	Inconel 625 --	371.86	513.38	-- --	- -	Needs Repair
7	XF-818	12RN72 N-155	--	267.51	45.9 359.7	- 38.48	OK
8	SAF-11	Sanicro 31H	--	--	405.6	.72	Casting Failed

HEATER HEAD DEVELOPMENT

From experience in developing the Mod I engine, it is known that the internal flow distribution of the working gas through the heater tubes and casting manifolds is crucial to the effectiveness of the heater head as a heat exchanger in transferring energy from the combustion gases to the working gas in the working cycle. If the flow system of manifold volume above the expansion space, tube volume, and manifold volume above the regenerator space is not matched, wide variations in temperature, efficiency, and power can develop. The flow distribution of the RESD/Mod II manifolds were evaluated and compared to the upgraded Mod I. A rig was set up to quantitatively measure the airflow through acrylic models of both heater heads. Pressure drop was also measured. Results from these tests were related back to the design of the RESD/Mod II heater heads, and revisions were made.

A preliminary design of the RESD/Mod II heater head castings was completed and, when flow tested in the model, gave a better flow distribution than that measured on the upgraded Mod I.

Based on these results, a set of casting drawings were completed and procurement was initiated. Since the design calls for investment castings, the process for casting such a complex design requires development. A three-dimensional view of the RESD/Mod II annular heater head configuration is shown in Figure 1-2.

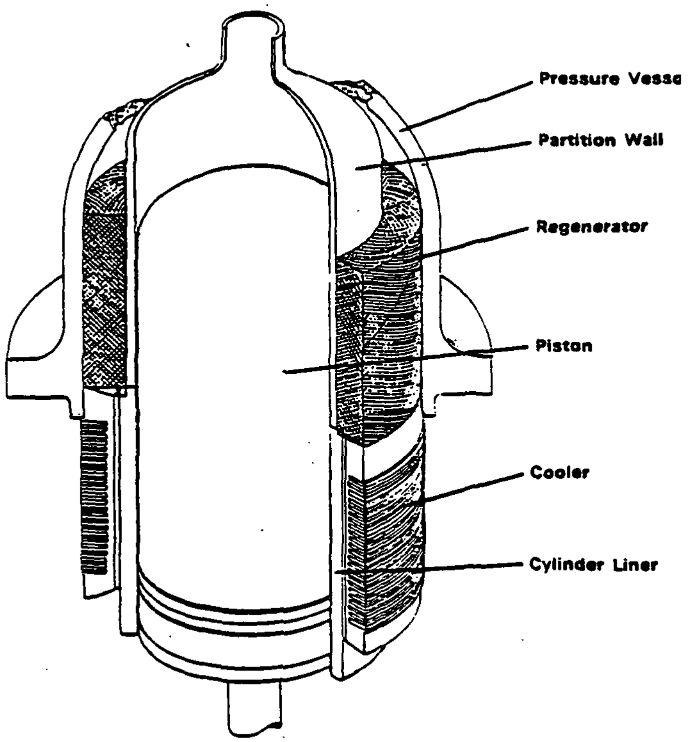


Fig. 1-2 RESD/Mod II Annular Heater Head

FUEL NOZZLE DEVELOPMENT

The evaluation and selection of an alternative fuel nozzle was completed during this report period. As discussed in MTI Report No. 83ASE334SA4, three candidate nozzles required testing: a radial nozzle, a reduced flow Delavan Airo nozzle, and an MTI conical nozzle. A replacement is required for the fuel nozzle currently used in the Mod I engine, since it is prone to coking and plugging. It also requires a large amount of atomizing airflow, which is a performance penalty to the engine.

Spray and combustion tests of the three nozzles have been conducted. Even after design changes were made to the radial nozzle, carbon continued to form on the face of the nozzle, and its combustion characteristics were found unacceptable at high fuel flows.

While the spray characteristics of the Delavan Airo nozzle were very good, it too performed poorly during combustion tests. The conical nozzle, on the other hand, displayed both superior atomization and wider spray angle than the radial design. When the airflow characteristics of the conical nozzle were compared to those of the bill-of-material (BOM) Mod I nozzle, a 50 to 60% reduction in atomizing air pressure and flow was found.

After limited testing in upgraded Mod I engine No. 5, performance of the conical nozzle was similar to that of the BOM nozzle, except no plugging was experienced. A higher temperature differential was measured on the heater tubes however, most probably because the spray angle was too narrow. As a result, design changes are now underway to improve spray quality. The conical nozzle will continue to be tested in an engine environment until fully developed.

Mod I Engine Test Program

Six engines were operational within the ASE program during the latter half of 1983: 1) Mod I Engine No. 1 - installed in the Lerma at MTI; 2) Mod I Engine No. 3 - endurance life testing at USAB; 3) Upgraded Mod I Engine No. 5 - SES* performance testing at MTI; 4) Upgraded Mod I Engine No. 6 - BSE** performance testing at USAB; 5) Mod I Engine No. 7 - seals life testing at USAB; and, 6) Upgraded Mod I Engine No. 8 - ITEP engine scheduled for GM Spirit at MTI. As of December 31, 1983, a total of 4570 test hours have been accumulated in the Mod I engine test program. The profile over time in reaching this level is shown in Figure 1-3, while the total hours accumulated on each engine, active and inactive, are shown in Figure 1-4.

Mod I engine No. 1, installed in the Lerma vehicle, and used extensively in transient testing, was used to conduct initial transient development tests on a Digital Air/Fuel System. The objective was to duplicate emissions levels obtained with the existing K-Jetronic Air/Fuel System. The vehicle was also used in a series of cooling system tests at the Canadian Fram wind tunnel, where it was subjected to different road loads and air speeds over a range of ambient temperatures. The most severe test, intended to cause boil-over, was conducted at 100 km/hr, a 22.1-bhp road load, and 100°F ambient temperature. The maximum top tank temperature reached was 84.6°C.

Mod I engine No. 3 successfully completed the scheduled 1000-hour endurance test described in the last semiannual report. The results of this test provided significant input on the development path to be followed in achieving acceptable life in the External Heat System, Hot Engine System, and Auxiliaries.

*Stirling Engine System (complete engine with auxiliaries and control system).
**Basic Stirling Engine (complete engine less auxiliaries and control system).

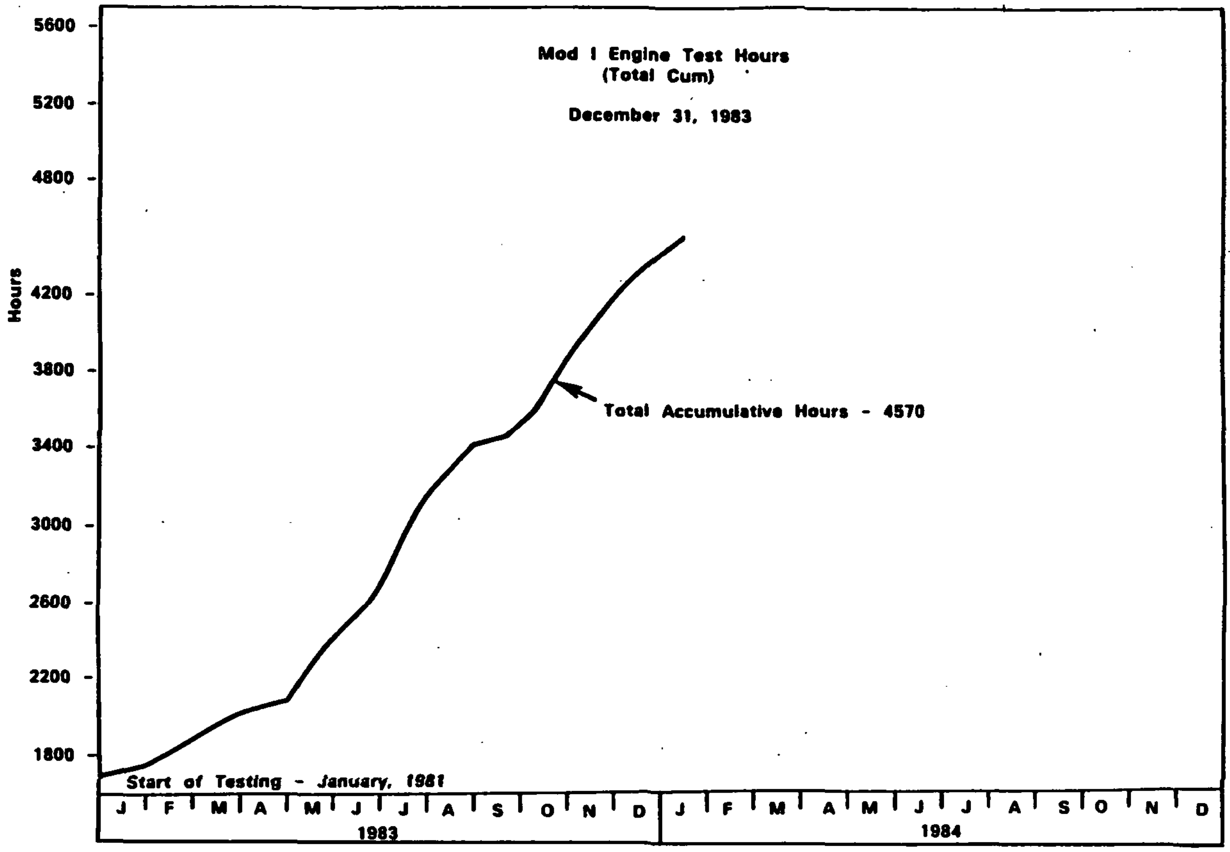


Fig. 1-3 Total Mod I Engine Test Hours as of December 31, 1983

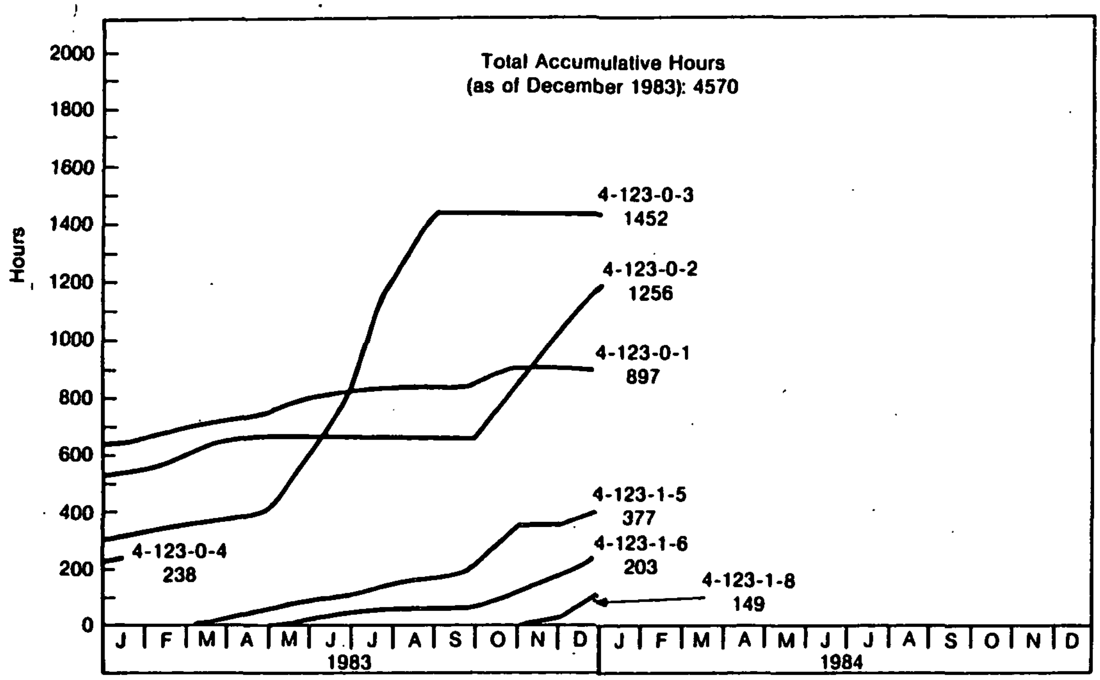


Fig. 1-4 Total Mod I Engine Test Hours by Engine as of December 31, 1983

Specifically, the test revealed preheater clogging after every 200 test hours, fuel nozzle coking and clogging, thermal distortion in certain areas of the heater head, cracks in the crankcase, blower free-wheel life, and H2 compressor connecting rod bearing life. The engine has gone through an extensive teardown and inspection to properly document these findings. It is currently being reassembled to resume endurance testing on the External Heat System to develop MTI's conical fuel nozzle, and to determine the causes for soot formation in the combustion system.

Upgraded Mod I engine No. 5, configured as an SES, has been used exclusively to evaluate the performance of the engine, and characterize the upgraded Mod I engine at 720°C. This engine has successfully met the major program milestone to characterize the upgraded Mod I by September 30, 1983. Later in the report period, the engine was removed from the MTI test cell, so that the first ITEP engine (upgraded Mod I engine No. 8) could be acceptance tested. Simultaneous to this acceptance test, engine No. 5 was installed in the GM Spirit vehicle to check out its systems, and prepare it for engine No. 8 when it left the test cell.

Upgraded Mod I engine No. 6 has been engaged in extensive BSE development/acceptance testing. The engine was used to evaluate the modifications made to the contours of the cylinder housing, performance changes at 720, 770, and 820°C heater head temperatures, hairpin and tubular-to-elliptical CGR combustors, and investigative testing to confirm design estimates for engine losses.

Mod I engine No. 7 was used exclusively for seal life testing, which was conducted to compare Pumping Leningrader (PL) seals made from HABIA and Rulon J materials. A modified seal housing vent system was also evaluated. Unfortunately, testing was interrupted in September when a crack was discovered in the crankcase housing. After retrofit, the engine was rebuilt with steel crosshead guides of reduced operating clearance, and tested according to a prescribed seal life duty cycle.

Upgraded Mod I engine No. 8 has been built for eventual installation in a Spirit vehicle scheduled to be tested by General Motors under the ITEP. The engine was assembled and acceptance tested during this report period. It will be installed in the Spirit during the first half of 1984, and delivered to General Motors in April.

Work Planned for Next Report Period

The following major activities are scheduled for the ASE Program during the first half of 1984:

- upgraded Mod I engine No. 8 will be evaluated in a Spirit vehicle by means of dynamometer, test track, and cooling tunnel tests conducted by GM at their Research Center;
- upgraded Mod I engine No. 9 will be assembled and acceptance tested at MTI in preparation for delivery to John Deere & Co. for test and evaluation in the ITEP;
- initial castings will be made of the RESD/Mod II V-engine block and annular heater head for evaluation of casting quality and dimensional accuracy;
- an independent manufacturing cost study of the RESD, conducted by the Pioneer Engineering & Manufacturing Company, will be issued;
- evaluation and tests of different partition wall materials will be completed on the P-40R engine;
- the final annular heater head design concept will be selected for the RESD/Mod II;
- testing at 820°C on the upgraded Mod I engine will be initiated;

- preliminary tests of ceramic preheater segments supplied by Coors will be completed;
- a single "H"-piston-ring configuration will be tested in a Mod I engine; and,
- proof-of-concept testing of a lobe blower will be completed.

II. REFERENCE ENGINE SYSTEM DESIGN (RESD)

With the completion of the RESD Design Review in May of 1983, two major tasks remained to be accomplished:

1) the completion of a manufacturing cost analysis by an independent contractor; and,

2) the determination of the manufacturability of novel designs in the RESD.

These tasks required that a preliminary design be completed so that manufacturing samples would be obtained as part of the "proof-of-concept" for the RESD.

Manufacturing Cost Analysis

Preliminary estimates of the 1983 RESD manufacturing costs were established by MTI during the first half of 1983*.

In order to accurately determine the costs, an independent contractor with experience in the automotive manufacturing arena, Pioneer Engineering and Manufacturing of Detroit, Michigan, was retained during the last half of 1983.

A detailed costing addressing 100% of the engine parts is currently being performed, with completion scheduled for the early part of 1984.

See MTI Report No. 83ASE334SA4.

III. COMPONENT AND TECHNOLOGY DEVELOPMENT

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

Activity during the last half of 1983 focused on developing technology that was justified in the May 1983 update of the RESD. Emphasis was placed on near-term, proof-of-concept demonstration of technologies that could be applied to a Mod II engine design derived from the updated RESD.

During the first half of 1984, primary emphasis will be placed on preparation for the design phase of the Mod II. Dominant activity will be substantiation of Mod II design concepts in prototype configurations, development of Mod II concept components, and engine and component rig tests to define Mod II design parameters, and quantify Mod II performance/durability improvements.

External Heat System (EHS)

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

Activity during this report period focused on completing the design/test of reduced flow, air-atomized fuel nozzles, rig evaluation of ceramic-coated and all-ceramic liners, and the design and procurement of a generic CGR combustor. The ceramic preheater test sections have been ordered, and the necessary Preheater Rig modifications have been completed.

An analysis of USAB CGR combustor designs and upgraded Mod I EGR performance have also been completed. It became apparent during this report period that effort was needed to improve combustor/preheater durability, soot formation, and fuel nozzle/preheater plugging to reduce unnecessary engine down time.

An EGR durability and performance evaluation program is planned for the first half of 1984. The goal of this program is to achieve reduced soot formation and improve mechanical durability. These design techniques will be applied to future CGR development efforts.

FUEL NOZZLE DEVELOPMENT

Three improved atomizing nozzles (shown in Figure 3-1) were tested during this report period: a) an MTI radial nozzle; b) a reduced flow Delavan Airo nozzle (a commercially available, reduced airflow, internally mixing nozzle); and, c) an MTI conical fuel nozzle.

The fuel spin chamber, atomizing air swirl sleeve of the radial nozzle was redesigned during this report period in an attempt to improve atomization and widen the spray angle.

The conical design was intended to reduce soot formation on the nozzle face (a problem exhibited by the radial nozzle) by elimination of the flat surface exposed to the flame. The helical slots were necessary to provide constant-area atomizing air channels.

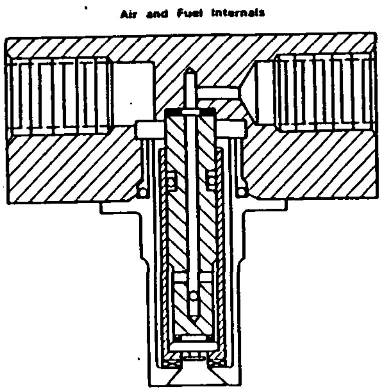


Fig. 3-1a Reduced-Flow External Air-Atomizing Fuel Nozzle

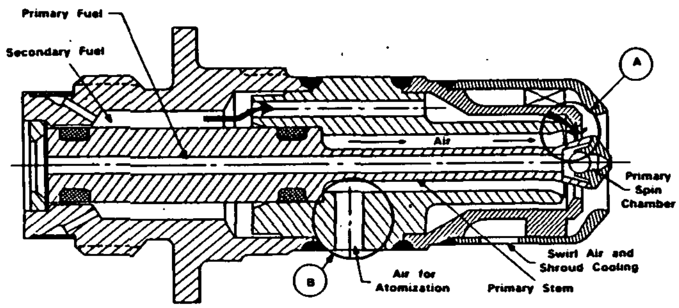


Fig. 3-1b Delavan Air-Blast Nozzle #37112

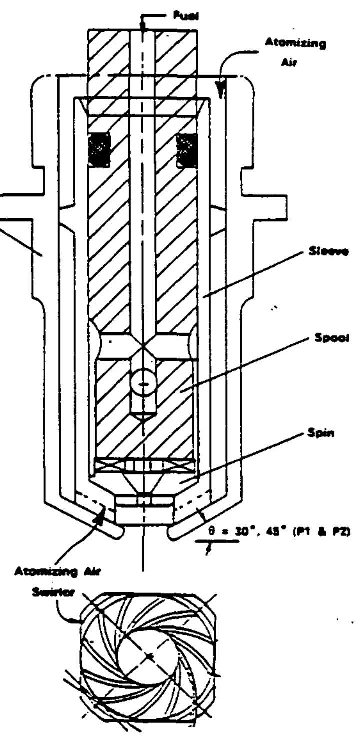


Fig. 3-1c MTI Conical Fuel Nozzle

The design changes made to the radial nozzle resulted in little or no performance improvement. Carbon continued to form on the face of the nozzle near the exit hole, and high fuel flow combustion characteristics were still unacceptable. CO emissions still exceeded the goal of <27 EI for the design lambda (λ), and tube ΔT exceeded the goal of 110° maximum spread.

While the Airo nozzle's spray characteristics were very good, it too performed poorly during combustion tests. A major concern was the inability to control the nozzle, which led to 50°C tube temperature excursions. The internal mixing more than likely resulted in the same instabilities exhibited by the Bill-of-Materials (BOM) Mod I nozzle (USAB), but with a more pronounced effect.

Spray testing of the candidate nozzles eliminated all but one version of the conical nozzle, which displayed both superior atomization and wider spray angle than the radial design. A comparison of the airflow characteristics of this "prototype" nozzle and the BOM Mod I nozzle is shown in Figure 3-2. The net result is a 50-60% reduction in atomizing air pressure and required flow.

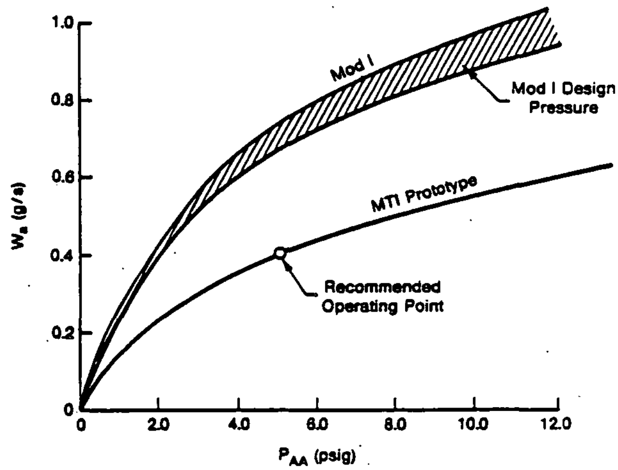


Fig. 3-2 MTI and Mod I Nozzle Airflow Versus Pressure

A total of 10 hours of testing with the conical nozzle were achieved with upgraded Mod I engine No. 5 at a tube temperature set point of 720°C. Figures 3-3 through 3-5 compare the emissions with that obtained with a clean BOM Mod I fuel nozzle. The rear-row temperature differential is compared in Figure 3-6.

Performance of the prototype conical nozzle was similar to that of the BOM Mod I with the very important exceptions that no nozzle plugging was evident, and very little carbon was present on the nozzle face.

The slightly higher CO and ΔT shown in Figures 3-4 and 3-6, respectively, were probably due to the fact that the spray angle was still too narrow; as a result, modifications were made to improve the spray quality.

Development of this upgraded conical nozzle will continue in the first half of 1984.

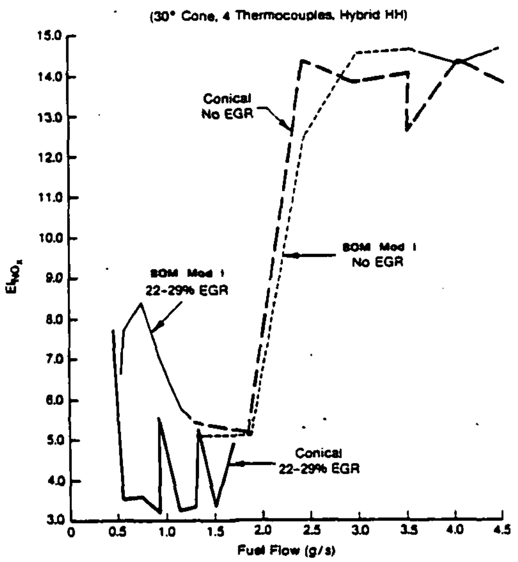


Fig. 3-3 EI_{NOx} for Mod I Engine No. 1 and the Conical Nozzle

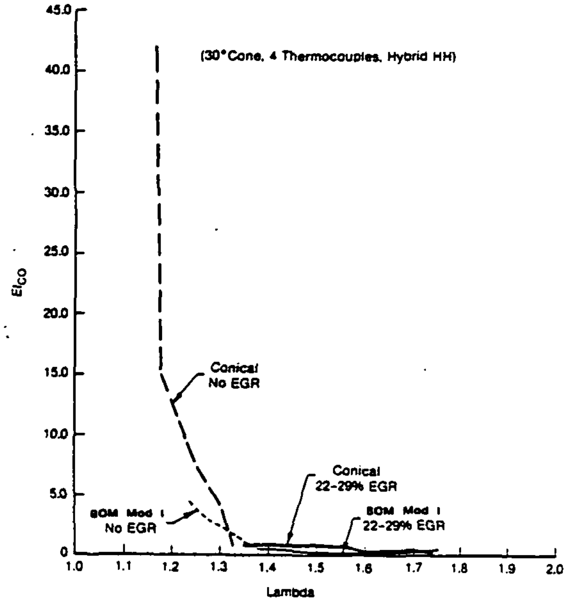


Fig. 3-4 EI_{CO} for Mod I Engine No. 1 and the Conical Nozzle

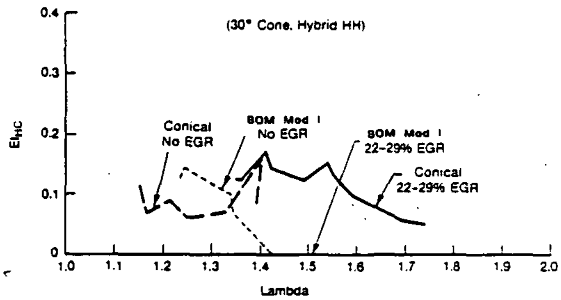


Fig. 3-5 EI_{HC} for Mod I Engine No. 1 and the Conical Nozzle

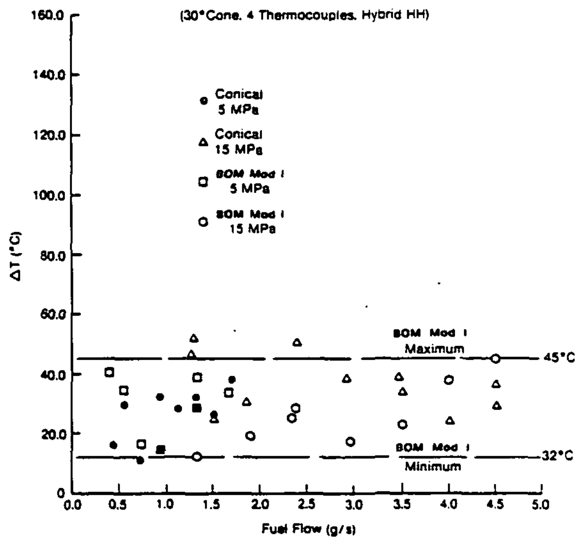


Fig. 3-6 Delta T for Mod I Engine No. 1 and the Conical Nozzle

THERMAL-BARRIER COATING AND CERAMIC EVALUATION

The high-temperature environment and high radiative loadings (particularly for high fuel flows and zero gas recirculation) severely limit the life of the metallic combustor. A ceramic-coated liner and an all-ceramic liner were tested in an effort to improve the longevity of the combustion system.

A developmental combustor* (Figure 3-7) was coated with a three-layer plasma-sprayed zirconia, and instrumented with eight intrinsic thermocouples.

A noncoated combustor was also instrumented, and back-to-back tests were run in the Combustor Performance Rig. A comparison of the average metal temperatures and calculated heat flux to the liner show little difference between coated and uncoated combustors; however, the coating did appear to reduce the oxidation of the liner.

A 200- to 400-hour upgraded Mod I endurance test of a coated combustor is scheduled for 1984.

An EGR combustor (shown in Figure 3-8) with a siliconized silicon-carbide combustion liner was also tested in the Performance Rig for one hour, during which time the liner cracked in several places.

A post-test material analysis indicated that the cracks began during the heat-up cycle, and propagated further during the cool down. Variations in liner thickness and severe thermal gradients were the prime contributors to the failure.

Development of alternate materials and coatings will continue throughout 1984.

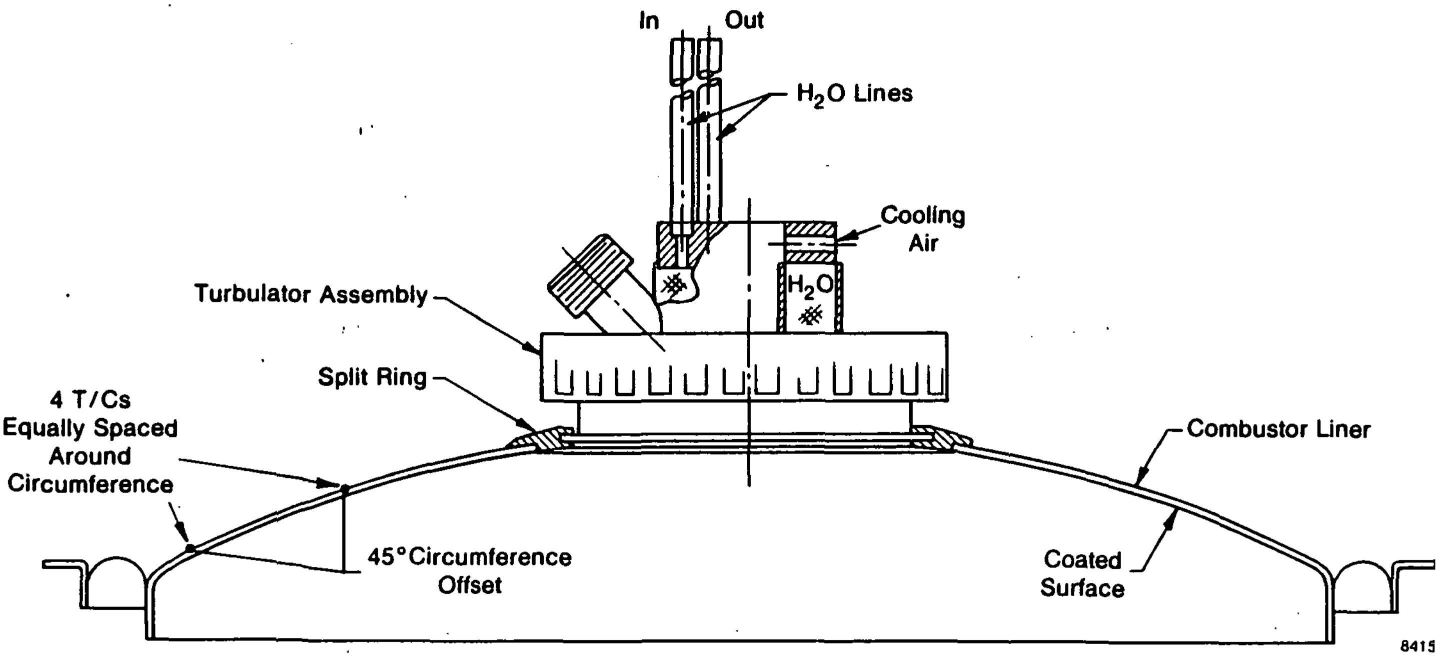


Fig. 3-7 Water-Cooled, Zirconia-Coated Combustor

*Liner originally designed for use with the Delavan air-blast nozzle, and has a water jacket for nozzle cooling. Tests run at .3-gpm H2O with the conical nozzle proved the jacket to be effective in reducing nozzle temperature.

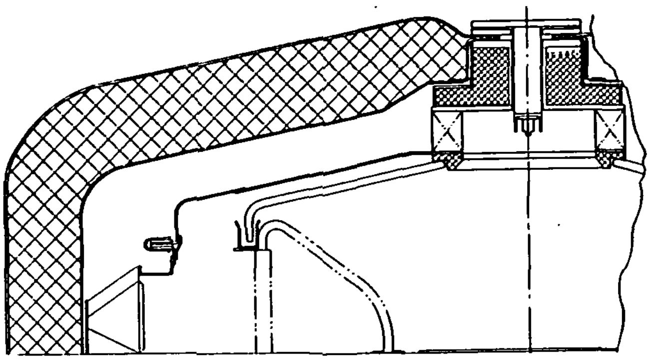


Fig. 3-8 EGR Combustor with Siliconized Silicon-Carbide Combustor Liner

UPGRADED MOD I CGR COMBUSTOR DEVELOPMENT

Tubular-to-elliptical and hairpin CGR combustor designs were tested in upgraded Mod I engine No. 6 during this semiannual report period, and the data was compared to earlier Mod I tubular CGR combustors. A design comparison of the three combustors is shown in Table 3-1, and their emissions/temperature differentials are compared in Figures 3-9 through 3-11.

Table 3-1
CGR Combustors

Mod I Combustor Description	No. of Ejectors	Ejector Size (mm)	Mixing Tube Size (mm)
Tubular	8	13.0	30.0
Tubular	8	12.0	30.0
Upgraded Tubular and Elliptical	12	10.0	27.0
Upgraded Hairpin	36	1x27	6x30

The HC emissions of all combustors were very low. It should be noted that the ΔT of upgraded Mod I engine No. 6 is based on two thermocouple measurements per quadrant. The Mod I data, however, is based on only one thermocouple per quadrant (accounts for higher ΔT of upgraded engine No. 6, since the hydrogen flow distribution disparity contributes to as much as 40°C ΔT per quadrant). The emissions of the tubular designs are very similar above 1.0 g/s fuel flow, with the 12-mm diameter tubular performing somewhat better overall. This is not unexpected since the designs are basically similar - the main difference being in the pressure drop and number of tubes, which determines the uniformity and mixing strength, and is influential at very low flows. This would account for the higher CO exhibited by the 13-mm combustor below 1.0 g/s. The measured ΔP of the 13-mm ejector combustor was the same as that of the straight guide vane BOM Mod I combustor. The elliptical shape seemed to have some stabilizing influence on the combustor at low flows.

The hairpin design performed significantly worse than the tubular. Although cold-flow tests of a segment indicated as much recirculation as the equivalent 13-mm ejector (Figure 3-10), cold-flow tests of the complete combustor showed only ~10-15% CGR. Major contributors to low hairpin CGR levels are the adverse effect of converging flow as it leaves the turbulator, and the path the recirculated flow must take (Figure 3-11).

Improved generic CGR combustor designs have concentrated on enhancing combustor mixing by creating an additional mixing zone downstream of the ejector section. The ejector designs chosen were tubular and radial (shown in Figures 3-12 and 3-13, respectively) with mixing channels similar to that of the hairpin design. As pure ejectors, both designs have been proven effective.

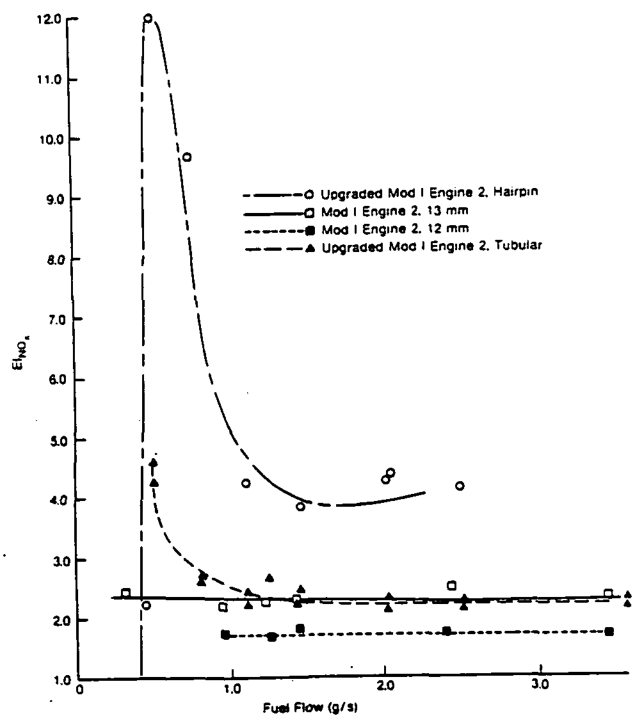


Fig. 3-9 Tubular-to-Elliptical and Hairpin CGR Combustor Designs Emissions/Temperature Differentials - EI_{NOx} Versus Fuel Flow

To determine the effect of the back pressure created by the secondary mixing zones, a Cold-Flow Rig (see Figure 3-14) was constructed, and various ejector/restriction combinations tested. Data (shown in Figures 3-15 and 3-16) demonstrated the feasibility of having an additional combustion mixing zone down stream of the ejector mixing tubes.

Further testing is planned in 1984, along with testing in the new rig (Figure 3-17), which more closely simulates the actual combustor flow pattern.

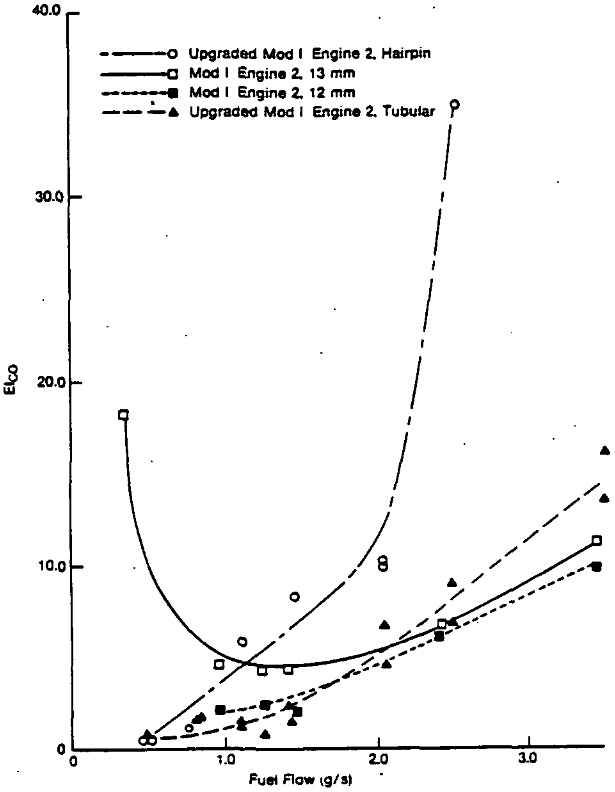


Fig. 3-10 Tubular-to-Elliptical and Hairpin CGR Combustor Designs Emissions/Temperature Differentials - EI_{CO} Versus Fuel Flow

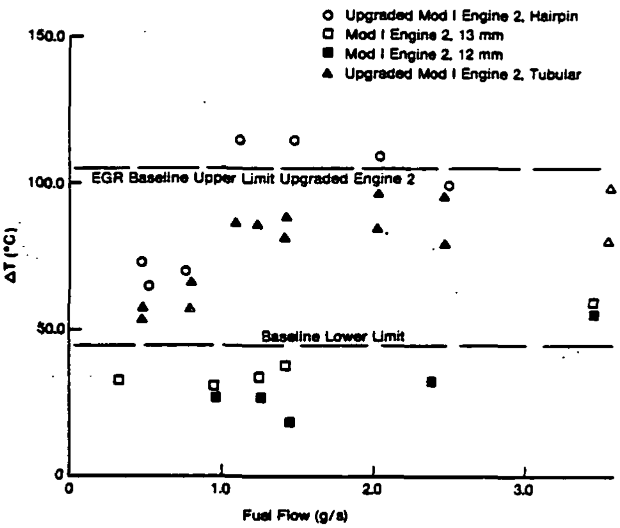


Fig. 3-11 Tubular-to-Elliptical and Hairpin CGR Combustor Designs Emissions/Temperature Differentials - ΔT Versus Fuel Flow

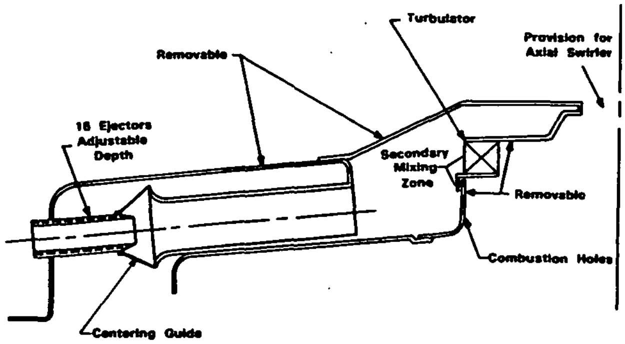


Fig. 3-12 CGR Combustor Concept

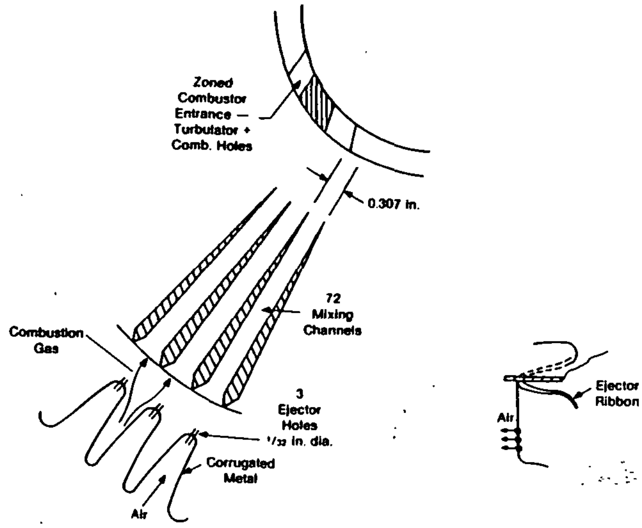


Fig. 3-13 Radial Combustor Concept

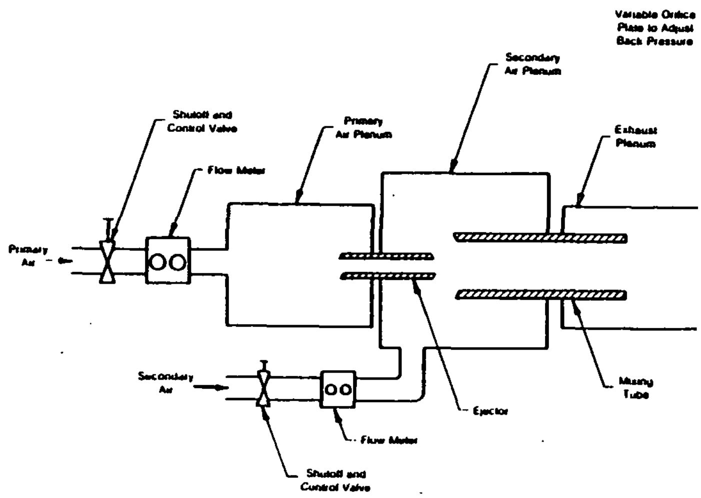


Fig. 3-14 CGR Flow Rig Schematic

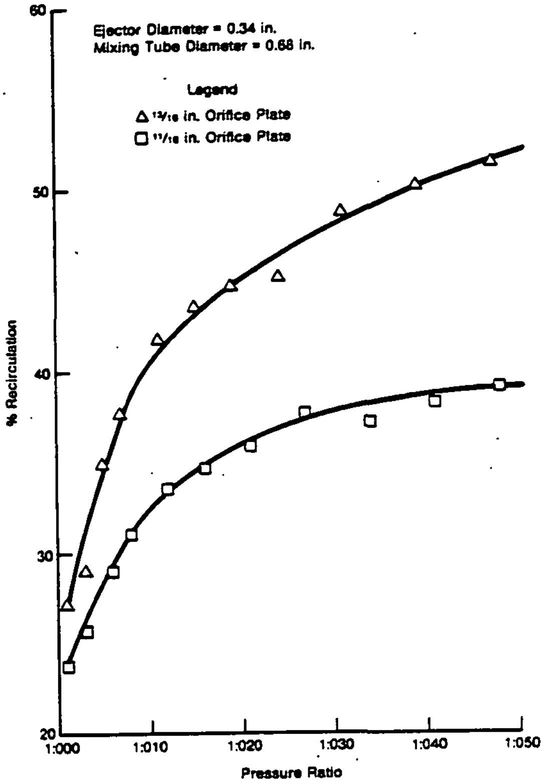


Fig. 3-15 % Recirculation Versus Pressure Ratio

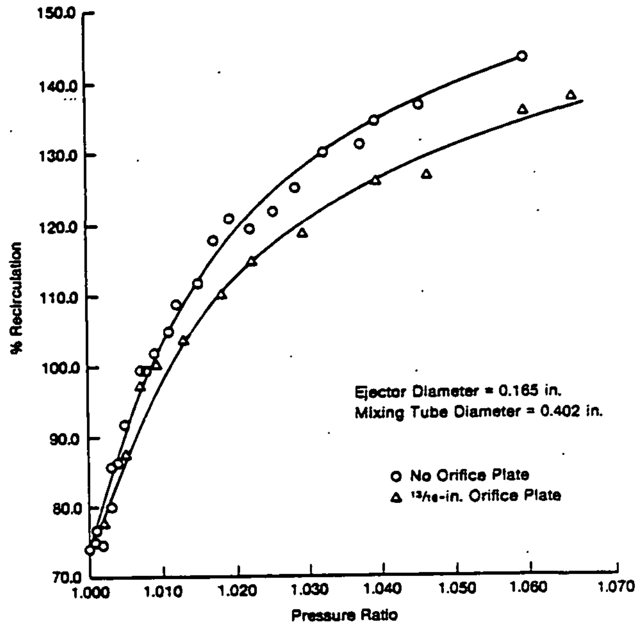


Fig. 3-16 % Recirculation Versus Pressure Ratio

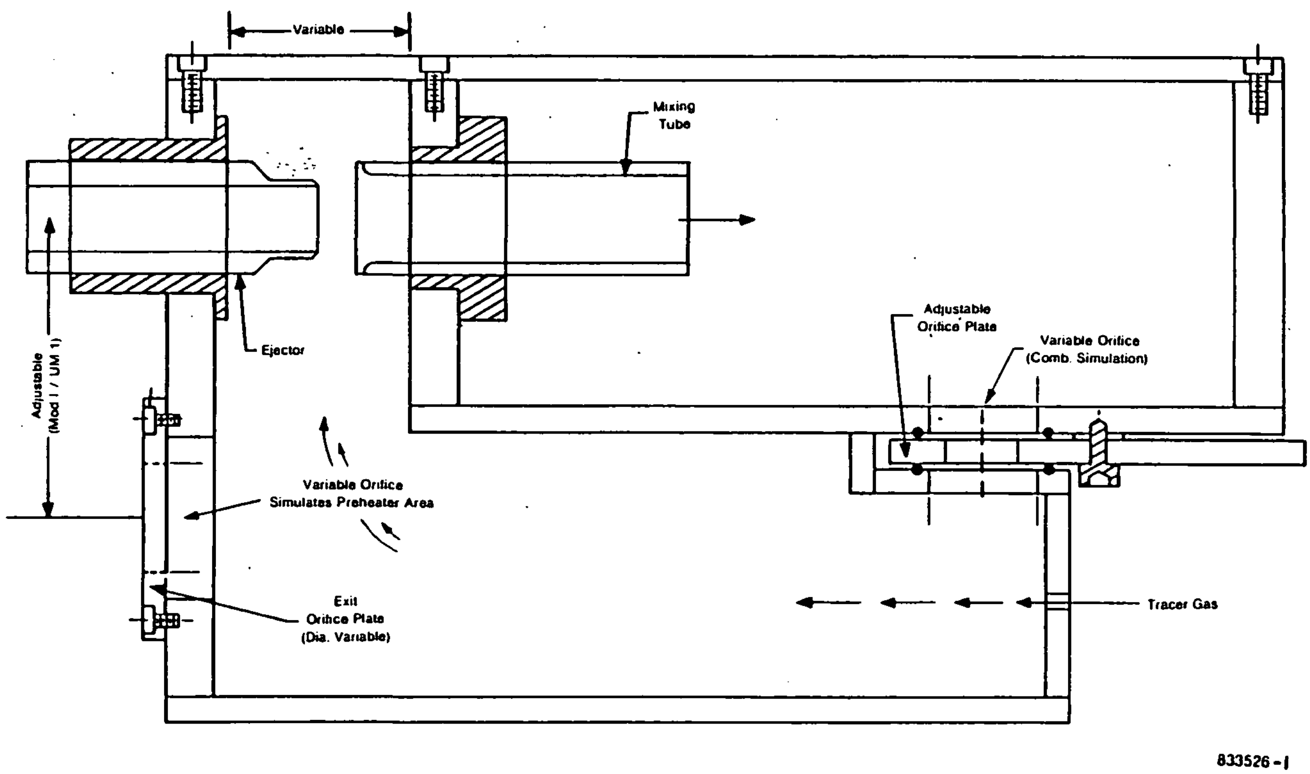


Fig. 3-17 CGR Flow Rig (No. 2)

CERAMIC PREHEATER DEVELOPMENT

In order to reduce the cost of the present metallic preheater, a program was initiated to develop a ceramic preheater. The first phase, which was completed early in 1983, evaluated a ceramic preheater design developed for a gas-turbine application. This evaluation, which took place in the Preheater Rig, resulted in the conclusion that the performance of a ceramic preheater is comparable to a metallic preheater; however, durability needs to be improved.

A manufacuturing study was conducted to determine mass-production costs to evaluate the benefits of further ceramic preheater development. The results showed a significant cost reduction from the metallic preheater (a factor of four). As a result of this work, a ceramic heat exchanger plate was designed for a Stirling engine application (Figure 3-18).

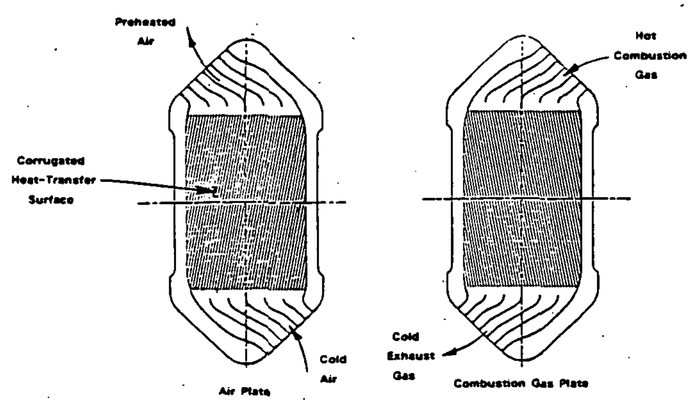


Fig. 3-18 Ceramic Preheater Plates

An order was placed with Coors Porcelain Company for the tooling and four sets of two ceramic preheater test sections. A test fixture was designed and procured to mate the test sections to the Preheater Rig in order to evaluate the performance and durability of the preheater. The rig utilizes the same type of spring-loaded compression seal intended for the engine application, thus enabling development of the sealing mechanism to take place during rig testing of the ceramic test sections.

Testing of the first set of two ceramic test sections (made from Cordierite, the same material used for the gas-turbine test piece evaluated earlier), scheduled for early 1984, will serve as a comparison of the lower stressed Stirling-engine preheater design (no sharp corners) to the gas-turbine preheater design. The second set of test sections, also scheduled for evaluation in early 1984, will be made from a new, more thermally shock-resistant material developed by Coors. The material and schedule for the last two sets of test sections will be determined after the evaluation of the first two sets is complete.

ENGINE/VEHICLE EMISSIONS ANALYSIS

Evaluation of engine/vehicle emissions has continued. Measurements were made in the engine cell on upgraded engine No. 5 at both 720 and 820°C. Figures 3-19 and 3-20 show the CO and NOx emissions for the respective EGR schedules shown in Figure 3-21. EGR schedules A and C correspond to those required to yield predicted CVS NOx emissions of 1 g/mi based on a 12-point CVS cycle simulation.

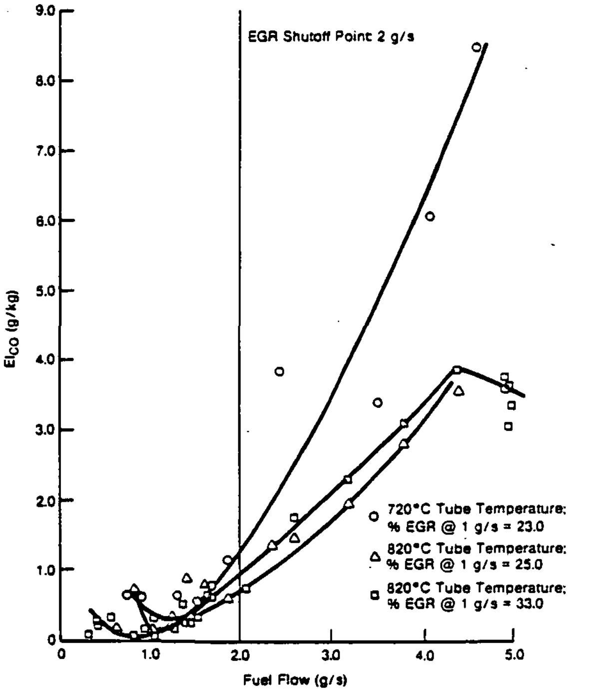


Fig. 3-19 Upgraded Mod I Engine No. 5 Acceptance Test - EI_{CO} Versus Fuel Flow

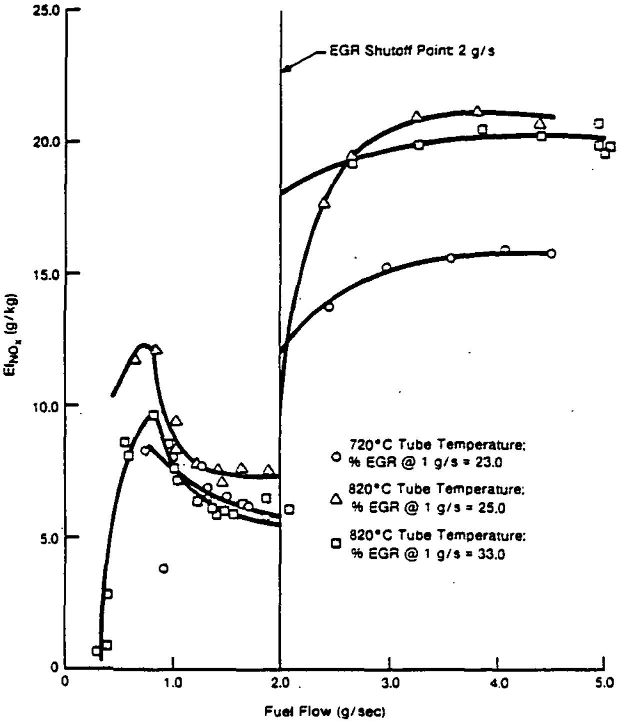


Fig. 3-20 Upgraded Mod I Engine No. 5 Acceptance Test - EI_{NOx} Versus Fuel Flow

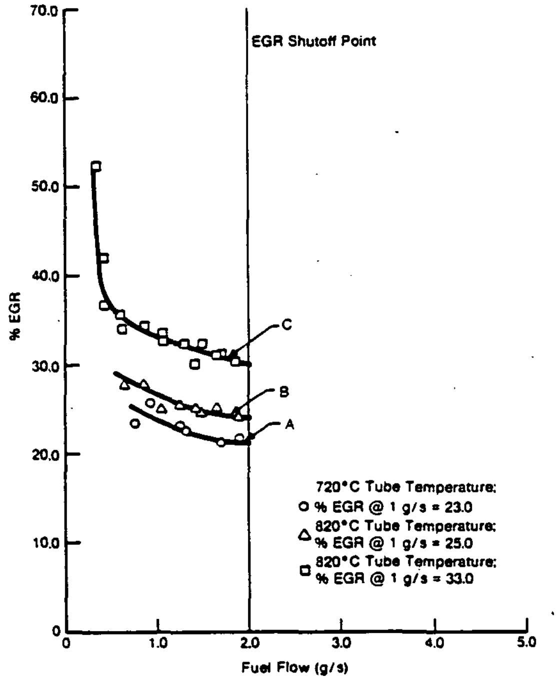


Fig. 3-21 Upgraded Mod I Engine No. 5 Acceptance Test - %EGR Versus Fuel Flow

Estimated CVS emissions are shown in Table 3-2. As expected, operation at higher tube temperatures requires additional EGR to achieve the same NOx levels, while the increased temperature reduces the CO emissions. HC emissions (not shown) were very low. Emissions data was also acquired during acceptance testing of upgraded Mod I engine No. 8.

Table 3-2
Average CVS Cycle Emissions

	720°C 23% EGR*	820°C 25% EGR	820°C 33% EGR
NOx (g/mi)	0.90	1.23	0.87
CO (g/mi)	0.14	0.22	0.04
mpg (g/mi)	22.6	23.3	23.3

*%EGR by volume at 1 g/s fuel flow

The effect of relocating the EGR valve at the EHS exhaust manifold, and closer to the air throttle valve and K-Jetronic, was determined during this report period. The reduced line length, which considerably reduced the valve opening required for a given EGR, appears to be more effective in reducing NOx*; however, the temperature at the inlet of the blower is substantially increased, thus limiting the blower output.

Hot Engine System (HES) Development

The primary goals of this task are to develop HES components that perform with high efficiency, while costing less to manufacture than the currently designed components. One of the key areas for engine performance is the ability of the heater head to transfer a very high percentage of the available combustion gas heat into the working gas. In order to accomplish high heater head efficiency, the flow distribution within the heater tubes must be uniform. Experimental evaluation of this flow distribution was required early in the design process to ensure that the RESD/Mod II heater head

*The amount of water in the EGR stream is thought to be greater since less is lost through condensation upstream of the EGR valve.

castings contained manifolds that gave uniform flow to the heater tubes.

An engine component having significant impact on engine performance is the regenerator. A high-cost wire cloth that delivers high performance is currently in use. In order to reduce the manufacturing cost of the engine, a search for a lower cost alternative material, that has as good or better performance than the current material, is in progress.

Activity during this report period centered on evaluating the flow characteristics of the prototype RESD/Mod II heater head manifolds, and evaluating alternative regenerator matrix materials. Primary objectives during the first half of 1984 will focus on the manufacture of prototype hardware for the RESD/Mod II heater head and partition wall.

REGENERATOR DEVELOPMENT

Tests have been conducted in the Regenerator Single-Blow Test Rig on three alternative regenerator matrix materials: powdered metal, expanded metal, and carbon-carbon composite. Results of these tests are shown in Figure 3-22. The NTU per unit length, expressed as a function of the air mass flow through the test section, represents the heat-transfer characteristics of these materials. In order to relate this parameter to Stirling-engine performance, NTU numbers were calculated for Mod I geometry and engine conditions. The regenerator effectiveness was then calculated from these numbers for the baseline sintered screen, powdered metal, carbon-carbon composite, and Metex knitted wire regenerator materials.

The results of these calculations are graphically depicted in Figure 3-23, which also shows the power and efficiency penalties associated with using knitted wire (3.5-mil wire diameter) regenerators in lieu of the current sintered screen design (2.0-mil diameter).

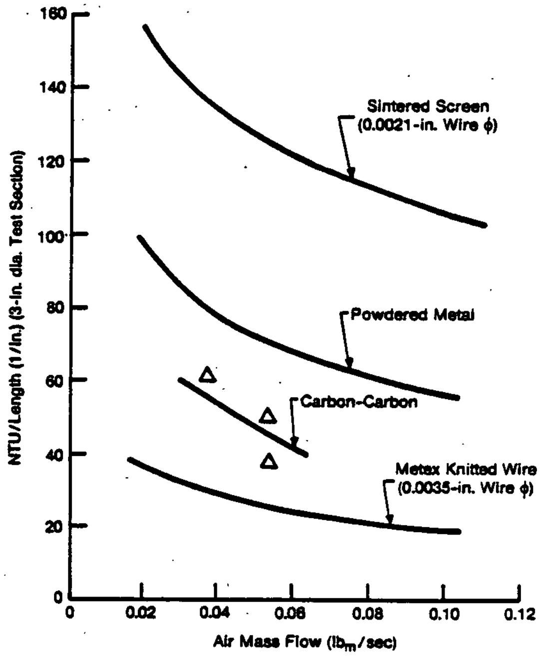


Fig. 3-22a Carbon-Carbon Testing - Heat-Transfer Results from Single-Blow Rig

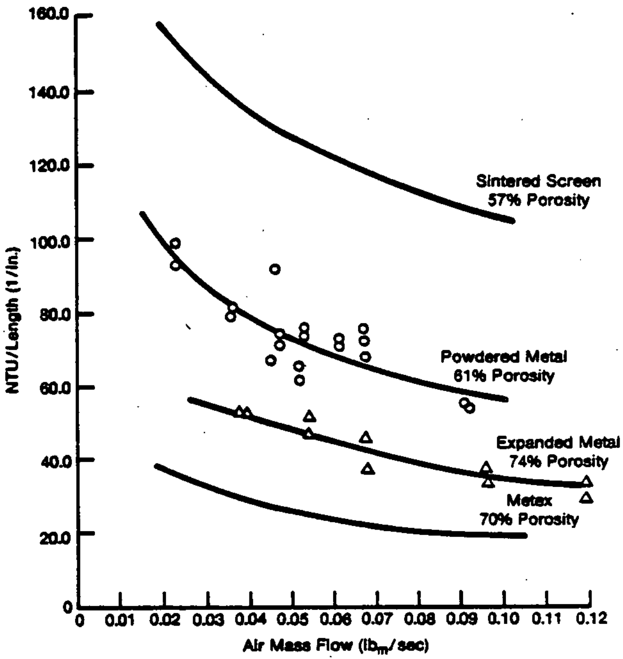
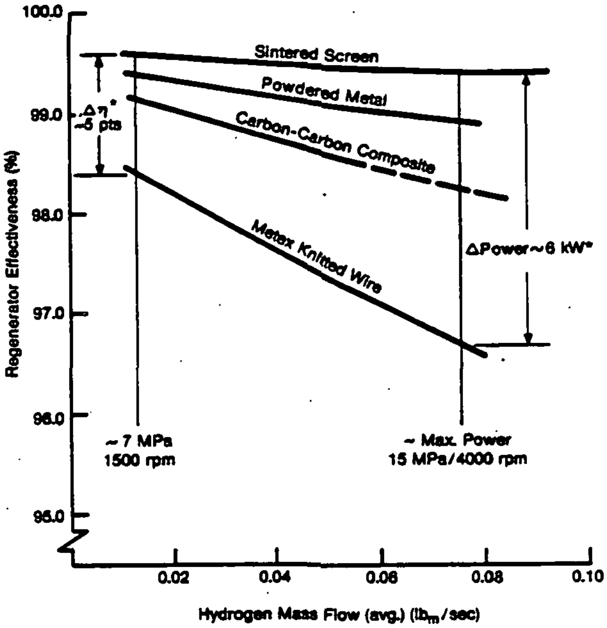


Fig. 3-22b Powdered and Expanded Metal Test Results

Taking into account the offsetting gain of reduced pumping power for the knitted wire regenerator, there is about a 9-10 kW (12-13.4 hp) loss in power. The impact of using a powdered metal regenerator (the best alternative tested to date) is a power reduction of ~1-1/2 to 2 kW (~2.0 to 2.7 hp) at full-power conditions, and a reduction of 1/2 to 1 points in net engine efficiency at the average operating point for a CVS cycle;

however, these performance penalties are offset by a significant cost reduction when using this component.



*From Mod I Engine Testing of Screen and Knitted Wire Regenerators

Note: If reductions in pumping losses are included, impact of lower regenerator effectiveness would be greater (Δ Power would be ~ 9-10 kW).

Fig. 3-23 Alternative Regenerator Materials Performance Impact

Silicon carbide regenerators have been fabricated and sent to NASA/Lewis for testing in their P-40 engine in early 1984. Further evaluations of sintered wire screen materials of differing wire diameters and porosities will take place during the first half of 1984.

HEATER HEAD DEVELOPMENT

Activity in this area during the second half of 1983 centered on the evaluation of the flow distribution of RESD/Mod II and upgraded Mod I heater head manifolds, as well as support of the design and procurement of the prototype RESD/Mod II heater heads.

The flow evaluation was accomplished by using a rig that passed a measured quantity of air through acrylic models of heater head manifolds, and then measured the pressure drop of every other tube. The mass flow through these tubes was calculated and then divided by the total mass flow divided by the number of tubes. This ratio, which relates the actual mass flow to the ideal mass flow (perfectly distributed flow) is graphically depicted in Figure 3-24 for the upgraded Mod I and RESD/Mod II manifolds.

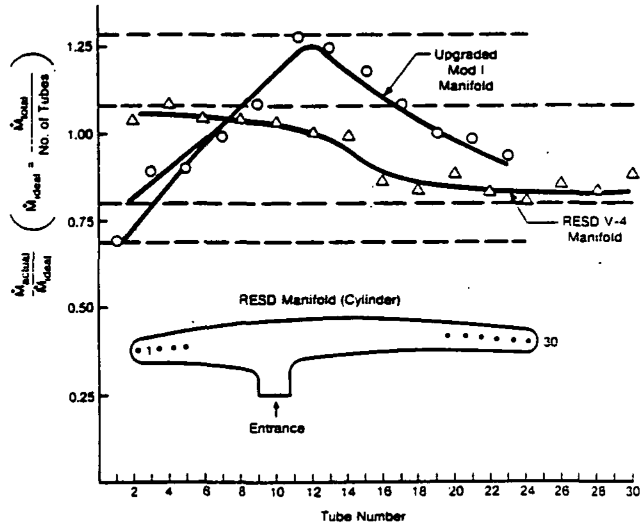


Fig. 3-24 Manifold Flow Distribution for the RESD and Upgraded Mod I Heater Heads

A conclusion can thus be drawn that although not ideal, the flow distribution of the RESD/Mod II is superior to that of the current upgraded Mod I. The prototype RESD/Mod II heater head design was completed incorporating some of the changes deemed necessary from earlier flow tests. Procurement of the heater head castings has been initiated.

Further flow testing of improvements to the RESD/Mod II manifold will take place in early 1984, along with procurement of the prototype RESD/Mod II castings.

Materials and Process Development

The focus of this task is the utilization of materials that will survive 3500 hours of automotive duty cycles with the least cost and with minimal strategic element content. Efforts have been concentrated primarily on the hot section of the engine, i.e., the combustor and heater head. Efforts during the latter part of 1983 included:

- design support testing;
- an experimental determination of XF-818's thermal conductivity; and,
- selecting CG-27 heater tubing for the RESD/Mod II, and evaluating hydrogen permeability in Stirling-engine heater tube alloys.

Plans for the first half of 1984 include:

- determining the tensile/fatigue strength of welded XF-818;
- selecting a new braze material and process for 820°C heater heads;
- initial design support testing for RESD/Mod II; and,
- creep rupture testing of CG-27 to determine upper temperature limit for 3500-hour life.

DESIGN SUPPORT TESTING

Two upgraded Mod I cylinder housings manufactured from alloy XF-818 were subjected to a hydraulic fatigue test during this report period to verify their material/design acceptability. Testing was conducted using an MTS servo-actuated hydraulic fatigue testing machine and a sine wave test cycle that generated a cyclic stress of ±5 MPa. Both housings were tested at ambient temperature for more than 10^7 cycles.

One cylinder experienced 14,906,900 cycles, while the other experienced 12,120,100 cycles, each without noticeable failure. Subsequent sectioning of the castings and liquid-dye-penetrant testing verified the acceptability of both the cylinder housing design and the selection of XF-818.

The long-term creep rupture testing of five alternate heater tube alloys continued during the latter half of 1983. The alloys tested and their compositions are shown in Table 3-3. Testing at 750°C and 850°C is nearly complete. Results show conclusively that alloy CG-27 is the strongest of the five in creep rupture, as reported in the last semiannual report. Using CG-27 will provide the big heat safety margin. Creep rupture testing of CG-27 at higher temperature to determine the upper bound temperature for 3500 creep rupture life of 28 MPa will begin during the first half of 1984.

THERMAL CONDUCTIVITY OF XF-818

The thermal conductivity of alloy XF-818 has been measured from ambient temperature to in excess of 1000°C (see Figure 3-25) by Properties Research Laboratory. Laser flash diffusivity and differential scanning calorimetry techniques were used to obtain both the thermal diffusivity and the specific heat data required for the determination of the thermal conductivity of alloy XF-818.

HIGH-TEMPERATURE ENGINE TESTING

High-temperature engine (HTP-40) testing of alternate heater head casting/heater tube materials continued. The purpose of this test was to rank the heater head candidate alloys in an engine environment. The tested combinations of casting/heater tube materials are shown in Table 3-4. Quadrants 1, 2, 5, and 7 were refitted with N-155 heater tubes when it was realized that creep-induced rupture of weaker tubes precluded an accurate determination of the heater head castings' fatigue characteristics.

In order to offset the effects of excessive heater tube creep, and allow for a valid analysis of casting material failure characteristics, an accelerated test cycle was implemented to maximize fatigue damage. The testing comprised three phases. Phase 1, the purpose of which was to accumulate environmental exposure, consisted of 1000 hours at 820°C with the internal mean pressure cycling between 4 and 7 every 90 seconds. Phase 2, designed to initiate fatigue damage to the casting, and some creep damage to the heater tubes, consisted of 500 hours at 820°C with the internal pressure cycling between 4 and 9 MPa every 90 seconds. Phase 3, designed to induce failure in the castings, took place at 820°C with the mean pressure cycling between 4 and 15 MPa. In order to maximize fatigue damage to the casting while concurrently minimizing creep damage to the heater tubes, the time of the test cycle was reduced to 38 seconds. Diagrams of the mean pressure variations and cycle times for the three test phases are shown in Figure 3-26, and a summary of the test results accumulated are presented in Table 3-4.

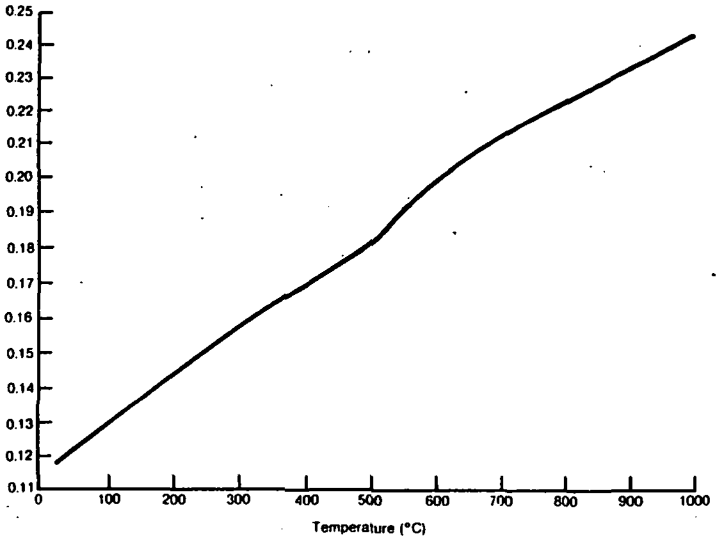


Fig. 3-25 Thermal Conductivity of Alloy XF-818

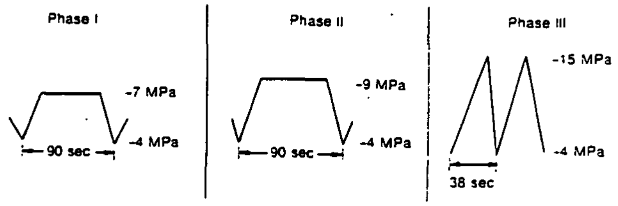


Fig. 3-26 Testing Cycle for Upgraded Mod I Castings

Table 3-3
Alternative Heater Tube Materials - Nominal Chemistry

Alloy	Co	Cr	Ni	Mo	W	C	Al	Ti	B	Cb	Mn	Fe	Si	N
Multimet[1] (N-155)	19.75	21.25	20	3.00	2.5	0.12	-	-	-	1.0	1.50	29.70	1.0	0.15
Alternatives:														
CG-27[2]	None	13.00	38	5.75	-	0.05	1.6	2.5	0.010	0.7	-	38.00	-	-
Inconel 625[3]	None	21.50	61	9.00	-	0.05	0.2	0.2	-	-	0.25	2.50	0.2	-
Sanicro 32[4]	None	21.00	31	-	3.0	0.09	0.4	0.4	-	-	0.60	42.80	0.6	-
Sanicro 31H[4]	None	21.00	31	-	-	0.07	0.3	0.3	-	-	0.60	46.13	0.6	-
12RN72[4]	None	19.00	25	1.40	-	0.10	-	0.5	0.006	-	1.80	51.80	0.4	-

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

TABLE 3-4
HTP-40 Quadrant Engine Experience Summary

Quadrant	Casting Material	Tube Material	Time Accumulated During Test Period: First 1000 Engine Hours (4-7 MPa)	Next 500 Engine Hours (4-9 MPa)	4-9 Equalization (509 Engine Hours)	4-15 Phase	Status
1	HS-31	Inconel 625/N-155	940.14	--	--/295	--/37.76	OK/OK
2	CRM-6D	Sanicro 32/N-155	568.28	343.48	45.9/64.6	--/4.46	OK
3	XF-818	12RN72/N-155	940.14	241.63	--/288.48	--/38.48	OK
4	SAF-11	CG-27 3-Tube N-155	940.14	513.38	-- --	- -	OK
5	HS-31	Sanicro 32/N-155	--	169.9	--/242.2	--/34.02	OK
6	CRM-6D	Inconel 625/--	371.86	513.38	--/--	--/--	Needs Repair
7	XF-818	12RN72/N-155	--	267.51	45.9/359.7	--/38.48	OK
8	SAF-11	Sanicro 31H	--	--	405.6	.72	Casting Failed

With testing ongoing, two significant conclusions have been reached:

1. alloy CG-27 has not experienced any failures during testing, and has thus been selected as the RESD/Mod II heater head to be machined; and,

2. quadrant No. 8, with SAF-11 castings, catastrophically failed after very short exposure to the Phase 3 test cycle (casting split into two pieces, indicating a very low fracture-toughness level); it has thus been removed from the program for further consideration as an RESD/Mod II heater head casting material.

HYDROGEN PERMEABILITY

A study was conducted that reviewed the findings of recent research aimed at reducing hydrogen permeability through Stirling-engine heater tubes. The study concluded that for CG-27 heater tubes and an engine operating condition of 820°C with peak pressures of 20 MPa, a six-month recharge interval is attainable with preoxidized CG-27 tubes and commercial-grade hydrogen.

Primary leakage concern is with the static seal and block porosity (for the RESD Nodular Cast-Iron Block). Efforts have been initiated at USAB to test static seal leakage on a test rig to provide an emperical test to evaluate RESD static seal design leakage rates. Efforts are also underway to define hydrogen permeability in cast iron.

Cold Engine System (CES) Development

The primary objective of CES activity is to develop reliable, effective, long-life rod seals and piston rings. Development activity during the second half of 1983 concentrated on the evaluation of Pumping Leningrader (PL) seals in engines and in the Exploratory Rig, evaluation/development of alternative piston rod seals in the Exploratory Rig, and testing of single, pressure-balanced piston rings in the motored engine.

The primary objective for the first half of 1984 is to develop the single, pressure-balanced piston ring system using motored and hot engines. Alternate piston rod seals will also be evaluated in the Exploratory Rig.

MAIN SEALS

Main seal testing in the Exploratory Rig continued during the second half of 1983. The Exploratory Rig has facilities for measuring dynamic friction and gas leakage past individual seals, and is used to screen alternative main seal designs through short-term testing for comparison of their performance with the BOM PL seals. Potential candidates must then be thoroughly evaluated by engine testing. Table 3-5 is a summary of the testing conducted in the Exploratory Rig.

Table 3-5
Exploratory Seals Rig Test Summary

Seal Set[1]	Seal Type	Seal Body Material	Preload (lbs)	Test Hrs.	Reduced Life Cycle
35	PL	Rulon J	80	510	Orig.
36	27c[2]	Vespel SP21	80	1	"
37	27a	Vespel SP21	80	1	"
38	27b	Rulon J	80	8	"
39	27d	Vespel SP21	100	10	"
40	27a	Vespel SP21	80	1	"
41	27e	Torlon 4203	80	1	"
42	27e	Torlon 4203	80	2	"
43	PL	HABIA PTFE[3]	80	577	#1
44	PL	HABIA PTFE	80	604	#2

[1]Two seals per set.
[2]Refers to Figures 3-27a through 3-27e.
[3]polytetrafluoroethylene

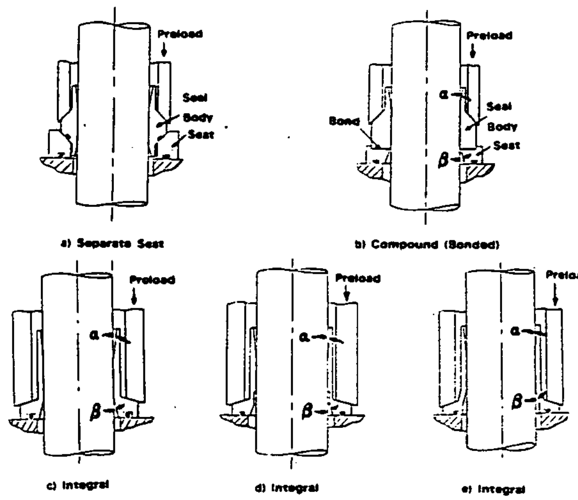


Fig. 3-27 Double-Angle Seals

Testing of Rulon J PL seals (Seal Set 35) was performed in parallel with engine testing of similar seals. Rig testing was performed on a 24-hour duty cycle (Figure 3-28) that had previously been used to evaluate the BOM HABIA PL seals. Overall performance of the Rulon J seals was very similar to that of the HABIA seals.

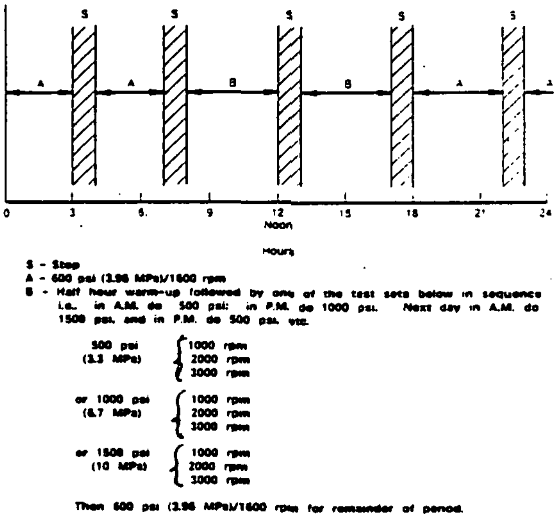


Fig. 3-28 Original Duty Cycle

Over the nominal 500-hour test, hydrogen leakage was consistently low (Figure 3-29), and no oil leakage was detected (a post-test inspection confirmed this).

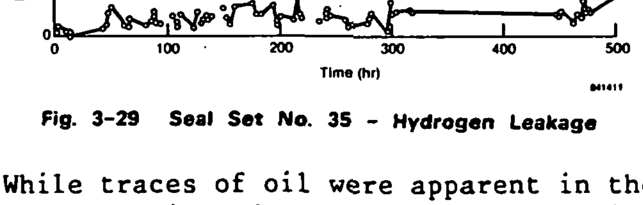


Fig. 3-29 Seal Set No. 35 - Hydrogen Leakage

While traces of oil were apparent in the seal seat interfaces, as was a light film of oil on some sections of the loading sleeves, no liquid oil was present in the seal cavity. The seals had not extruded through the rod/seat clearances, nor deformed under the action of the loading springs. Both seals gained weight during the test due to oil absorption, suggesting that wear, if any, must have been very small.

The next phase of rig testing concentrated on double-angle seal designs (Figure 3-27a to e) based on the principles described in MTI Report No. 83ASE334SA4. Seals C, D, and E differ only in the configuration of the entrance area of the low-pressure region of the seal entrance.

Seal Set 36 was a double-angle, integral seal design with the seal and seat machined as a single unit (Figure 3-27c). The seal, manufactured from Vespel SP21, has an O-ring in the base that acts as a secondary gas/oil seal. A significant advantage of the integral seal arrangement is that it eliminates the potential leak path between the seal body and seat in the current PL seal design. The seal material must also serve as a structural member; as a result, Vespel SP21 was chosen. One disadvantage of this approach is that the more rigid seal material does not readily conform to the rod surface, and the bore of the seal must be manufactured with minimal out-of-roundness in order to effect an efficient gas seal.

To maintain a static gas seal, the seal must be an interface fit on the rod; however, with the higher modulus material, this must be less than that which can be tolerated with a filled PTFE material.

Seal Set 36 was first tested with a spring preload of 80 lbs. Oil was detected in the seal cavity after ~1 hour of operation at 600 psi/1000 rpm. To ensure that the oil was not leaking past the O-rings, the test was repeated first with 100-lbs. preload, and then with 150-lbs. preload. These changes did not make a significant difference; oil leakage was detected after ~3 hours.

In the design of Seal Set 36, the lower oil entry section was formed in the larger diameter base section of the seal, which was clamped by the loading sleeve. The combination of material and clamping would result in a high radial stiffness in the entry section, possibly inhibiting the desired double-pumping action, and leading to poor seal performance. To investigate this, the seal design was modified by adding an undercut to extend the lower inlet section into the thin- wall section of the seal (Figure 3-27d). This design, evaluated in Seal Set 39, showed little improvement in oil leakage (detected after ∽10 hours of operation).

An alternative design of the integral seal (shown in Figure 3-27e) has a continuously converging lower entry section that extends into the thin wall section of the seal. This design, evaluated in Seal Sets 41 and 42, gave very poor performance, with oil leakage detected after one and two hours, respectively.

Figure 3-27a is a double-angle seal design in which the seal and seat are separate components. Previous tests of this configuration using seals made from filled PTFE demonstrated the viability and effectiveness of the double-angle principle; however, the seals failed after a time by being extruded under the gas pressure, deforming the lower inlet section and producing oil leakage. Seals of this design were retested in Seal Sets 37 and 40 with the seals made from Vespel SP21 to overcome the extrusion problem. A small O-ring was also added between the seal and seat. After as little as one hour of operation, both sets of seals failed due to oil leakage.

Figure 3-27b is a compound double-angle seal design in which the seal body is bonded to a rigid seal, and then machined as a single unit. The objective of this design was to allow the use of a filled PTFE material for the seal body, and to overcome the extrusion problem by maintaining a minimum clearance between the seal and rod. The lower inlet section is machined in the seat, but extends slightly into the seal body to ensure no contact between the seal and rod. Using Rulon J material for the seal body, this concept was investigated in Seal Set 38. After 8 hours of operation, oil was detected in the seal cavity.

One of the major objectives of the Exploratory Seals Test Rig was to provide a means of screening and comparing alternative seal designs through short-duration tests. With the original duty cycle shown in Figure 3-28, baseline HABIA PL seals completed more than 500 hours of testing with acceptable hydrogen leakage and essentially zero oil leakage.

For screening purposes, it is highly desirable for realistic failures of HABIA PL seals to be generated in a much shorter time. Reduced Life Duty Cycle #1 (Figure 3-30) was adopted in an attempt to achieve this. The start/stop sequence is the same as the original duty cycle shown in Figure 3-28, but the seals operated mainly at 1500 psi/3000 rpm instead of the more modest 600 psi/1000 rpm used in the original duty cycle.

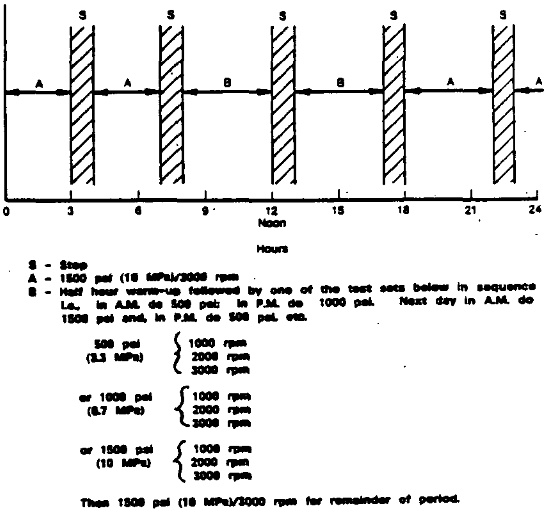


Fig. 3-30 Reduced Life Duty Cycle No. 1

A pair of HABIA PL seals were tested in Seal Set 43 using Reduced Life Duty Cycle #1. During testing, which was discontinued after 577 hours, the seals maintained a low level of gas leakage, and no oil leakage was detected. Overall, there was no significant difference in the performance of the seals under Reduced Life Duty Cycle #1 when compared to the original duty cycle.

Rig testing under mainly high pressure and speed did not have the desired effect. It can be argued that low-speed operation may be more demanding on a seal than high-speed operation, so Seal Set 44 was tested using Reduced Life Duty Cycle #2, which is basically the same as the original cycle shown in Figure 3-30, but with periods of operation at 1500 psi/1200 rpm instead of 1500 psi/3000 rpm.

A pair of HABIA PL seals completed 604 hours under Reduced Life Duty Cycle #2 before testing was discontinued. During this time gas leakage was low, and there was virtually no oil leakage.

In a continuing effort to generate PL seal failures in a reasonably short time, testing with Reduced Life Duty Cycle #3 is now underway. This cycle incorporates numerous starts and stops.

A pumping-ring-type rod seal is under evaluation at USAB. Seals have exceeded 1000 hours without failure in a single-cylinder rig, and seals of the same design have now been installed in a P-40 engine.

To provide a basis of comparison, all engine testing of main seals during the latter half of 1983 was performed using the accelerated duty cycle shown in Figure 3-31.

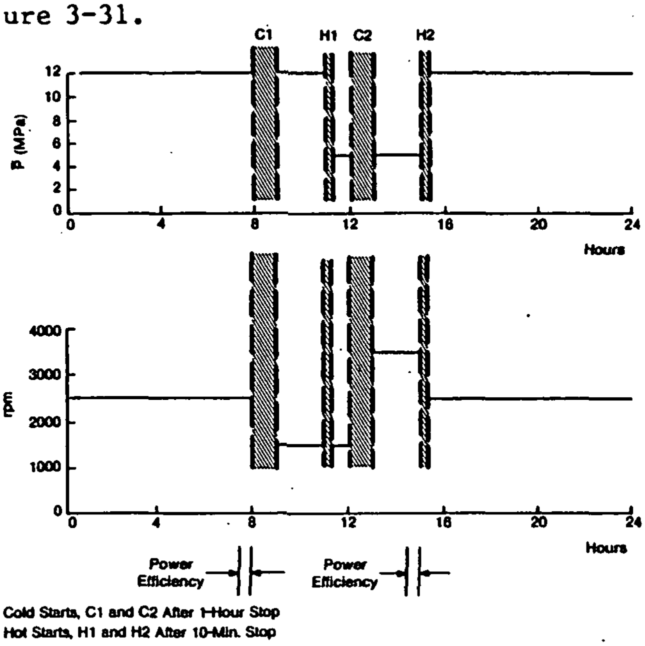


Fig. 3-31 Accelerated Duty Cycle

As discussed in MTI Report No. 83ASE308SA3, three sets of HABIA material PL seals completed 500 hours of accelerated duty cycle testing in P-40 engines without a single failure or degradation in engine performance.

When the testing was completed, it was found that all the seals had formed a feather edge where the seal material had been extruded into the clearance between the rod and conical seal seat by the gas pressure loading. Experience has shown that this type of deformation can eventually lead to failure through oil leakage.

While it may not be possible to eliminate extrusions with the type of material required for the PL seal design, the use of material with higher creep resistance would extend the life of the seal. Rulon J, a commercial filled PTFE material, was considered to be a potential candidate. Rulon J PL seals were tested in P-40 engines using the accelerated duty cycle. During the 500 hours of testing, all the seals but one completed the tests without failure. The single seal failure was due to oil leakage. There was some indication that the extrusion of the Rulon J was less than that which occurred with the HABIA material.

On the basis of the 500-hour tests, Rulon J appears to be an acceptable alternative to the HABIA material; however, testing of seals to failure will be necessary in order to determine whether Rulon J is a superior material. A post-test inspection revealed that the crosshead for the piston rod where the single Rulon J seal failed had excessive clearance (75 μm compared to 20-50 μm for the BOM), probably the reason for the premature failure of the seal.

The main objective of the P-40 seal testing was to investigate factors that might explain the short seal life being experienced in the Mod I engines. The tests proved that the seals could be confidently expected to exceed 500 hours without failure and, within reason, that changes in seal material or variations in their properties would not have a significant effect on seal performance. This indicated that the short life of the PL seals in Mod I engines was due to the detailed seal design and/or adverse operating conditions imposed on the seals.

A comparison of the P-40 and Mod I seal designs showed they were essentially the same except for the dimensional changes required to accommodate the slightly larger rod diameter in the Mod I engine. Spring loading of the Mod I seal is compatible with the increased size, and the seal housing in both engines is maintained at the minimum cycle pressure.

The PL seals are designed to act as seals, not as guides for the rod. To minimize any static radial loading, the seals are mounted on separate conical seats, allowing them to move laterally and align themselves with the piston rods during assembly. When assembly is complete, the seals are effectively clamped in position by the seal loading spring, so no movement of the seats should occur during engine operation.

The motion of the piston rod during engine operation is a function of the crosshead/guide clearance, piston/liner clearance, and alignment of the crosshead guide and cylinder liner. Manufacturing tolerances and assembly must be controlled to minimize the radial dynamic loads on the seals, which adversely affect their performance.

Manufacturing drawings showed that there would be no significant difference in the motion of the piston rods in the P-40 and Mod I engines; however, this did not take into account that the Mod I engine crosshead guides were were made from aluminum, and those in the P-40 engine were made from steel.

Analysis revealed that the differential expansion of the aluminum crosshead guide relative to the steel crosshead would result in an operating crosshead clearance approximately twice that at room temperature (see Table 3-6), thus having a definite adverse effect on seal operation, and probably precipitating premature failure.

To investigate this further, P-40 engine No. 4 was assembled with different clearances on the four crossheads (actual clearances were 40, 82, 100, and 120 μm, respectively), and run for 270 hours under the accelerated duty cycle. During this time, no seal failure or degradation in engine performance was evident. When the engine was disassembled, oil leakage was found to be significant past the main seals, the leakage increasing with the crosshead clearance.

Table 3-6
Comparison of Crosshead Clearances Based on BOM Dimensions

Engine	Crosshead Clearance
P-40	20-50 μm
Mod I at 20°C	30-60 μm
Mod I at 85°C	80-120 μm

With the 40-μm clearance, a trace of oil was found between the seal and seat, and between the seat and its mating surface, but no oil was present in the seal housing. With the 82-μm clearance, oil was found at the upper edge of the seal, and traces of oil were found on the seal loading sleeve. With the 100-μm clearance, liquid oil was found over all the components in the lower part of the seal housing and halfway up the loading spring. With the largest crosshead clearance (120 μm), oil was found to be present not only throughout the seal housing, but it had also passed through the cap seal and supply bushing, and entered the working cycle. Oil was also present on the surfaces of the piston base and piston rings.

To generate additional information as to the effect of crosshead clearance, two additional tests are currently underway. P-40 engine No. 9 is operating under the accelerated duty cycle with four different crosshead clearances (18, 41, 52, and

65 μm, respectively), covering a narrower range than that previously used in P-40 engine No. 4. To date, the engine has completed more than 400 hours with no indication of oil leakage or degradation in engine performance.

Mod I engine No. 2 was fitted with steel crosshead guides with clearances of 40, 43, 45, and 51 μm, respectively, for the second test. This engine has also completed more than 400 hours of operation under the accelerated duty cycle with no indication of oil leakage of performance degradation. In order to generate seal life data, this test will be continued until seal failures occur.

Oil entering the working cycles of an engine is known to have a serious effect on engine performance, mainly by contaminating the regenerators and/or piston rings; as a result, any leakage of oil past the main seals is considered unacceptable, and constitutes a seal failure, even if the oil has not yet entered the working cycles. This could lead to inconsistent seal life data, since it is open to interpretation by the individual conducting the inspection. It could also generate pessimistic data when seals are replacing due to the presence of a slight trace of oil in an attempt to preclude a subsequent catastrophic failure.

Experience has also shown that engines can continue to function normally after a large amount of oil has leaked into the cycles, suggesting that an engine does have some degree of tolerance for oil contamination. Unfortunately, it is not possible to quantify or monitor this since the effect on the engine may vary depending on the form of oil leakage past the main seals, and the route the oil takes to enter the working cycles.

In order to generate more realistic seal life data, a new criteria has been adopted to define seal failure due to oil leakage, i.e.: "the elapsed time before oil enters the working cycles and degrades the performance of the engine by a specified and measureable amount."

Standard procedures have been drawn up to ensure consistent application of this definition in development engines. In addition, all test cell engines will be fitted with four absorption filters (one in each of the Pmin lines connected to the bases of the seal housings) which will prevent oil that has leaked into the seal housings from being recirculated with the hydrogen, and allow the oil leakage to be monitored.

PISTON RINGS

Piston ring testing during the last half of 1983 concentrated on the Mod I Motoring Rig, which was assembled as an upgraded Mod I Lightweight Reduced Friction Drive (LRFD). The engine was initially assembled with the BOM Mod I split/solid piston ring system. Measurements of motoring power were made over a range of speeds and pressures to provide baseline data that was consistent and had minimal scatter (shown in Figure 3-32).

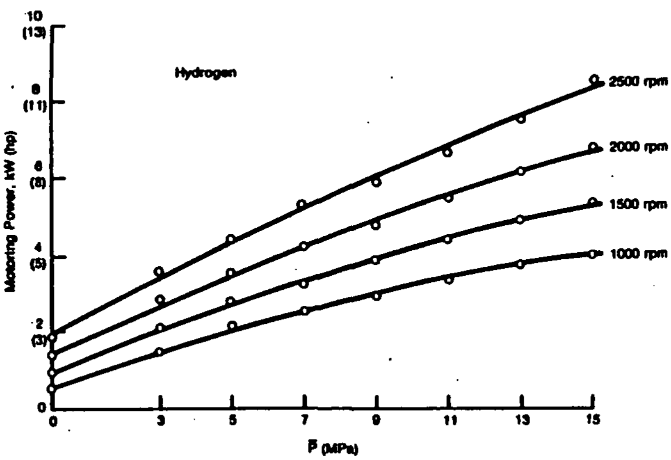


Fig. 3-32 Baseline Motoring Power

Following the baseline tests, new pistons were installed to test an alternative pressure-balanced piston ring system with only one ring (H-ring) per piston (Figure 3-33). The two O-rings isolate the inner surface of the piston ring from the cycle pressures.

Pressure balancing is provided by radial passages that communicate the mean pressure in the leakage path between the ring and cylinder wall to the inner surface of the piston ring. Radial loading is applied to the piston ring by the expander rings (in this case, standard, elastomer Quad-rings).

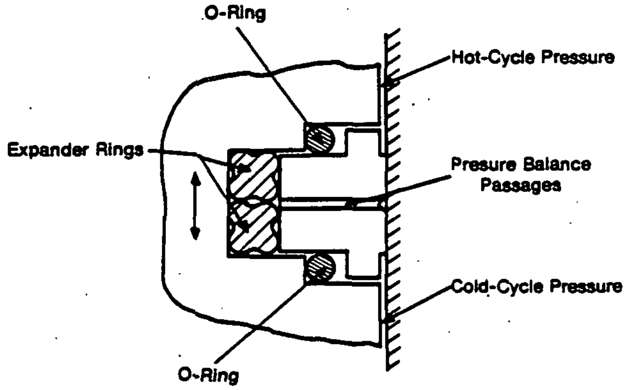


Fig. 3-33 Single H-Ring

The piston ring dimensions in Version 1 were chosen to give a nominal 25% compression of the Quad-rings when assembled on the pistons and installed in the cylinders. Solid, uncut piston rings were used in this version, and the piston rods were the same as those used for the baseline system, but with the redundant Pmin connections through the hollow rods sealed off by plugs inserted in the upper ends of the rods.

Over the same range of speeds and pressures used for the baseline tests, the piston rings in Version 1 provided effective cycle-to-cycle sealing, with the variations in cycle mean pressures significantly lower than those measured in the baseline tests (see Figure 3-34).

In a running engine, 0.45 MPa (~70 psi) is considered to be the maximum acceptable cycle-to-cycle pressure variation. During the baseline tests, the cycle-to-cycle pressure variations were generally within this limit. The limit was exceeded in two cases, however, i.e., the single H-ring cycle-to-cycle pressure variations were mainly less than 20 psi, and the maximum was less than 40 psi.

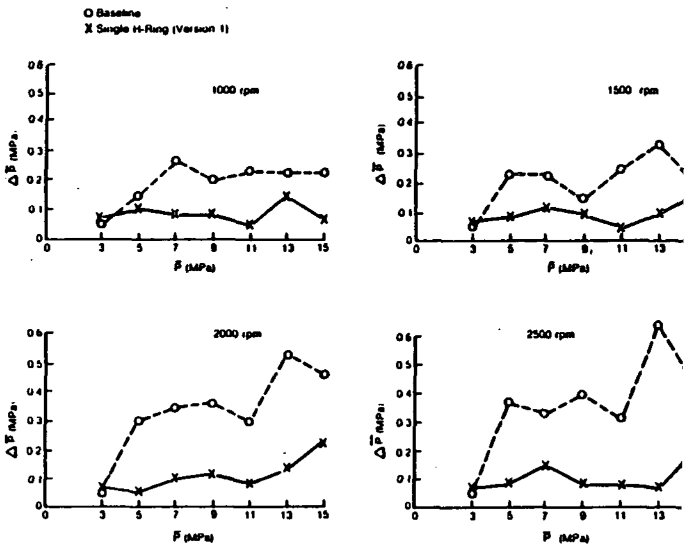


Fig. 3-34 Maximum Difference Between Cycle Mean Pressures, ΔT With Hydrogen

The primary objective of the pressure-balanced piston rings was to eliminate pressure loading of the rings in order to reduce friction, and reduce the power required to drive the motored engine; however, the opposite occurred (shown in Figure 3-35). The power required to motor the engine with the single H-rings was greater than the baseline values in all cases.

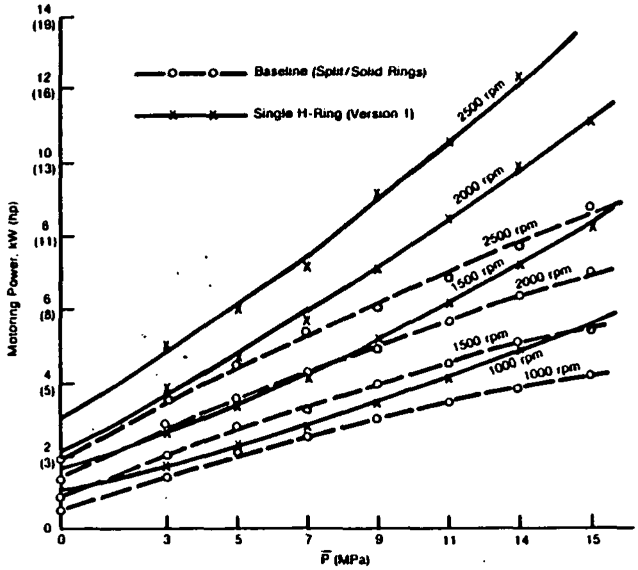


Fig. 3-35 Tests Carried Out With Hydrogen (Version 1)

The differences in motoring power increased with both speed and pressure. At 2500 rpm/15 MPa, the motoring power with the H-rings was more than 5 kW (6.7 hp) greater than the baseline power, initially suggesting that the friction associated with the single H-rings was substantially greater than that of the Mod I split/solid ring system, but the single H-rings would have to be very heavily loaded to account for the total difference in motoring power.

Excessive wear of the single H-rings would be expected with the heavy loading required, but this was not the case. When the rings were examined after ∿50 hours of operation, they were not deformed, and the wear was compatible with that normally encountered during run-in.

Overall, it would appear that the other factors in the motored engine system must have contributed significantly to the increased drive power requirements. Gas leakage across the piston rings in the Mod I BOM piston ring system always flows from the cycles into the Pmin manifold; with the single H-ring system, gas leaks from one cycle to another. This difference could have an effect on the form of the cycle pressures and their phase relationship with the crankshafts, which could be reflected in the drive power requirements. Visicorder traces of the cycle pressures showed that the cycle pressure ratios with the H-rings were essentially identical to those in the baseline tests, but improvements in the Data Adquisition System will be necessary to accurately compare the different cyclic pressures.

Motoring tests have also been performed with a second set of single H-rings (Version 2) in which the axial dimensions of the piston rings were reduced slightly to prevent the possibility of interference between the rings and their grooves, and to ensure that the O-rings could not contribute to the radial loading. The radial dimensions of the rings were changed to reduce the compression of the quad-rings to a nominal 17%.

Motoring power requirements with the Version 2 rings (Figure 3-36) are still greater than the baseline, but the differences are less than with the Version 1 rings. The difference in drive power at 2500 rpm/15 MPa is ∿2 kW (∿2.7 hp).

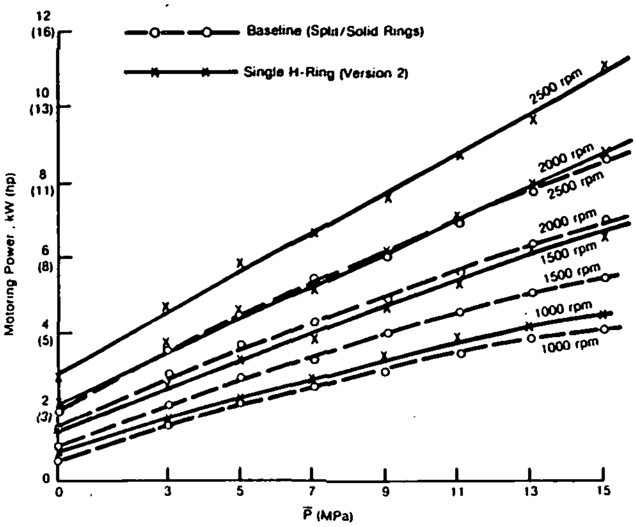


Fig. 3-36 Tests Carried Out With Hydrogen (Version 2)

Two teeth failed on the drive shaft gear during the final stages of Version 2 H-ring testing, necessitating complete teardown of the engine to replace the gears and remove debris. During this procedure, the surface of one of the crankpins was found in an advanced state of deterioration due to fatigue. To rectify this and avoid the possibility of similar failures on the other crankpins in the short term, both crankshafts were replaced. The engine has been rebuilt, and tests will be conducted to establish a new baseline before proceeding with further single H-ring testing.

Engine Drive System Development

The primary goal of this task is to develop a Lightweight Reduced Friction Drive (LRFD). Activity during the first half of 1983 concentrated on testing the drive in upgraded Mod I engine No. 5. During the last half of 1983, the LRFD experienced six failures (see Tables 3-7 and 3-8) - four on upgraded engine No. 5, and the other two on the Motoring Rig.

The locations of the six failures (shown in Figure 3-37) can be divided into three main categories: drive shaft/gear bolt failures, crankshaft/bearing failures, and wrist pin bearing failures. A set of tasks to investigate the cause of the failures, along with their design solutions, were established (Table 3-9) and completed.

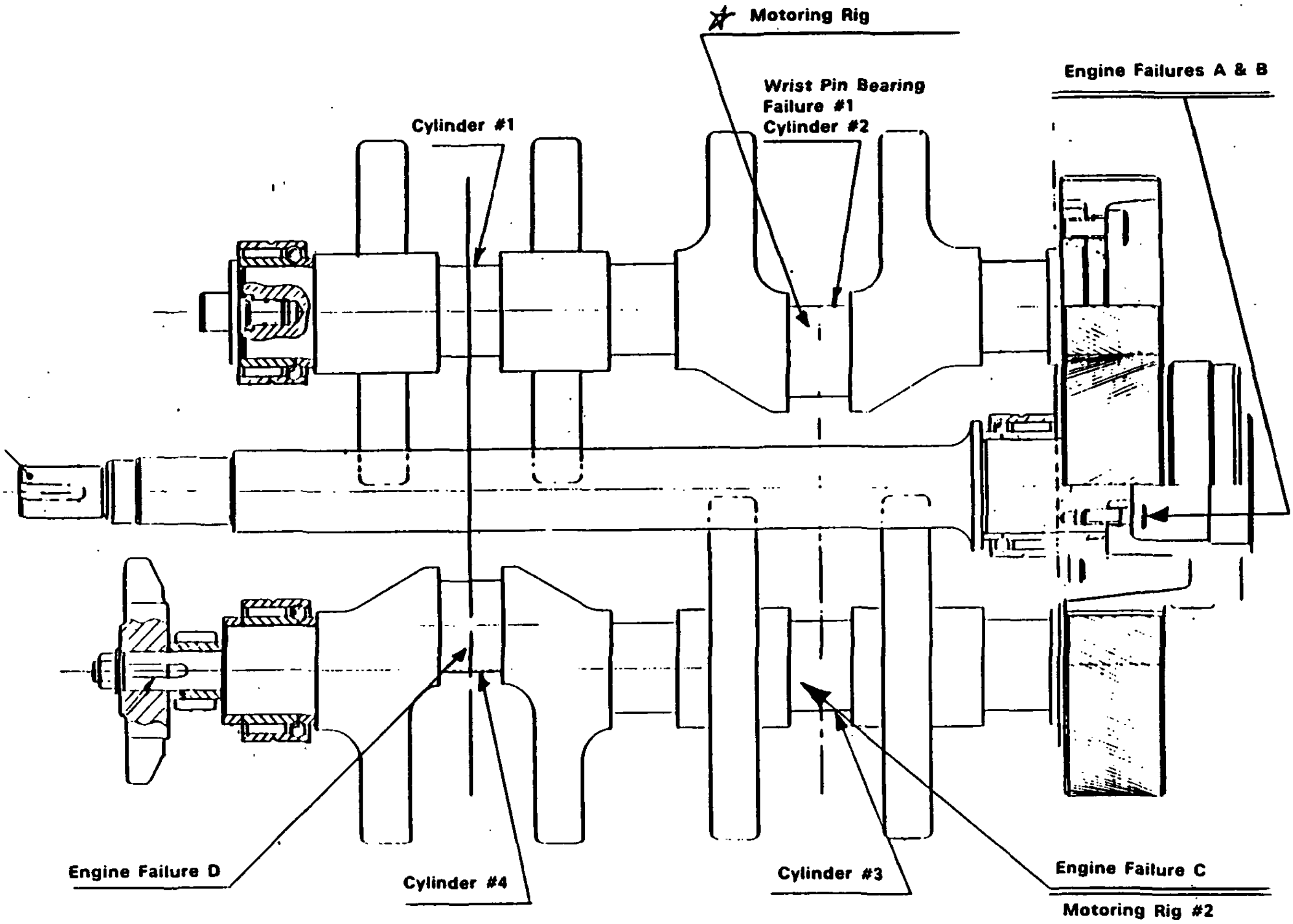


Fig. 3-37 Drive System Failures

Table 3-7
LRFD Engine Failure History

Engine Time (Hrs)	LRFD Total Times (Hrs)	Prime Failure	Comments
119.9	169.9*	Output Shaft Bolt Failure A	Engine stopped in 3-5 sec (200-300 rev) from 15 MPa/4000 rpm (Ref #1).
129.9	179.9*	Output Shaft Bolt Failure B	Same as above, but with gear and shaft damage (Ref #1 and #2).
136.5	186.5*	Extreme spalling over 180° of shaft journals in cylinder #3 - Failure C	Evidence of spalling on all journal surfaces and wrist pins (Ref #1 and #2).
212.0	37.5*	Bearing failure at cylinder #4 - discolored connecting rod indicates overheating - Failure D	Occurred while running 15 MPa line. Evidence of spalling as on prior failure (Ref #2).

*50 hours endurance on Motoring Rig added

Table 3-8
Motoring Rig LRFD Failure History

Rig Time (Hrs.)	Failure	Comments
82	Failure of wrist pin outer races on cylinder #2. Wrist pin scored and badly pitted on high load surface. (Failure # 1)	Secondary damage occurred to crank bearing. Investigation terminated, as prior analysis indicates redesign in order (Ref #16).
56.4	Gear tooth broken; cylinder #3 crankshaft bearing damaged (almost identical to first failure of engine No. 5) (Failure #2)	No further action taken.

Table 3-9
LRFD Failure Analysis

Drive Shaft/Gear Bolt Failures

Task	Results
Metallurgical Analysis	Failure A - Bolts failed in tension
Metallurgical Analysis	Failure B - Bolts failed in tension; one bolt loosened several turns before all bolts failed. Chatter pattern and scratches on inner and outer bearing races indicated races were not secure.
Design Review	Axial clearances between the bearing races, drive shaft, gear, and crankcase allowed excessive axial movement of the shaft and "hammering" of the shaft and gear. ● Worst Tolerances - .029" ● Shaft Measurements - .015"
Rebuild and Test	45 hours of trouble-free operation with races shimmed to 0.001-0.002" interference. Technicians observed "gear noise" was considerably reduced on this build.

Failure C - LRFD Bearing Failure Analysis

Task	Results
Dimensional inspection of parts	All within specification.
Inspection of hardened surfaces	Crankshaft - inadequate case depth. Wrist pins - adequate case depth.
Materials analysis by bearing manufacturer (INA)	Material inclusions; 70 Rc surface hardness "too hard" (58 Rc minimum, maximum not specified).
Reevaluate crankshaft material selection	Different shaft material will be required: ● 4320 - ASTM-A-45 selected. ● Discussions with metallurgist at Mercury Marine and Atlas Steel resulted in above "equivalent forging" quality ASTM-A-534.

Failure D - LRFD Bearing Failure Analysis

Task	Results
Dimensional inspection of failed parts	All measurements within specifications.
Lubrication system inspection	Two spray nozzles were plugged - no or minimal lubrication provided to #4 bearing. Machining chips found in passages of nozzle block.
Review of lubrication system design	Recommend nozzle blocks be redesigned with "last chance" filters and passages enlarged. Recommend system designed specifically for rolling-element bearings.
Review wrist pin/ bearing design	Design marginal - depends on equal loading of bearings. Value of rolling-element questionnable since rotation is only 5° (estimated gain in power loss <.1 kW). Recommend return to Mod I sleeve bearing design.
Investigate cause of crosshead and crosshead guide damage	In progress - H2 compressor crosshead being sectioned and examined.

The investigation resulted in the following conclusions and recommendations for the LRFD.

CONCLUSIONS

1. Bolt failures were caused by loose axial clearances on bearing races, allowing chucking that hammered bolts and resulted in tension failure of bolts.

2. The first crankshaft drive bearing failure resulted from a combination of inadequate hardened case depth, high surface hardness, and the presence of inclusions in the crankshaft material. There is no evidence of crankshaft bearings being overloaded or underdesigned.

3. The second drive bearing failure resulted from oil starvation in the oil spray nozzles.

4. The Motoring Rig wrist pin failure was expected, i.e., the bearings were undersized to fit into limited space, and depended on equal, balanced loading on each bearing.

5. The Motoring Rig crankshaft failure on cylinder #3 was identical to the crankshaft failure on upgraded Mod I engine No. 5 (item 2 above).

REDESIGN RECOMMENDATIONS

1. Review the design of all bearing races, and change where necessary to produce a 0.001" to 0.002" interference fit on all races.

2. Modify the drawings to reflect material and specification change to 4320 steel spec ASTM-A-45 "Equivalent Forging" quality of ASTM-A-534.

3. Review the oil-lubrication system, and modify it specifically for rolling-element bearing requirements.

4. Redesign the nozzle blocks to include "last change" filters and enlarged passages.

5. Eliminate rolling-element bearings on wrist pins, and return to the Mod I sleeve bearing design.

6. Review drive shaft bearing loads and L10 life, and evaluate the material specifications required for adequate shaft durability.

7. System life calculation approach for rolling-element bearings in a system.

8. Update design drawings to reflect current counterweight dimensions.

Control System/Auxiliaries Development

The major goals of this task are the development of reliable, efficient, low-cost engine control/auxiliaries systems. Specific control systems under development are a simplified mean pressure control, a reliable microprocessor-based electronic engine control, and a highly flexible Digital Air/Fuel Control (DAFC) with low pressure drop and low minimum fuel flow. The major auxiliary effort is the development of a positive-displacement combustion air blower. These hardware designs must be compatible with the extremes of an automotive operating environment.

Development during the last half of 1983 focused on evaluating combustion control system behavior during transient operating conditions, Digital Engine Control (DEC) software/reliability improvements, and conceptual design and analysis of a suitable combustion air blower.

Activities and goals for control systems development for the first half of 1984 will emphasize continuation of the reliability improvements in DEC software and hardware, and the hardware implementation of the DAFC.

Auxiliaries efforts will center on testing the positive-displacement combustion air blower, reviewing engine check valve design/reliability, and the conceptual design of a hydrogen compressor to provide improved pump-down capability.

COMBUSTION CONTROL

The primary goal of this task is to develop a DAFC that provides more system flexibility and better control of the air-to-fuel ratio during transients than the current Bosch K-Jetronic System. A wire-wrapped version of the DAFC was tested in September, 1983 in conjunction with an air-atomized nozzle installed in a Mod I engine (Lerma vehicle). Various λ curves were programmed during CVS cycle testing for evaluation of the DAFC and its effect on emissions and fuel economy during transient conditions. Table 3-10 compares the results of the Bosch and the DAFC. The effect on emissions and fuel economy was also determined for various blower speed map coordinates and airflow time constants.

Overall fuel economy was effected by the high HC concentration. Although DAFC ignition times were relatively short and comparable with the K-Jetronic A/F Control, a possibility exists that incomplete or unstable combustion may have taken place, causing high HC concentrations. This could be due to a number of factors, e.g.:

- insufficient atomizer airflow to the nozzle or too much fuel at ignition, causing inadequate mixing of the at-atomizing air/fuel in the nozzle; or,
- insufficient combustion air being supplied to the combustor turbulator due to an incorrect blower speed map and/or air throttle operation.

Decreased/increased airflow transducer time constants indicate an adverse effect, causing an increased HC concentration and lower fuel economy, thus leading to an initial conclusion of incomplete combustion occuring until the system stabilizes. Figure 3-38 indicates the trends in transient responses during CVS urban testing. Fluctuations of the A/F ratio during hard transients are shown to be higher than expected.

Table 3-10
CVS Cycles Comparative Test Data

Urban Cycles

Date		A/F Cont.	EGR	HC g/mi	CO g/mi	NOx g/mi	F/E mpg	T/C MS
9/82	Baseline	Bosch	I	.253	3.3	.9	19.3	N/A
5/83	Average	Bosch	M	.64	1.26	.522	19.7	N/A
9/83	Average	Bosch	M	.556	.71	.54	19.3	N/A
5/83	Average	DAFC	I	.656	5.24	.78	20.4	100
9/83	Average	DAFC	M	1.55	3.11	.50	18.2*	100
	Blower Map Adjusted	DAFC	M	1.09	3.2	.44	18.6*	100
	Airflow T/C Dec.	DAFC	M	2.1	2.06	.52	17.7*	50
	Airflow T/C Inc.	DAFC	M	2.2	2.28	.52	17.5*	200

Highway Cycles

Date		A/F Cont.	EGR	HC g/mi	CO g/mi	NOx g/mi	F/E mpg	T/C MS
9/82	Baseline	Bosch	I	.0043	.311	.66	32.1	N/A
5/83	Average	Bosch	M	.007	.1	.342	35.0	N/A
9/83	Average	Bosch	M	.002	.10	.32	36.0	N/A
5/83	Average	DAFC	I	.024	.75	.45	33.9	100
9/83	Average	DAFC	M	.0085	.23	.33	34.0	100

T/C = Airflow Transducer Time Constant; I = Intermediate; F/E = Fuel Economy; M = Maximum

*Hot 505 Cycle

A more compact printed circuit-board version of the DAFC has been designed, and procurement of hardware and fabrication is underway. Bench characterization and evaluation of alternative fuel metering pump/motor combinations are currently underway. Speed sensing via an integral tachometer will be used for the fuel feedback signal, thereby eliminating the fuel flow meter, reducing hardware cost, and improving component packageability.

Further development tasks scheduled for the DAFC during the first half of 1984 are:

- bench characterization of air mass flow meters (hot-wire and hot-film) configurations);
- Lerma vehicle testing (idle and transient conditions):
  - MTI conical nozzle testing and performance evaluation with the K-Jetronic/DAFC;

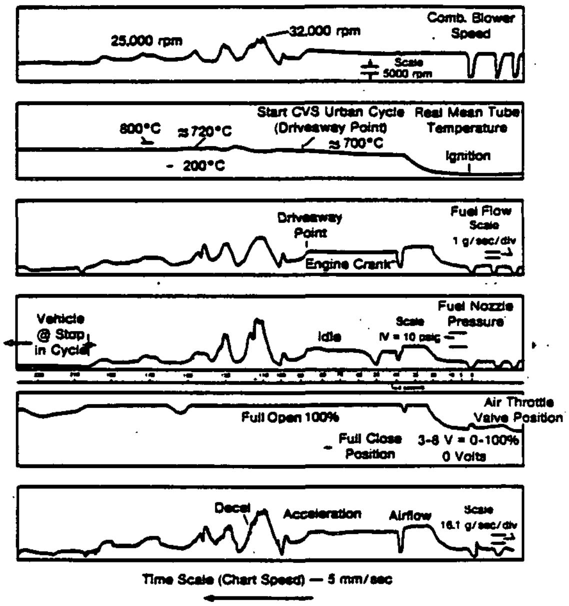


Fig. 3-38a CVS Urban Cycle (Hot 505)

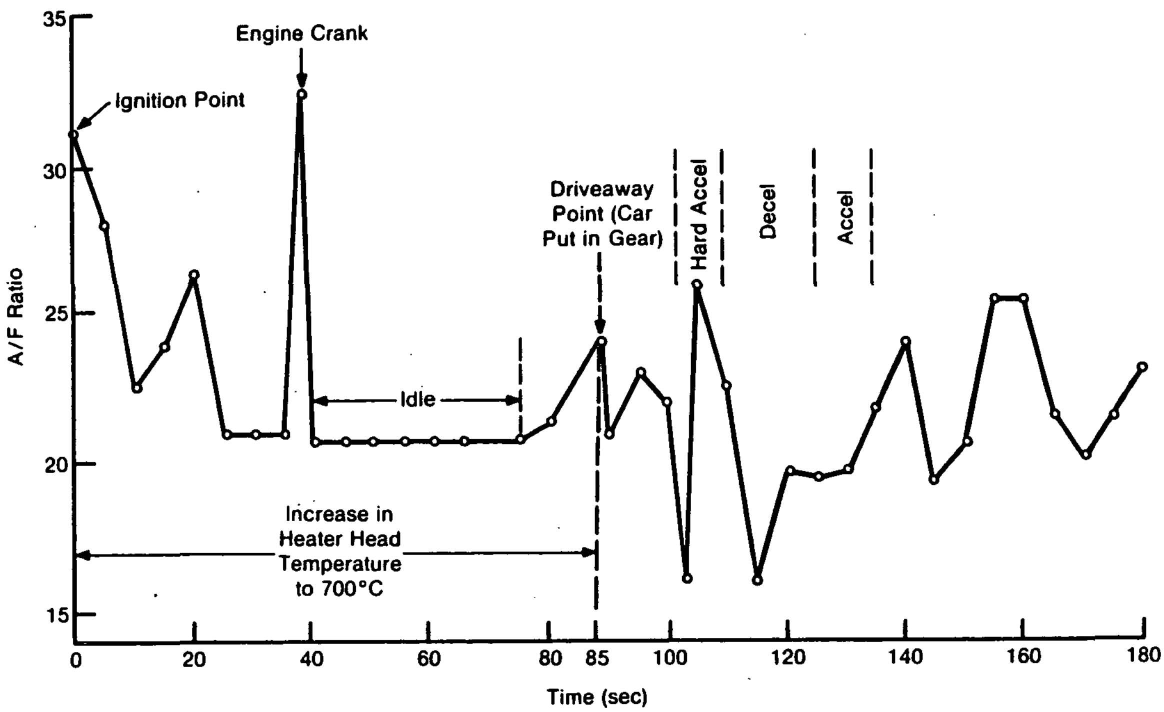


Fig. 3-38b CVS Hot Urban Cycle - A/F Ratio Versus Time

- performance evaluation of air mass flow meters;

- testing of electrical atomizing air compressor;

- evaluation of vehicle operation with different response times for airflow and fuel flow;

- evaluation of λ variation as a function of time; and,

- evaluation of the effect of fuel flow meter location on A/F ratio;

• linearization of the air throttle valve map; and,

• integration of the DAFC/DEC design into the Spirit vehicle.

DIGITAL ENGINE CONTROL (DEC)

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

The printed circuit-board version of the DEC installed in the ASE test cell continues to operate successfully in steady-state control. Minor modifications to the system ground paths were made to improve electrical noise immunity. Fully automatic engine starts are now standard operating procedure, thus demonstrating the operator's confidence in the control system.

A second DEC has been installed in the AMC Spirit vehicle. Prior to installation, this system was environmentally tested from -30°C to +70°C. The DEC started and controlled the engine properly during initial shakedown testing (performed on a chassis dynamometer)*. The following software modifications have been implemented:

• temperature control logic was changed to evaluate the mean temperature of the 8 rear-row T/C's with additional filtering on the value (open T/C's are not included in rear-row mean temperature evaluations, nor are temperatures significantly different from the value of the previous rear-row mean temp)**;

• communications routines for the DEC were modified to accommodate the monitor hardware/software package;

• vehicle cooling fan logic is now incorporated into the DEC algorithm, i.e., vehicle speed is read into the A/D converter channel, and the following logic evaluated:

  fan on if block temp > 80°C, or if block temp > 40°C, accelerator < 90% of full depression, and vehicle speed < 20 mph;

• hysteresis is applied about all fan control parameter breakpoints;

• temperature scaling logic was changed to utilize Type-K T/C's in the system (Type-J was used previously to measure water temperature);

• limit checks were incorporated on D/A outputs to keep the analog signal output at its max value when the digital value exceeded the corresponding max value (provides for overflow check);

• a significant amount of "housekeeping" has been done in the software to streamline as well as improve the algorithm's readability; and,

*All work on this vehicle was performed by MTI personnel, including construction and fabrication of a new dashboard panel and center console.

**Previous control temperature was the average of 4 rear-row T/C's.

* blower map coordinates have been changed to improve temperature control at idle conditions.

MEAN PRESSURE CONTROL

The objectives of this task activity are to develop simplified engine power-control valve that would mount directly on the engine, reduce the number of check valves in the engine and be electrically actuated. The valve will be designed for low leakage, seal reliability, and ease of fabrication/assembly.

A conceptual design was selected that mounted directly on the (V-block) engine, had rotary motion, and contained a single metering plate. The number of check valves required was reduced to four (one per cycle). These four valves are required to monitor the maximum cycle pressure (Pmax) for control purposes; however, their operating environment is much less severe than in the Mod I or upgraded Mod I engines.

A detail design of this concept has been completed, and analysis of the design revealed unexpectedly high operating torques (2-5 N•m). The electric motor required to operate (position) the valve with the required speed was determined to be quite large (200 watts). The analysis also indicated the necessity of keeping the hydrogen storage tank at a pressure ~5 MPa higher than in the Mod I in order to prevent engine cycle performance degradation during up-power transients.

This latter problem was deemed surmountable by modifying the engine's timed injection hardware and the power-control strategy. Reduction of the operating torque was attempted by using alternative bearings and seal materials, but it was unsuccessful. This design was not committed to hardware, primarily due to the high operating torque.

This task will continue during 1984, but with slightly modified goals. Due to the reliability shown by the check valves in USAB's Endurance Engine, their use will be continued in the Mod II. Eventual elimination or reduction in their numbers will be addressed at a later date. Near-term design efforts will focus on developing an alternative power-control valve to replace the Mod I/upgraded Mod I linear valve. Major areas to be addressed include electrical actuation instead of the current electrohydraulic actuator, more reliable seals, decreased sensitivity to contamination (dirt), interchangeability of valves with minimal control system adjustments, and ease of fabrication/assembly.

MOD I/UPGRADED MOD I CHECK VALVES

The objectives of this activity were to review the Mod I check valve design and failure history, investigate the causes of check valve failures, and determine what design/quality control changes were necessary. The Mod I check valve is a modified Hawe ER2 check valve. The plate and seat are removed from a Hawe check valve and installed in a new (hardened) holder designed by USAB (Figure 3-39).

These check valves have demonstrated a test rig life in excess of 600 hours; life in the Mod I engine is less. Primary failure modes are contamination (corrosion, dirt, etc.), poorly machined holders, worn or broken seats, leaking sealing washers, and broken springs. Review of the design yielded no improvements. A survey of U.S. manufacturers (to identify alternative vendors) yielded no results. A quality control review indicated a need to tighten quality control requirements for the check valve holder, and to standardize the assembly/test procedures for the valves.

The check valves used in the Mod I/upgraded Mod I engines were generally determined to be adequate except that they were more sensitive to contamination than would be desirable. No further activity is planned in this area.

Auxiliaries Development

COMBUSTION AIR BLOWER

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

The detailed design of a "lobe" blower was completed during the last half of 1983, and a prototype blower was ordered. Testing of the prototype will begin in April, 1984. This design offers numerous benefits to the combustion air system, including a reduction in the number of system components, control complexity, power consumption, and cost. The differences in system complexity between the upgraded Mod I system (with the DAFC) and the variable, positive-displacement, blower-based system are shown in Figure 3-40. The number of components and control signals have been greatly reduced.

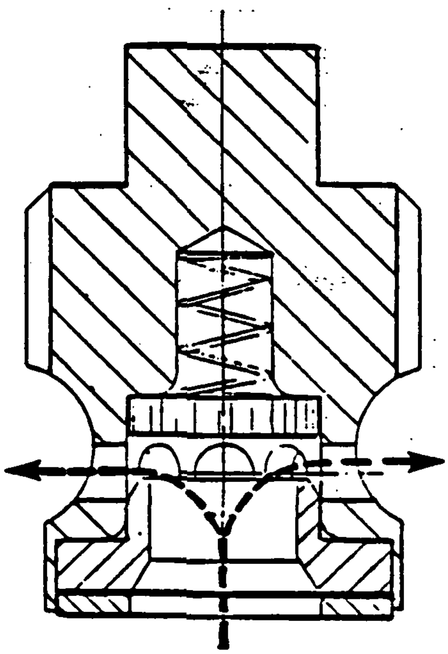


Fig. 3-39 ASE Mod I Check Valve

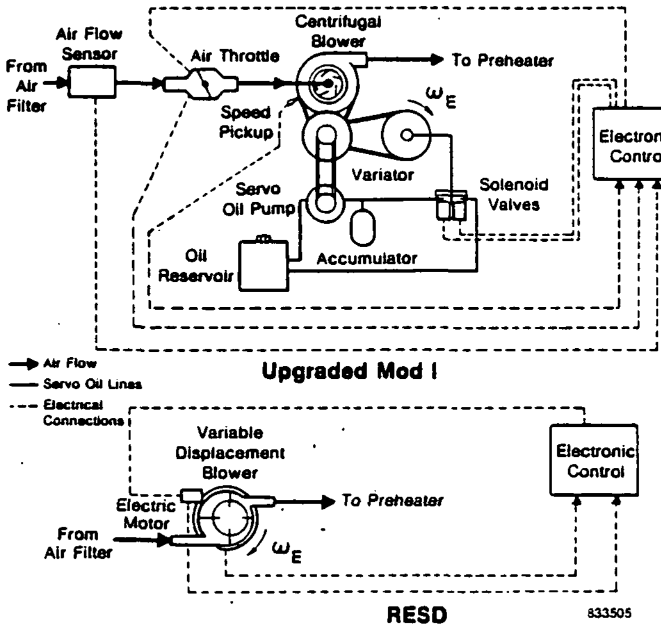


Fig. 3-40 Combustion Air System

IV. MOD I ENGINE TEST PROGRAM

The ASE engine numbering system required modification in order to more easily distinguish between the Mod I and upgraded Mod I engines, and to avoid confusion when rebuilding engines requiring major hardware changes. Since the original plan for the Mod IA and Mod IB models of the engine has been abandoned, the alpha numeric distinction is no longer necessary. Two major philosophical changes have been incorporated in changing the present numbering system, i.e., a "0" or "1" has been inserted to differentiate between the original Mod I and the upgraded version. The numbering series has also been assigned in sequential order. The following conversion table shows the old and new engine numbers.

Number of Cycles	Total Engine Swept Volume in c.c.	Mod I (0) or Upgraded Mod I (1)	Engine Serial Number
4	123	0 or 1	"X"

Old Engine Number	New Engine Number
4-123-1	4-123-0-1
4-123-2	4-123-0-2
4-123-3	4-123-0-3
4-123-10 (U.S.A.)	4-123-0-4
4-123A-10	4-123-1-5
4-123A-2	4-123-1-6
4-123-2 (Rebuild)	4-123-0-7
4-123A-11 (G.M.)	4-123-1-8
4-123A-12 (J. Deere)	4-123-1-9

Three Mod I/upgraded Mod I engines were operational in the United States and three at USAB during the last half of 1983. Those testing in the United States were engine No. 1 in the Lerma TTB, upgraded engine No. 5 primarily in MTI's test cell, and upgraded engine No. 8 in MTI's engine cell/ITEP Spirit vehicle. Those testing at USAB were engine No. 3 for endurance, upgraded engine No. 6 for performance, and engine No. 7 for seal life determination.

Testing during this semiannual report period focused on two areas: 1) upgraded Mod I performance development; and, 2) the building, testing, and installation of an upgraded Mod I engine in the ITEP program vehicle.

Mod I Engine No. 1

Mod I engine No. 1, installed in the Lerma TTB, has been extensively involved in transient Stirling-engine performance relating to emissions and fuel economy. The primary Lerma effort during the last half of 1983 was directed toward transient development of the DAFC System, and minor DEC System changes in an effort to improve mileage and emissions. As of December 31, 1983, the engine has accumulated a total of 900 operational hours.

The main objective of the CVS tests conducted during May and September on the MTI-design DAFC was to duplicate emissions levels obtained with the K-Jetronic A/F System. Previous testing was inconclusive because the system configuration had less restriction with the DAFC, thus creating less intake restriction, which affected EGR levels and, as a result, emissions levels. Tests were then conducted that created the same EGR levels experienced with the K-Jetronic. Resulting emissions were comparable between the two A/F ratio controllers.

A series of tests have been conducted on the TTB cooling system and the engine's emissions/fuel economy at various road loads. The tests were conducted at Canadian Fram's wind tunnel at speeds of 40, 60, 80, and 100 km/hr, at ambient temperatures of 60, 80, and 100°F, and at road loads of 7.4, 10.1, 11.1 (standard), and 22.1 bhp at 50 mph. The maximum top tank temperature reached during testing was

$84.7^{\circ}F$ at 100 km/hr, 22.1-bhp road load, and $100^{\circ}F$ ambient temperatures.

Figure 4-1 graphically represents the top tank temperature reached at 100 km/hour and various road loads and ambient temperatures. Of the four road loads evaluated, 7.4 bhp represents a Chevette-size vehicle, 10.1 bhp represents the AMC Spirit, 11.1 bhp represents the Lerma TTB as is, and 22.1 bhp represents the double road load condition for the Lerma. All testing was conducted with the same radiator core. The particular core configuration used was a Blackstone "Tropic."

A time-to-coolant boil test involving operation at 100 km/hr at $100^{\circ}F$ ambient with an 11.1-bhp road load was also conducted until stability was reached, at which time the speed was cut to idle, and the 100-km/hr wind was stopped while the top tank temperature was continuously monitored. Maximum top tank temperature during this test climbed from $68^{\circ}C$ to only $72^{\circ}C$ following the decrease in speed to idle. No boiling occurred.

Following the cooling systems tests, the Lerma was transported to the Mercedes-Benz Emissions Laboratory in Ann Arbor, Michigan for evaluation at the three different road load settings. The 11.1-bhp road load setting results compared favorably with the May, 1983 emissions/mileage results.

Figures 4-2, 4-3, and 4-4 are bar chart presentations of the emissions levels and fuel economy figures for the May and September, 1983 test runs. The best emissions levels and mileage figures occurred with the 7.4-bhp road load (Chevette simulation). Emissions levels at this particular load were .81 g/mi HC, .45 g/mi CO, and .42 g/mi NOx, with fuel economy figures of 22.4 mi/gal combined.

Future Lerma testing during the first half of 1984 will involve component development testing on EHS hardware. The engine will be removed from the Lerma in April, converted to an upgraded version of the Mod I, and sent to Daimler-Benz for subcontracted vehicle development testing in a light-duty delivery van.

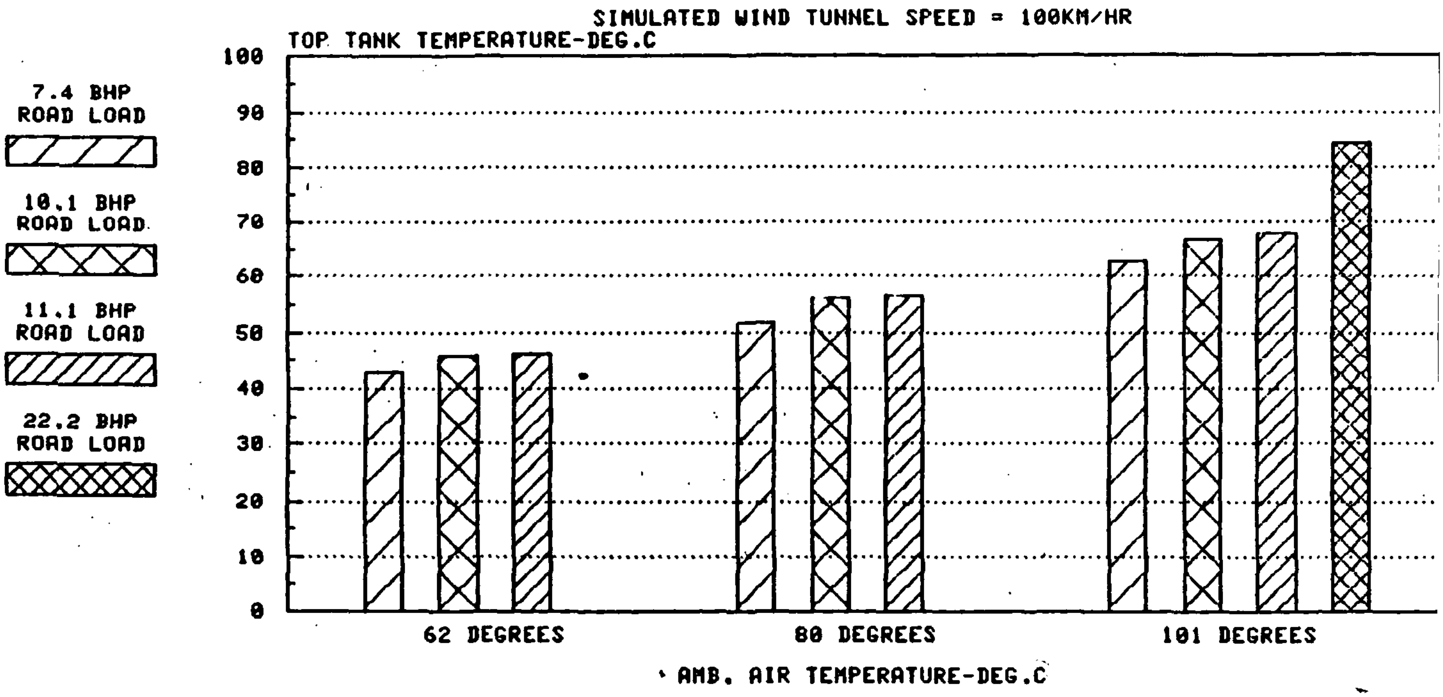


Fig. 4-1 Maximum Radiator Top Tank Temperature

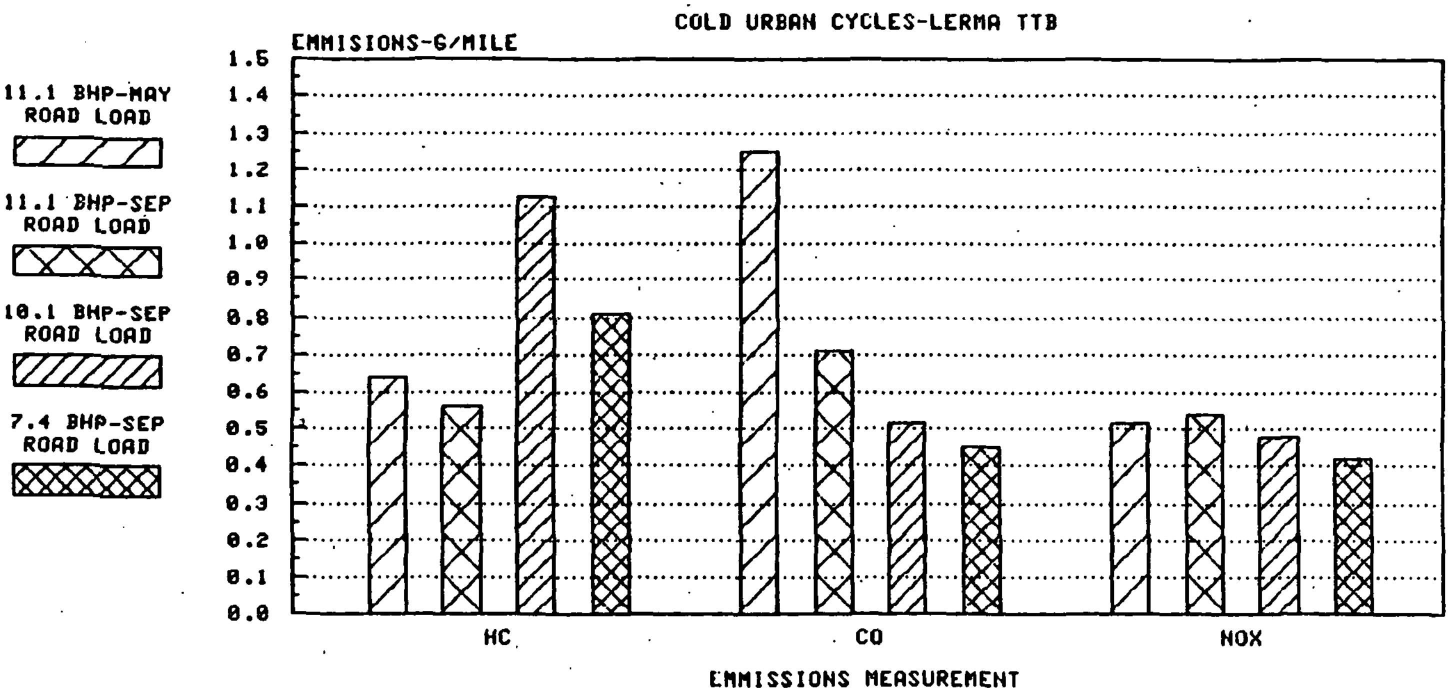


Fig. 4-2 Vehicle CVS Test Emissions

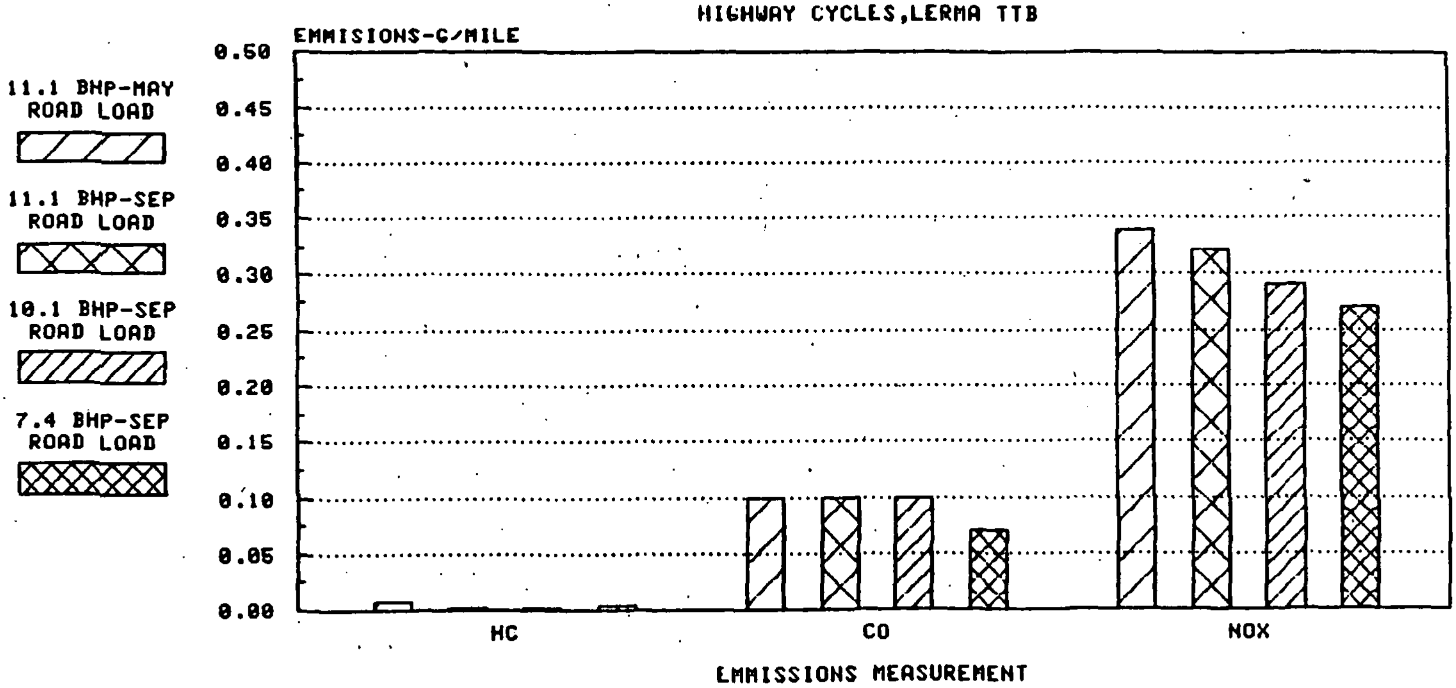


Fig. 4-3 Vehicle CVS Test Emissions

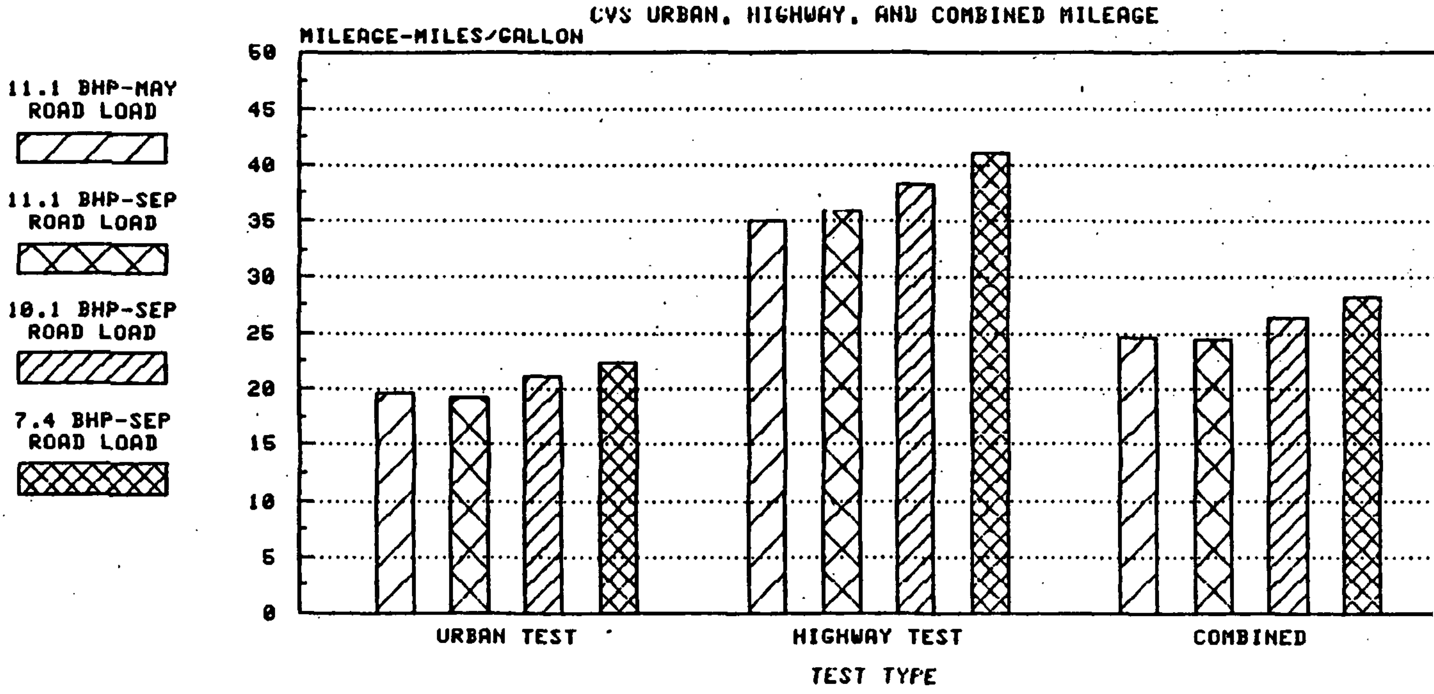


Fig. 4-4 Vehicle CVS Test Fuel Economy

Mod I Engine No. 3

During the second half of 1983, engine No. 3 successfully completed the first 1000-hour portion of its endurance test at 720°C, which began during the first half of 1983 in accordance with the test plan* (see Figure 4-5).

As expected, this test experience provided significant technical input for use in the ASE Program, with the most significant data applying to the External Heat System, auxiliaries, and Engine Drive System areas. The near-term development activity areas were identified, and component hardware and a specific test plan (MTI Report No. 84ASE362TP59) were generated for further endurance testing (to begin early in 1984). Key development areas defined by the test results are given in Table 4-1. Full analysis of the testing and the extensive teardown inspection that followed are given in MTI Report No. 84ASE361ER60, "ASE 4-123-0-3 Endurance Test Report."

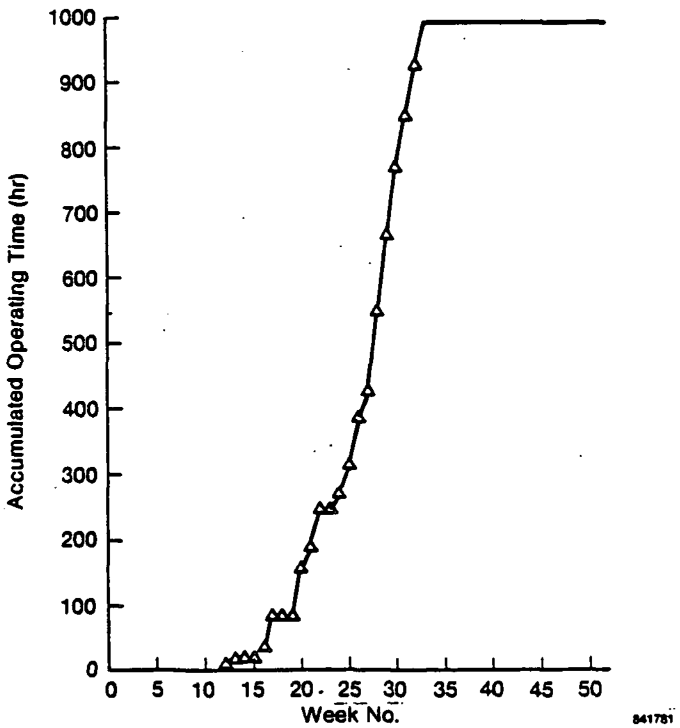


Fig. 4-5 Mod I Engine No. 3 Endurance Test - Accumulated Operating Time Through 12/31/83

*MTI Report No. 82ASE293TP44

Table 4-1
Key Development Areas Identified During 1000-Hour Endurance Test of Mod I Engine No. 3

System	Development Area
External Heat	Preheater Clogging Fuel Nozzle Clogging
Hot Engine	Heater Tube Gaps Regenerator
Cold Engine	Leaking PL Seals
Engine Drive	Crankcase Cracks
Auxiliaries & Controls	Power-Control Sys. Life; Blower Free-Wheel Life; Conn. Rod Bearing Life

Engine No. 3 completed 627 hours of testing during this report period, accomplished 42 starts, and collected 3 endurance check points. As of December 31, 1983, the engine was in final assembly and instrument installation in preparation for another extended testing period (200-400 hours).

RESULTS

Performance - The objective of this test was to monitor the operation of the Mod I engine during extensive, severe cyclic testing. To help track performance effects, a steady-state operating point (15 MPa/2000 rpm) was selected for periodic measurement.

A comparison between early and late endurance testing is given in Table 4-2, where power and efficiency are seen to go down despite the fact that combustor efficiency is somewhat higher during the later point. Nonetheless, the overall engine operation after 1450 hours at 15 MPa is still essentially at the minimum acceptance test requirements for a new engine (Figure 4-6).

Table 4-2
Performance Comparison - Pre and Post Endurance (15 MPa/2000 rpm)

Parameter	Initial (Pt. 128)	Final (Pt. 179)	%
P_{eff}	36.46 kW	33.34 kW	-8.5
η_{eff}	34.99 %	33.25 %	-5
η_B	86.53 %	89.44 %	+3.6

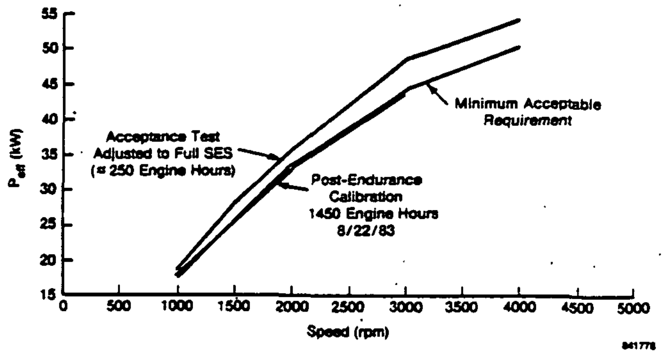


Fig. 4-6 Mod I Engine No. 3 1000-Hours Endurance Test - P_{eff} Versus Spread

Hardware - In addition to the items given in Table 4-2, other areas needing further development were discovered during the test, corrected, and improvements were confirmed. These areas included:

- the effect of an increased capacity oil filter for the seal house venting system on PL seal life;
- durability of the ASE working gas systems (Hot and Cold Engine Systems - all other engine parts exposed to hydrogen working gas) in maintaining their integrity and gross performance levels; and,
- rapid reclaiming in overall efficiency by periodic cleaning of the EHS preheater (Figure 4-7).

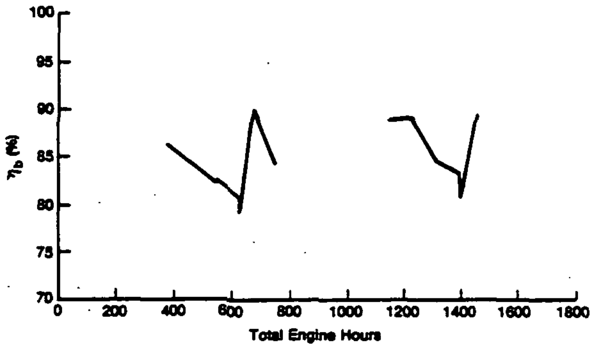


Fig. 4-7 Mod I Engine No. 3 1000-Hours Endurance Test - E_{T_a} Variation at 15 MPa/2000 rpm

Main Seal Life - Mod I engines have been plagued with short seal life (less than 100 hours) due to contamination of the working gas volumes by oil from the seal house. These engines were fitted with seal house venting systems with a small filter/check valve assembly. After the third seal failure (at 249.9 hours), the seal house system was fitted with a larger, replaceable cartridge filter in series with the existing Mod I filter/check valve (shown in Figure 4-8). This filter was monitored periodically (about every 100 hours) to measure oil leakage, and for replacement if necessary. With the installation of this modification, the engine was able to complete the endurance test without further seals failures or malfunctions.

The data obtained from weighing the filters (see Figure 4-9) was used to help set up an oil seal leakage criteria that is now used on the other engines to identify a particular failing seal area before it becomes necessary to perform a costly, time-consuming disassembly/cleaning/rebuild cycle for all working gas system components.

Although oil was found in the working gas space upon disassembly, the piston rings had functioned normally, indicating that improved but still-not-perfect seal systems will be sufficient for normal ASE vehicle operation.

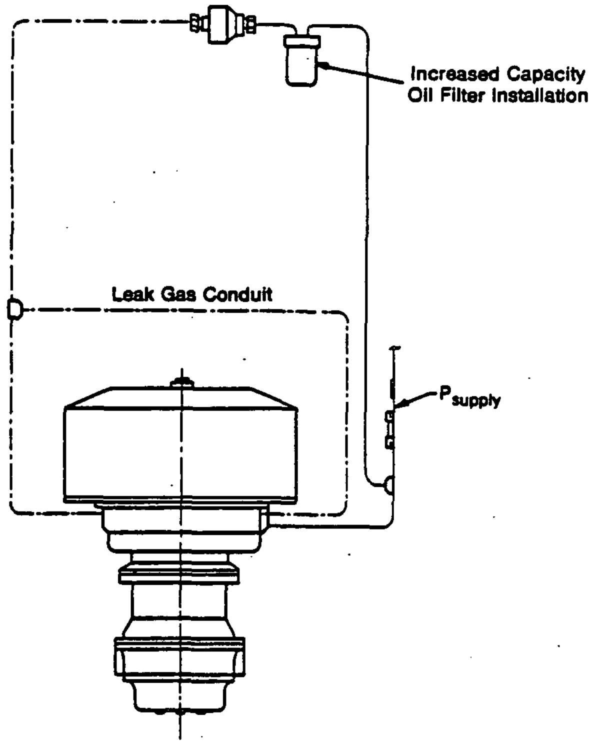


Fig. 4-8 Replaceable Cartridge Filters on Mod I Filter/Check Valve

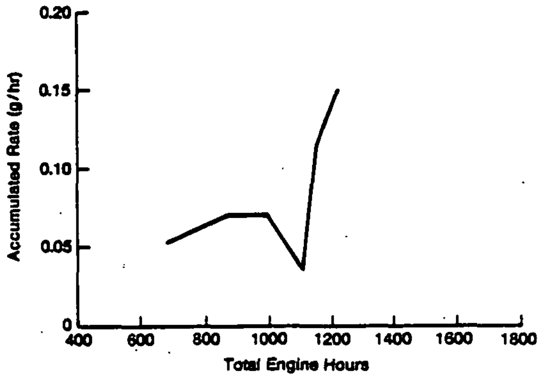


Fig. 4-9 Mod I Engine No. 3 Endurance Test - Oil Accumulation Rate (P-40 Filter)

Hardware Durability - Review of engine hardware revealed adequate durability for all engine parts exposed to high-pressure hydrogen working gas. Table 4-3 lists the subject parts and comments on their post-endurance conditions.

The deteriorated condition of the regenerators noted in Table 4-3 was not strongly reflected in post-endurance test performance levels (see Figure 4-6). The regenerators will be checked back-to-back during a future test to measure their current impact on power and efficiency levels. Post-endurance reaction of the regenerator matrix with normal atmosphere after disassembly is felt to contribute to their substandard condition.

Table 4-3
Working Gas System Part Condition Summary - Post 1000-Hour Endurance Test

Part	Comment
Heater Heads	Passed static pressure test; some growth of regen. housing cracks; still usable for further testing.
Domes	Minor contact with cyl. walls.
Pistons	Minor Rulon dust deposits.
Regenerators	Severely deteriorated (increased pressure drop); to be investigated further.
Coolers	Acceptable.
Piston Rings	Acceptable: uniform wear; lower rings worn more than upper.
Cap Seals	"Acceptable" wear.
Cylinder Liners	"Very little" wear.

<u>Effect of EHS Operation on Overall Efficiency</u> - The severe requirements of the CVS cycle upon which this endurance test was based include many rapid, large changes in fuel input to the EHS, i.e., ~100,000 load/unload cycles during the 1000-hour test, thus putting a heavy load on the A/F system to maintain a proper ratio at all conditions. It became apparent during the test that rich operation (insufficient air for the fuel supplied) was occurring due to a combination of engine/facility characteristics (will be addressed in future testing).

The result of the rich condition was the eventual, predictable (200 hours) buildup of carbon soot on the gas side of the preheater, and a consequent drop in heat transfer to the incoming combustion air. This energy loss resulted in the large efficiency decrease noted at the 15-MPa/2000-rpm check point. This deteriorated condition was found to be easily corrected by removing the preheater and cleaning it with compressed air (see Figure 4-7). Essentially, all the lost performance could be reclaimed in this manner, indicating the need to minimize a reoccurence of this "rich" condition in vehicle-installed ASE units, and the rapid "tune-up" effect of periodic, quick air cleaning of the preheater. All in all, the viability of the engine is encouraged by the findings of this extensive endurance/development test.

Upgraded Mod I Engine No. 5

Upgraded engine No. 5 was the first upgraded Mod I configuration to be built and tested. During the latter half of 1983, the engine accumulated hours conducting various performance tests, and was finally characterized at 720°C in October, 1983. At the time of its removal from the test cell, and its installation in the ITEP Spirit vehicle, the engine had accumulated a total of 345 operational hours.

Initial performance testing during the first half of 1983 revealed that the engine was below par when compared to performance predictions. When assembled at USAB, upgraded Mod I engine No. 6 had the same performance level problems, although not as bad.

The problem of the melting cylinder liner backup ring* was resolved by modifications to the cylinder housing neck passage. The original castings were weld-modified inside the passage to redirect the flow, and heater heads that use old Mod I cylinder housings (did not have the flow problem) were manufactured. Both of these modifications lowered wall temperatures, and allowed longer running times because of external backup ring life.

*MTI Report No. 83ASE334SA4

Several back-to-back performance tests were conducted on the engine, during which several important performance factors were determined. Table 4-4 summarizes these facts.

Two failures of the roller-bearing drive hardware occurred* during these performance tests. The problem turned out to be a crankshaft material-quality problem, but it meant returning to the journal bearings, and a reduction in engine power output/maximum efficiency.

Table 4-4
Back-to-Back Performance Test Summary Results of Upgraded Mod I Engine No. 5

	Effect on	
	Power (max)	Effic. (max)
Roller-Bearing Drive	+1.7 kW (+2.3 hp)	+2.5%
Hybrid Head (Mod I Cylinder Housing)	+1.4 kW (+1.9 hp)	+1.2%
Remanufactured Regenerators	0	0
Integrated Cylinder Liner/Duct Plate	-.3 kW (-.4 hp)	+.7%

During one of upgraded engine No. 5's down periods, several thermocouples were placed on the shadow side of the heater tube at the front row. They were located at the knee of the tube, at the midpoint, and at the top of the front row of one tube. Subsequent testing verified the USAB claims that the front-row tube temperature is almost identical to that of the rear row during steady-state running. During start-up, the knee area of the tube was found to exceed the 870°C maximum temperature limit set for the tubes. As soon as the engine starter engages, the temperature falls and eventually settles in at the back-row tube temperature setting.

Additional testing is scheduled for early 1984 to more thoroughly quantify the front-row tube temperatures during engine operation.

Upgraded engine No. 5 was characterized late in 1983. The original intent was to characterize at 720°C and again at 820°C back-row tube set temperatures. After running a few hours at 820°C, severe oxidation was observed on the back-row fins. As a result, running at 820°C was limited, and only the 5- and 15-MPa mean pressure level performances were measured. Attempts to run at 770°C were also terminated because of the oxidation problem. Using data that was run, it was determined that the increase in rear-tube temperature from 720 to 820°C will be worth 8.4 kW (11.3 hp) and ~1.4 points in maximum efficiency.

The fin oxidation problem was metallurgically examined and determined to be caused by a depletion of chrome from the 310 Stainless Steel fins during the brazing procedure of the heater head manufacturing process. An alternate procedure has been determined that will eliminate chrome depletion on future heads.

Characterization of the engine at 720°C rear-tube temperature was completed during this report period with the welded head and without the atomizing air compressor. Power and efficiency curves for this configuration and temperature are shown in Figures 4-10 through 4-13. Figures 4-14 through 4-17 plot the difference between the 720°C and 820°C temperature settings at 5- and 15-MPa mean pressure levels in power and efficiency with the weldable heads, and Figures 4-18 through 4-21 show the differences in power and efficiency with the hybrid heads at 720°C and 770°C.

During the last half of 1983, upgraded Mod I engine No. 5 was used to check out the MTI DEC in the ITEP Spirit vehicle. The engine was reinstalled for further testing in December.

*Analysis of the failures is covered in Section III, Drive System Development.

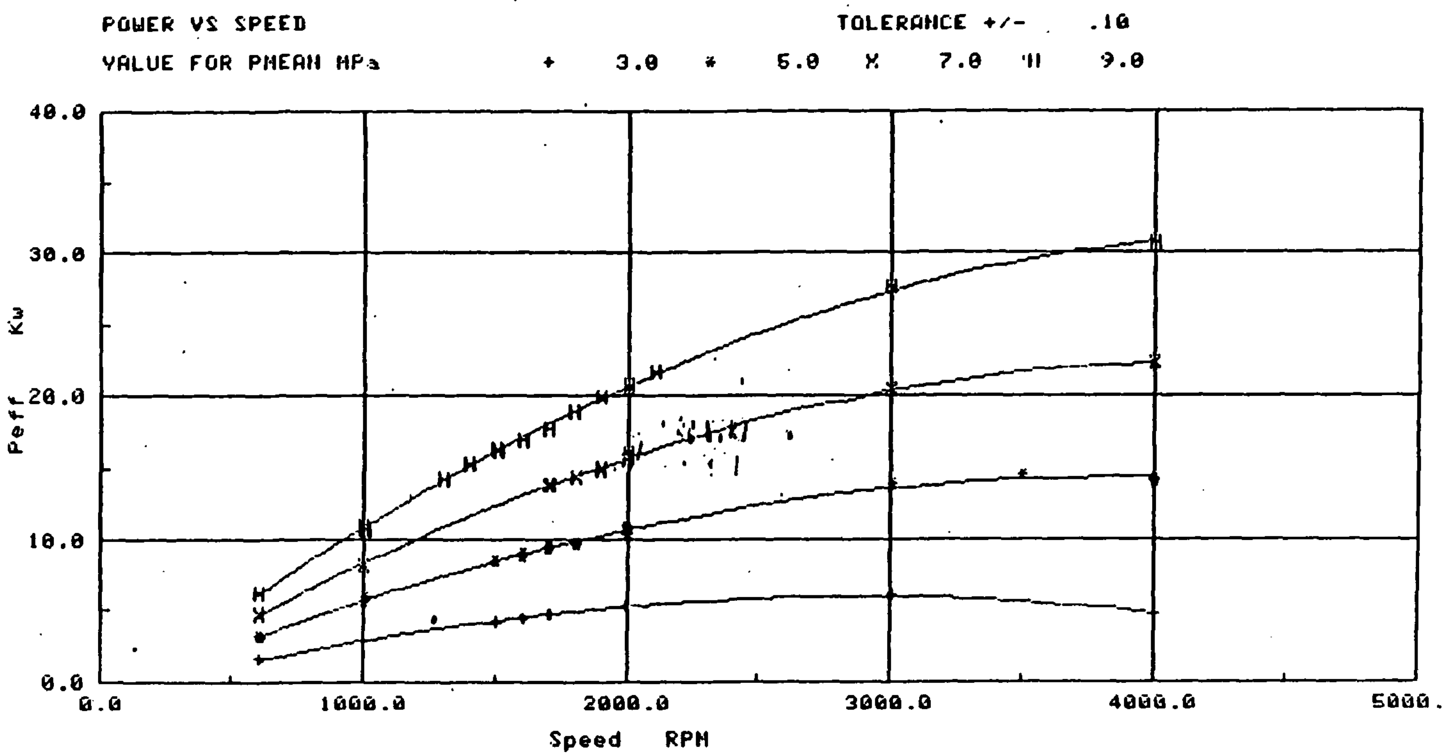


Fig. 4-10 Upgraded Mod I Engine No. 5 Acceptance Test Measured Data (720°C Tube Temperature) Welded Heads, No Atomizing Air Compressor

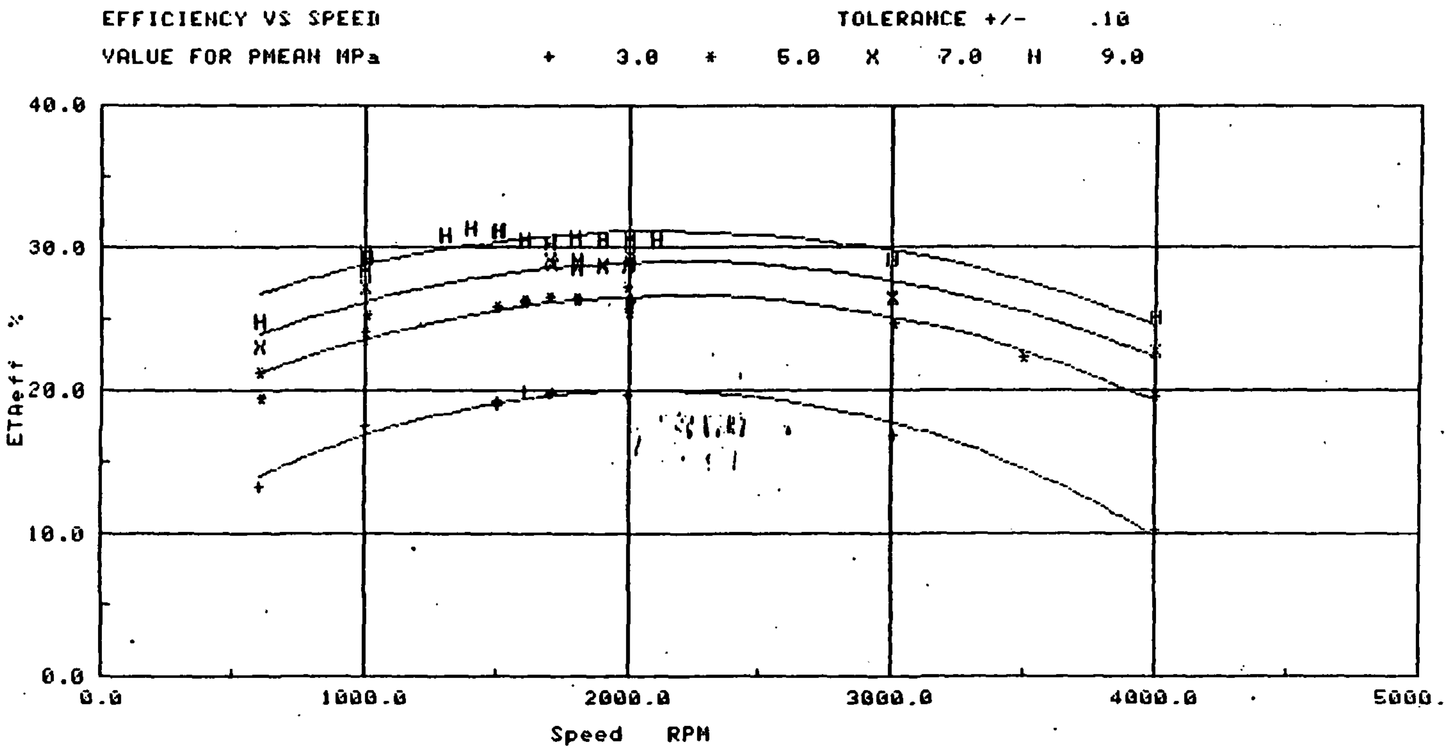


Fig. 4-11 Upgraded Mod I Engine No. 5 Acceptance Test Measured Data (720°C Tube Temperature) Welded Heads, No Atomizing Air Compressor

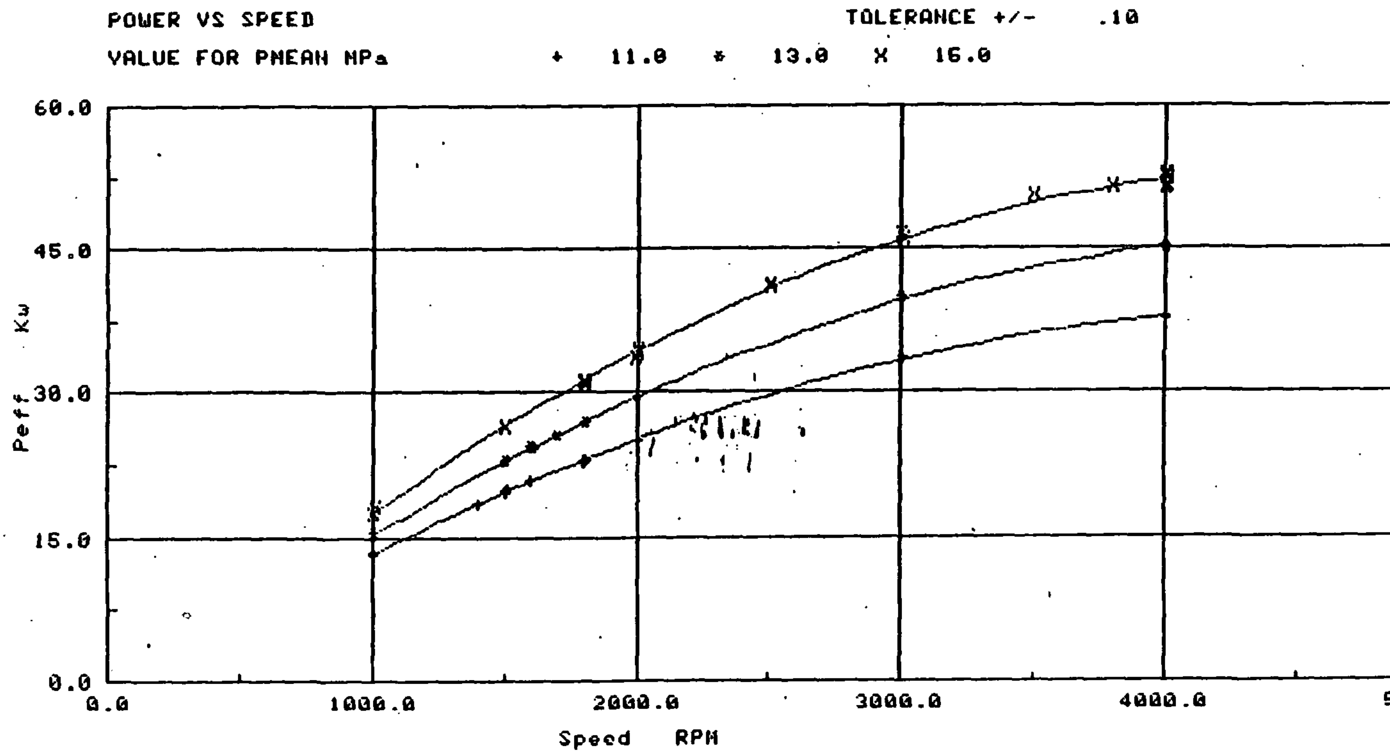


Fig. 4-12 Upgraded Mod I Engine No. 5 Acceptance Test Measured Data (720°C Tube Temperature) Welded Heads, No Atomizing Air Compressor

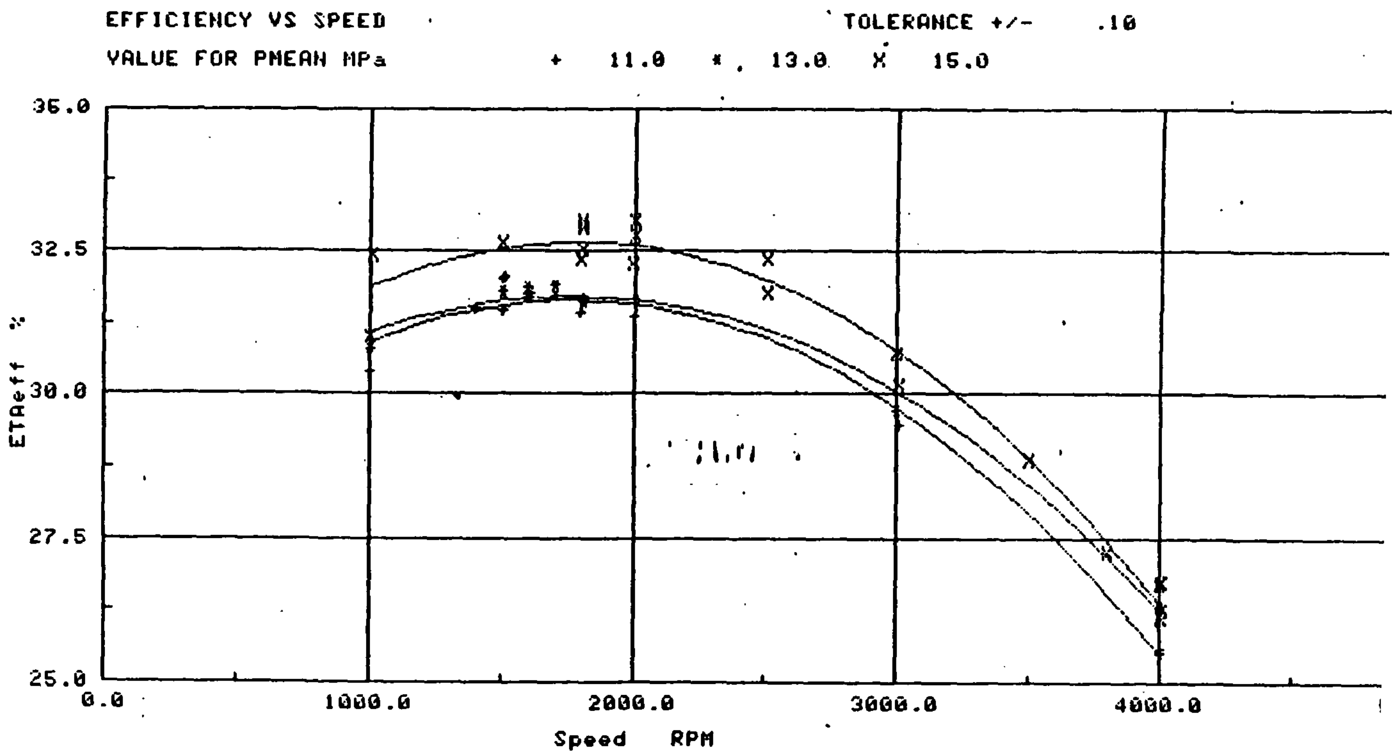


Fig. 4-13 Upgraded Mod I Engine No. 5 Acceptance Test Measured Data (720°C Tube Temperature) Welded Heads, No Atomizing Air Compressor

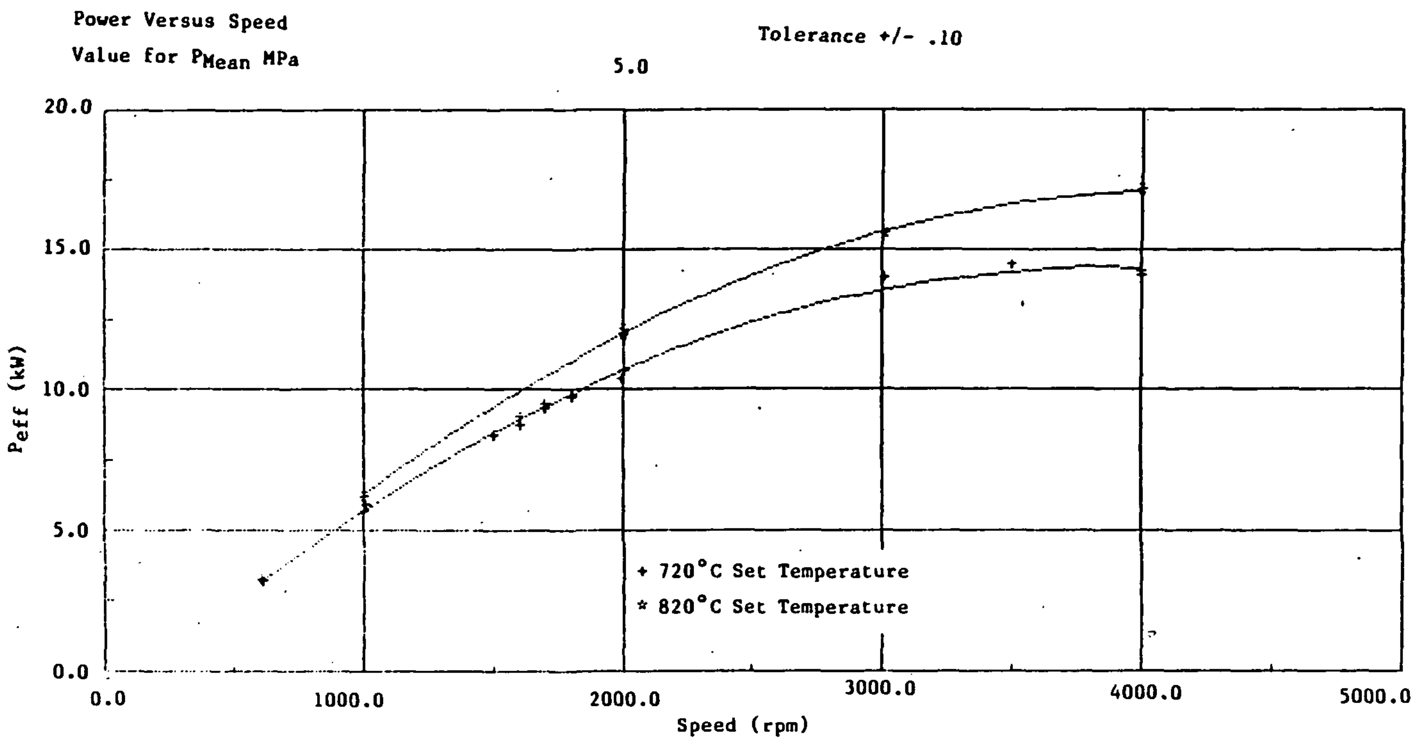


Fig. 4-14 Upgraded Mod I Engine No. 5 Welded Heads - Power Vs. Speed at 5 MPa

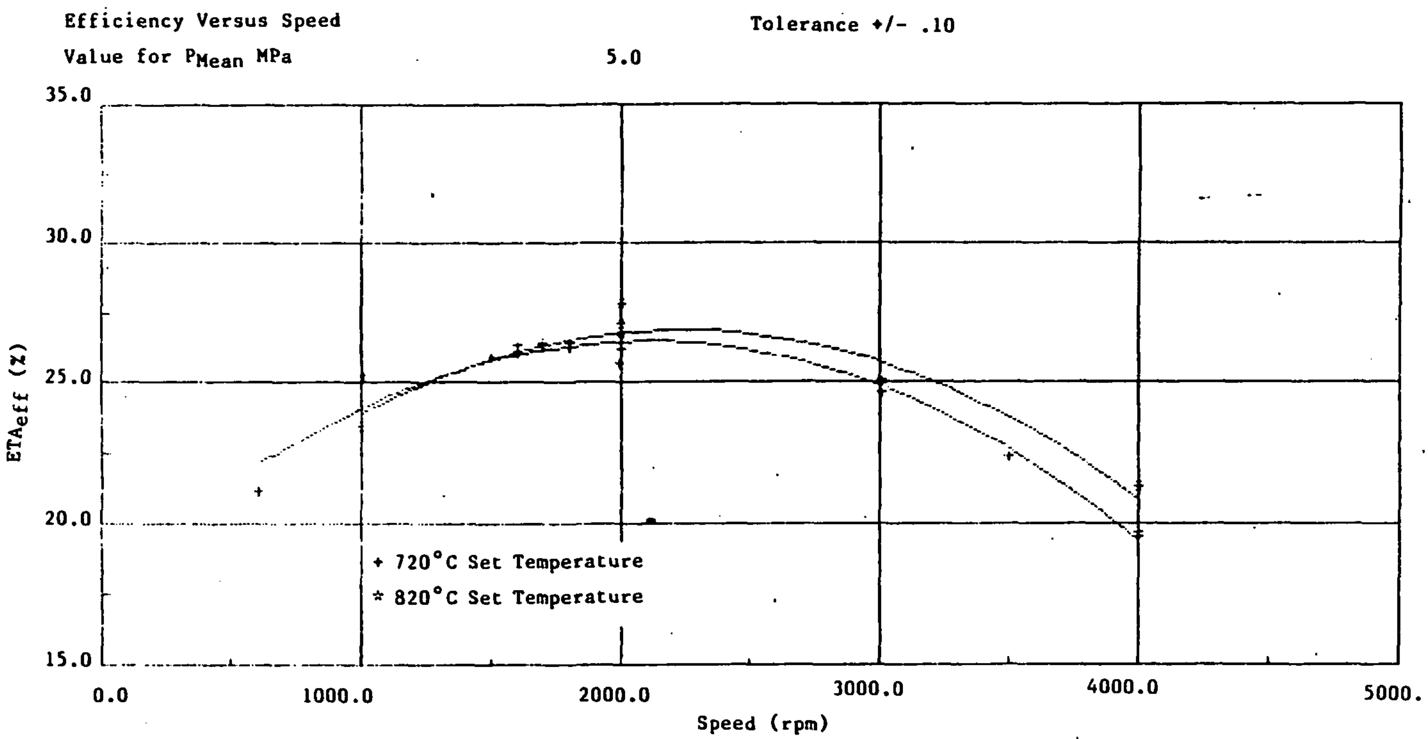


Fig. 4-15 Upgraded Mod I Engine No. 5 Welded Heads - Efficiency Vs. Speed at 5 MPa

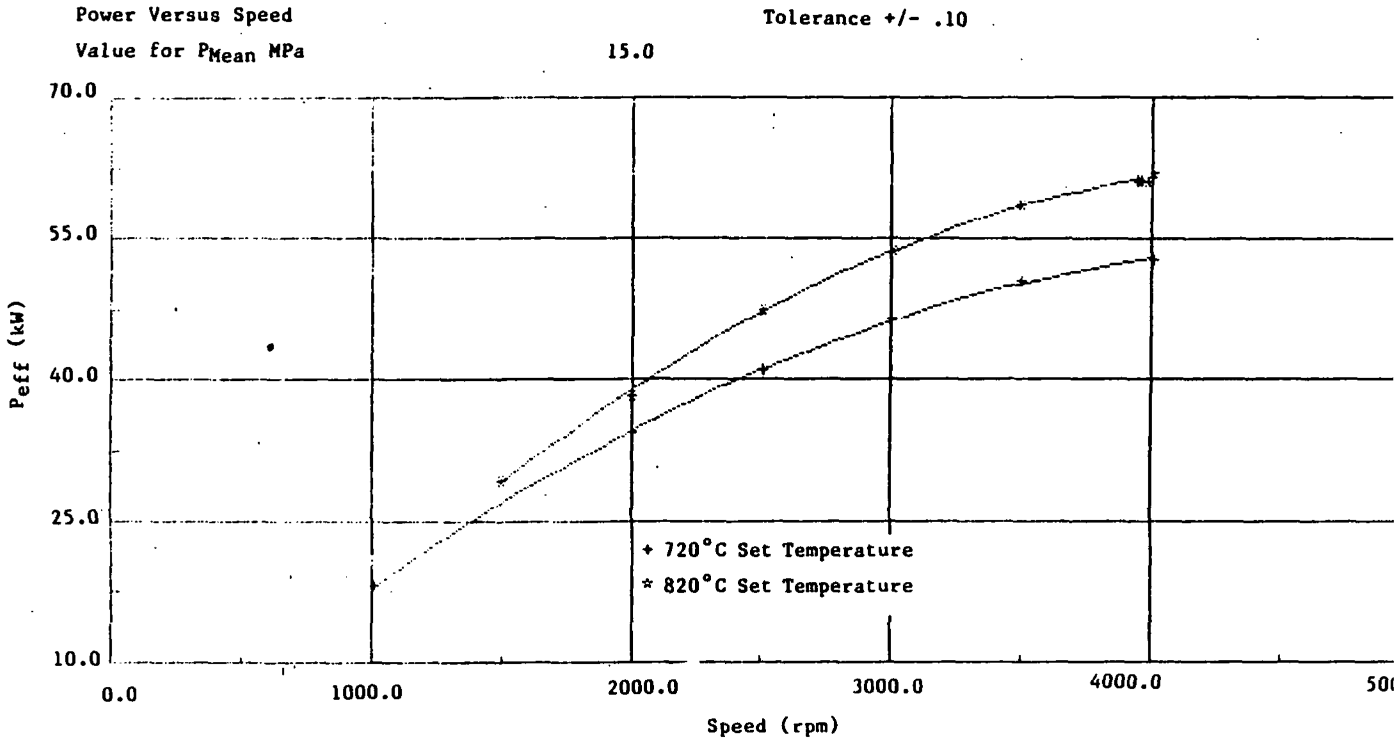


Fig. 4-16 Upgraded Mod I Engine No. 5 Welded Heads - Power Vs. Speed at 15 MPa

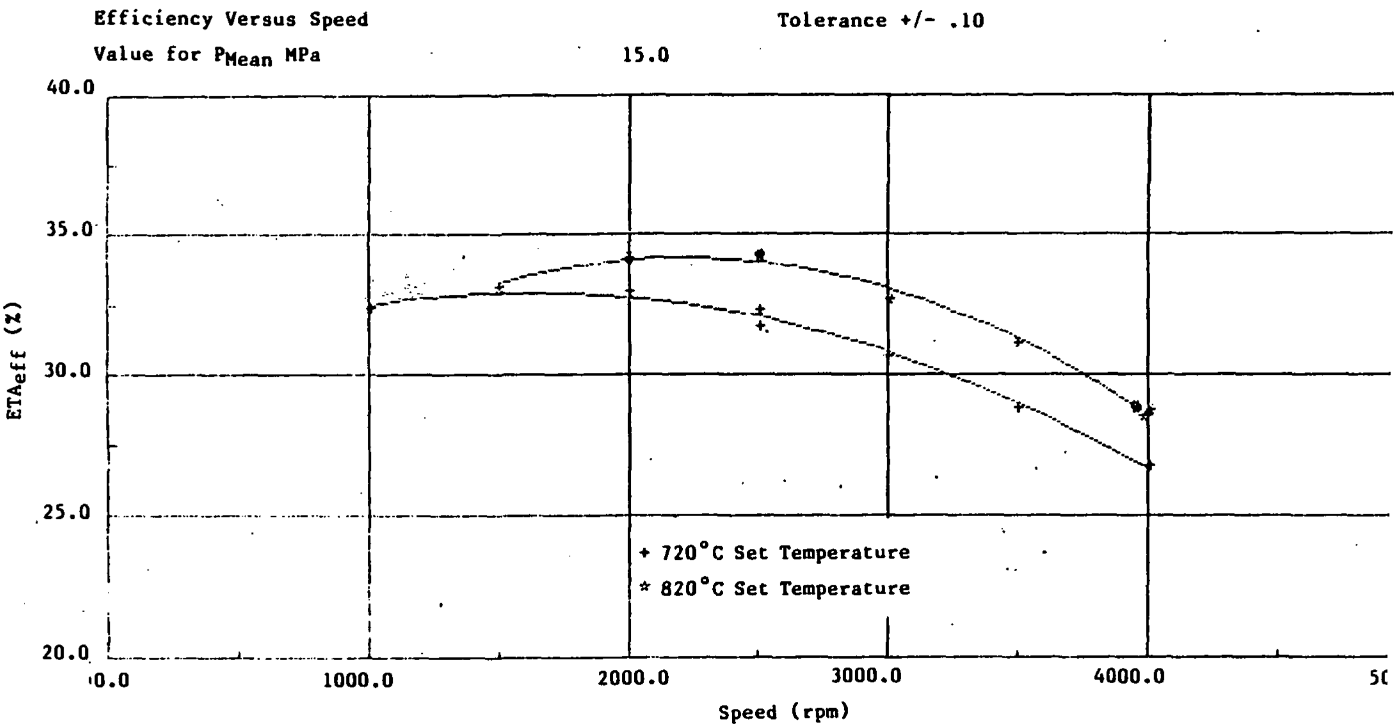


Fig. 4-17 Upgraded Mod I Engine No. 5 Welded Heads - Efficiency Vs. Speed at 15 MPa

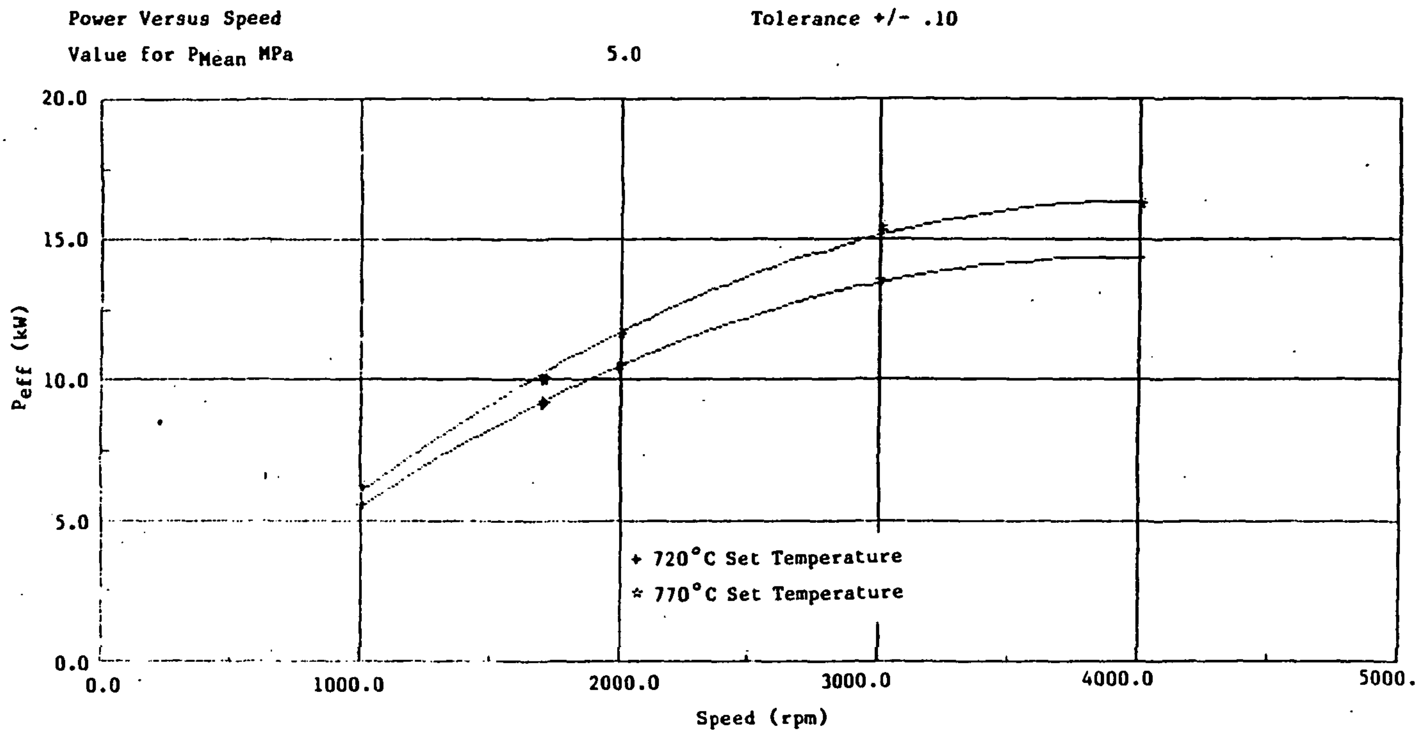


Fig. 4-18 Upgraded Mod I Engine No. 5 Hybrid Heads - Power Vs. Speed at 5 MPa

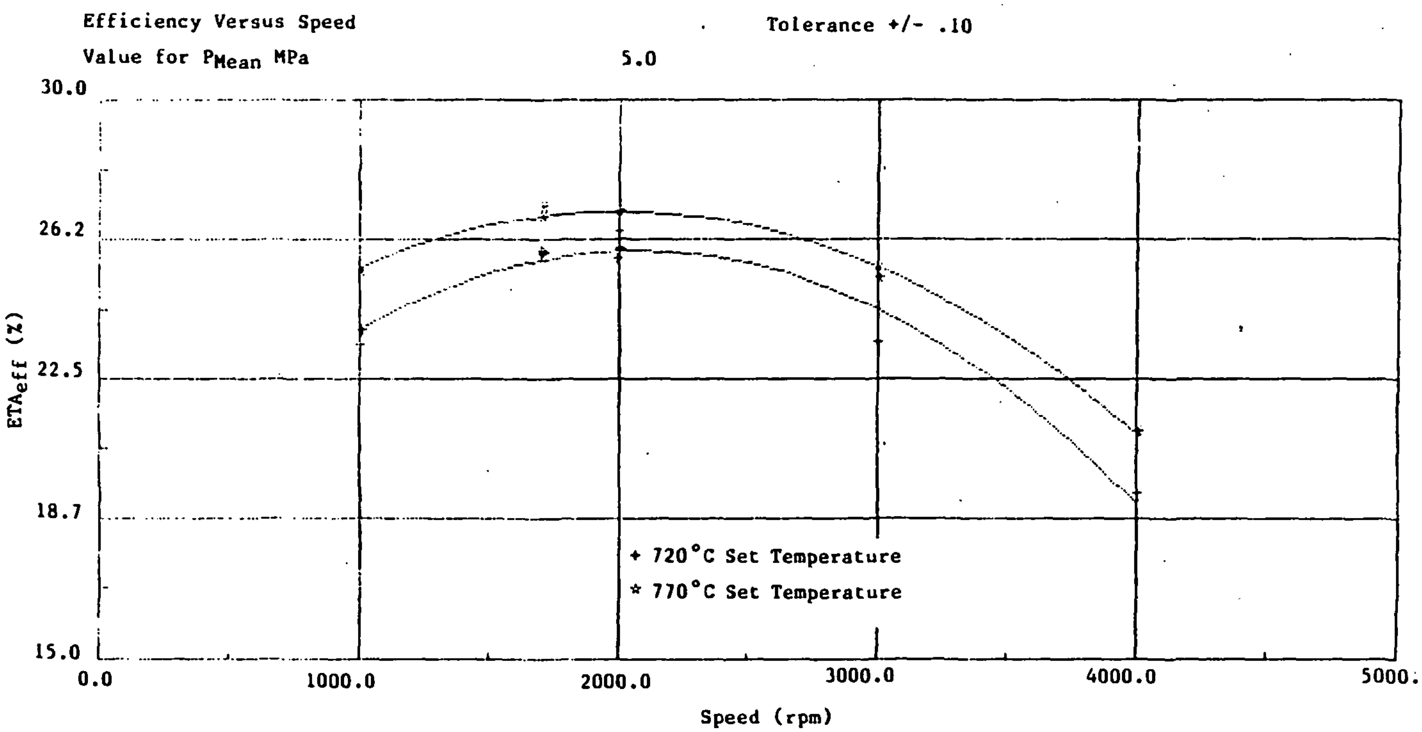


Fig. 4-19 Upgraded Mod I Engine No. 5 Hybrid Heads - Efficiency Vs. Speed at 5 MPa

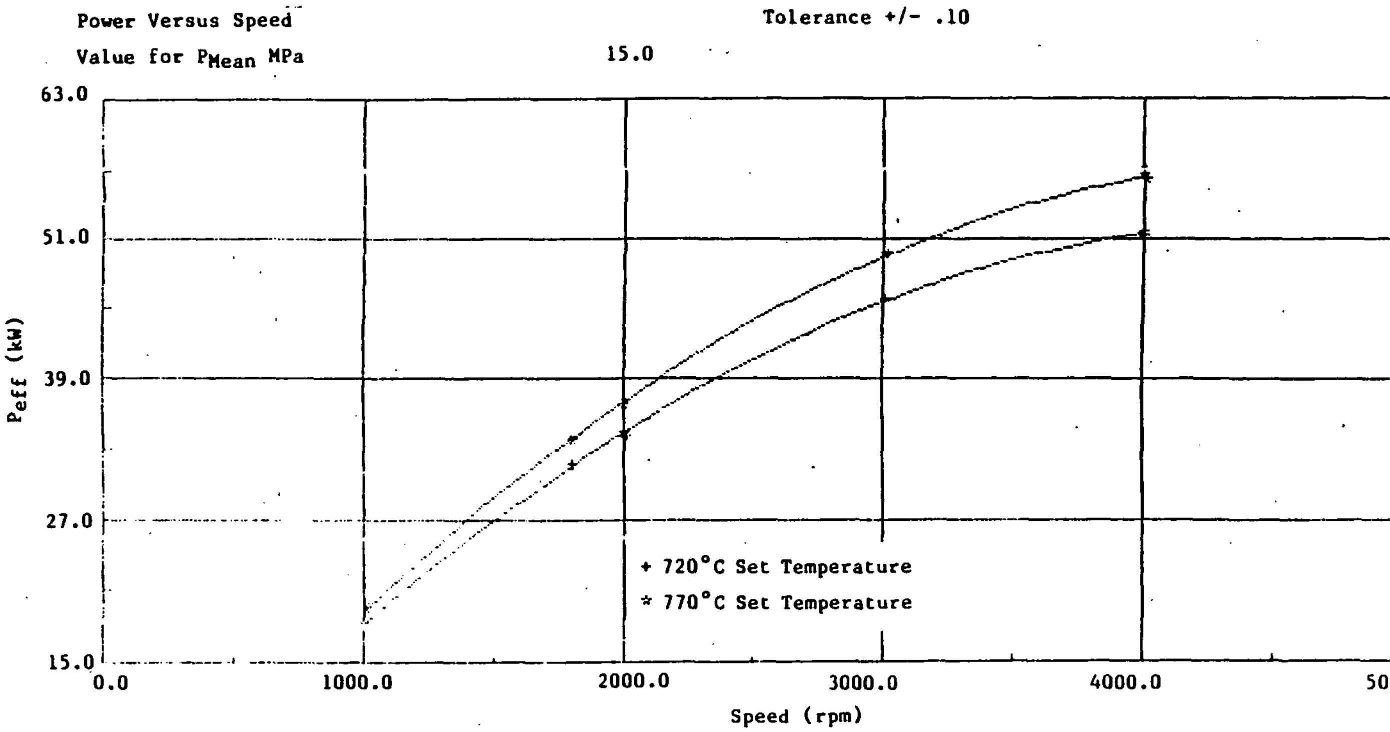


Fig. 4-20 Upgraded Mod I Engine No. 5 Hybrid Heads - Power Vs. Speed at 15 MPa

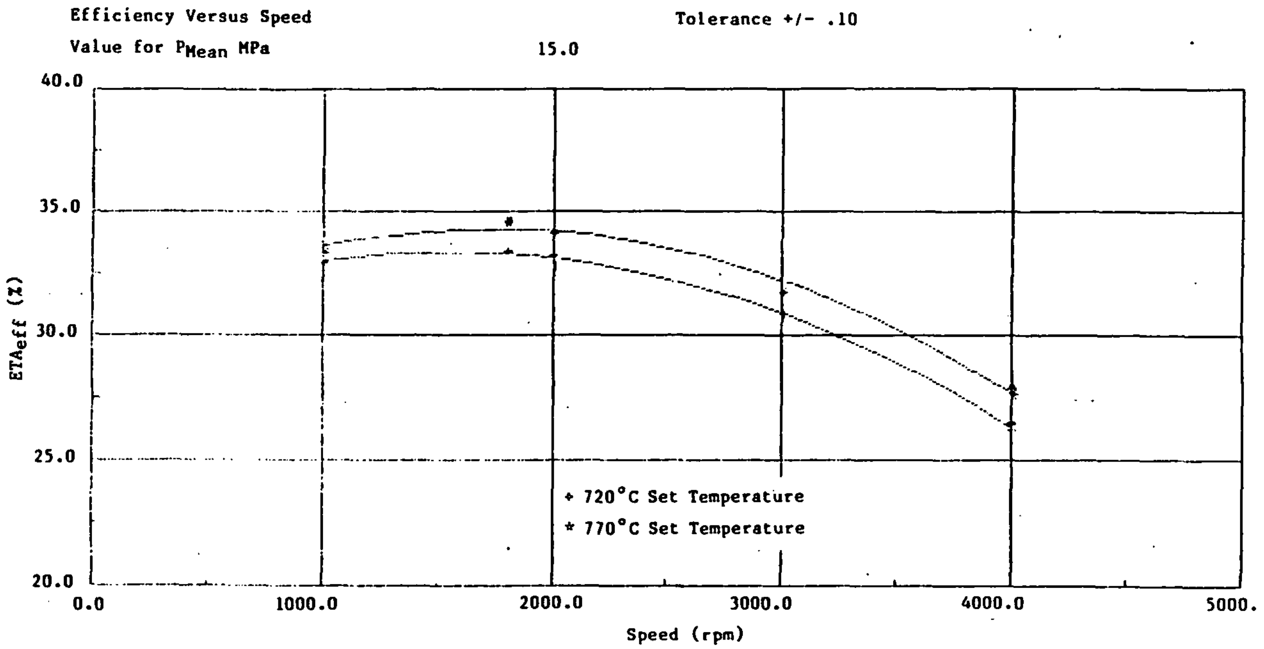


Fig. 4-21 Upgraded Mod I Engine No. 5 Hybrid Heads - Efficiency Vs. Speed at 15 MPa

Upgraded Mod I Engine No. 6

Upgraded engine No. 6 was engaged in extensive BSE development/acceptance testing during the latter part of 1983. Major accomplishments included:

- a demonstration of improved performance due to a modification to the contours of the cylinder liner neck of the upgraded Mod I heater head;
- successful acceptance testing of the BSE configuration at 720°C, 770°C, and 820°C;
- evaluation testing of two CGR combustors: hairpin and tubular-to-elliptical designs; and,
- investigative testing of the BSE to confirm design estimates for parameters affecting overall efficiency levels.

The engine was involved in 140 hours of testing, and collected ~300 data points (Figure 4-22). At the end of December, the engine was active in completing the performance testing outlined in MTI Report No. 84ASE364TP61.

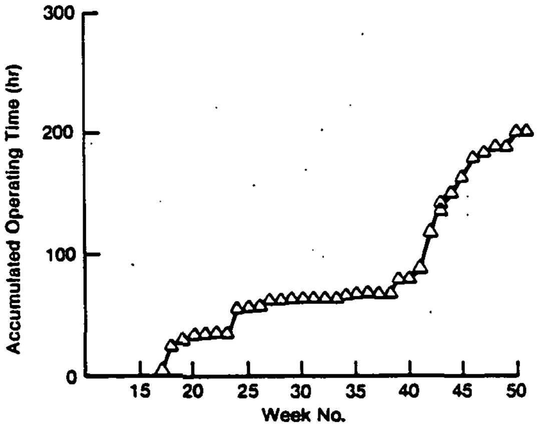


Fig. 4-22 Upgraded Mod I Engine No. 6 Performance Test - Accumulated Operating Time Through 12/31/83

IMPROVED CYLINDER HOUSING FLOW

Efforts at the beginning of this report period concentrated on eliminating the detrimental effect of nonaxial working gas flow in the cylinder housing. This problem was created by the revised neck contours of the upgraded Mod I housing (see Figure 4-23). Several attempts were made to modify the contours for engine testing, first on the Cold Flow Rig, and then using weld material. The Cold Flow Rig tests indicated that significant improvements in uniform flow distribution were possible (see Figure 4-24).

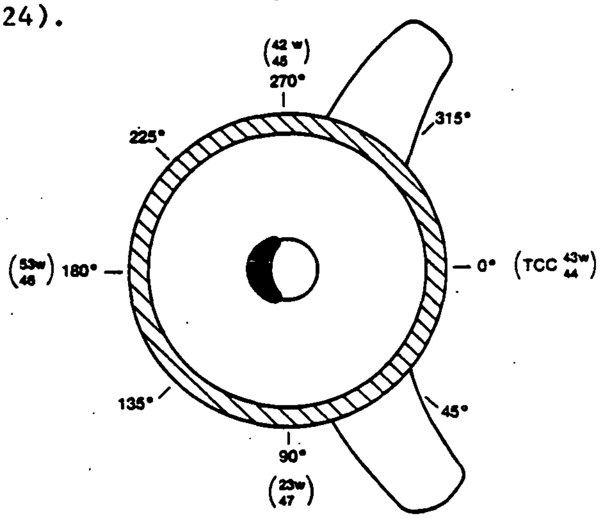


Fig. 4-23 Flow Distribution Test - Engine Test Welded Stuffer

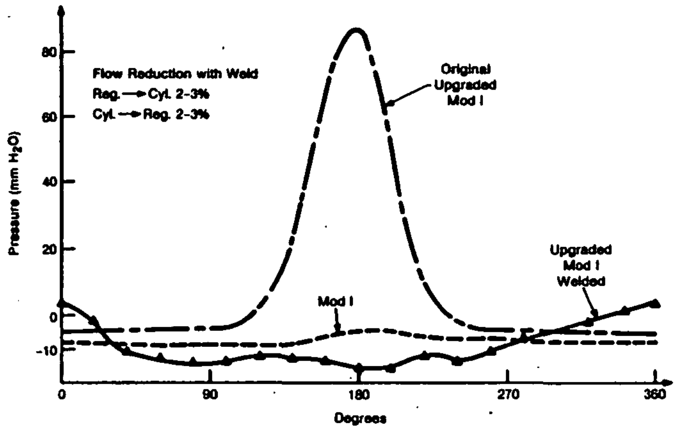


Fig. 4-24 Upgraded Mod I Engine No. 6 Cylinder Housing Flow Distribution/Temperature Tests

The upgraded heater heads were modified and tested, strongly confirming that the problem had been adequately addressed (see Figure 4-25). The heads were then tested at MTI before their return to USAB in August, 1983.

The weld modification was used to improve existing and in-process cylinder castings. Later upgraded Mod I heater heads used Mod I cylinder castings (the so-called "hybrid" heads) to provide a more uniform flow improvement device.

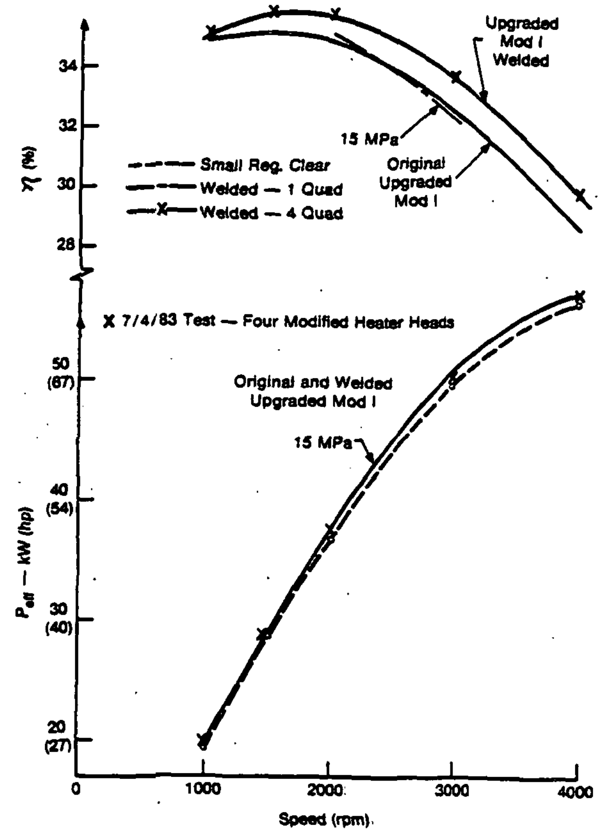


Fig. 4-25 Test Results of Modified Upgraded Heater Heads

PREHEATER AIRFLOW DISTRIBUTION

A test was conducted in late September to further investigate this problem, which contributes to higher than desired heater head temperature spreads. Different circumferential locations for the preheater housing exhaust outlets were attempted in order to improve flow uniformity. The preheater was rotated 180° to also ascertain its effect on the pattern. The tests revealed that the temperature spreads were not sensitive to these changes, thus focusing interest on the internal flow within the preheater itself. Engine testing then continued with the standard preheater housing.

COMBUSTOR DEVELOPMENT

Two CGR combustors (hairpin and tubular-to-elliptical designs) have been emissions tested on upgraded Mod I engine No. 6. A full discussion of test results can be found in Section III, Upgraded Mod I Combustor Development.

BSE ACCEPTANCE TEST

Upgraded Mod I engine No. 6 performed its acceptance test in October and November, 1983 in accordance with MTI Report No. 83ASE311TP49 (Revision 1), "Acceptance Test of the Upgraded Mod I Stirling Engine System." Performance data was obtained at the required 720°C and 820°C operating levels, as well as at an interim temperature setting of 770°C. Performance results were generally at or near the minimum acceptable values given in the revised test plan, although measurably different than levels promised from prior predictions (see Table 4-5).

Table 4-5
Upgraded Mod I Engine No. 6 BSE Acceptance Test Performance Results

BSE at 720°C

Condition		Prediction (115-12)		Minimum Accept.		Actual	
Speed k-rpm	Press MPa	kW	hp	kW	hp	kW	hp
1	15	20.22	27.11	19	25.5	18.85	25.28
2	15	38.26	51.30	37	49.6	37.00	49.6
3	15	50.57	67.81	51	68.4	50.7	67.99
4	15	54.99	73.74	58	77.8	58.3	78.18
Max η	15	39.04% (1.5 K)		35% (2.0 K)		35.4%	
2	5	35.10%		31%		31.0%	

BSE at 820°C

Condition		Prediction (114-09)		Minimum Accept.		Actual	
Speed k-rpm	Press MPa	kW	hp	kW	hp	kW	hp
1	15	22.65	30.37	21	28.2	21.4	28.70
2	15	43.46	58.28	43	57.7	42.0	56.32
3	15	58.96	79.06	59	79.1	58.7	78.72
4	15	67.08	89.95	68	91.2	69.8	93.60
Max η	15	41.52% (1.5 K)		37%		35.4% (2 K)	
2	5	37.16%		31%		31.0%	

BSE at 770°C

Condition		Actual	
Speed k-rpm	Press. MPa	kW	hp
1	15	20.24	27.14
2	15	39.59	53.09
3	15	54.77	73.45
4	15	63.91	85.70
Max η	15	36.56%	(2K)
2	5	31.28%	(2K)

Comparisons between predicted power and efficiency levels are shown in Figures 4-26 and 4-27, respectively. Figure 4-26 shows that the power prediction is generally within ±1.5 kW (±2.0 hp) at all pressure/speed combinations, except at maximum power (15 MPa/4000 rpm) where the prediction underpredicts power by ~3 kW (~4 hp).

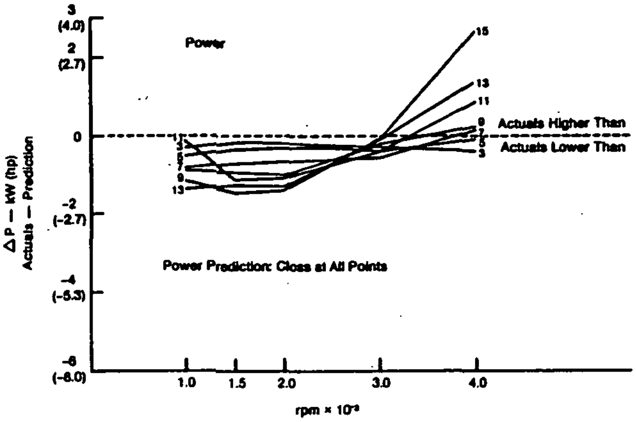


Fig. 4-26 Upgraded Mod I Engine No. 6 Acceptance Test - Predicted Power Levels

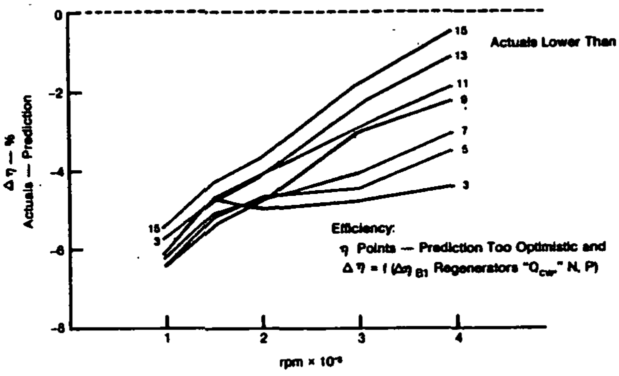


Fig. 4-27 Upgraded Mod I Engine No. 6 Acceptance Test - Predicted Efficiency Levels

The efficiency story given in Figure 4-27 indicates a significantly lower speed deficit of ~6 points. The data fits the prediction much better at the maximum power point. The overall differences identified the need for a post-acceptance test development test plan.

Another finding during high-temperature operation of the engine was the limitations of the currently used fin-to-tube braze material, i.e., while the braze was adequate at 720°C, it would start to degrade when operating temperatures were moderately raised (to 770°C). The effects of this finding prompted an effort to select a more durable braze material for this operating range (see the Materials and Process Development Section of this report).

At the end of this report period, a new braze alloy had been selected. Procurement of new upgraded Mod I heater heads for testing in the first half of 1984 is well underway.

POST-ACCEPTANCE DEVELOPMENT TESTING

As a result of the findings of the acceptance test, a short-term development program was activated in December*, the purpose of which was to locate and eliminate the sources of lower-than-expected efficiency levels within the EHS BSE system. A systematic approach was taken to improve EHS and heater head performance, to be followed by an investigation of actual operating temperatures in both the hot (heater head) and cold (cold connecting duct) locations, and to quantify heat-conduction losses in the working gas system (regenerator casting wall conduction).

Testing through December indicated that simple, direct solutions to the observed problems (such as excessive leakage flow in the combustor heater head area that was affecting EHS efficiency) were not to be found. Testing during the first half of 1984 will determine if the actual hot and cold operating temperatures need to be adjusted to improve overall efficiency levels.

Mod I Engine No. 7

Mod I engine No. 7, active as a seals development engine, has accumulated a total of 596 operational hours to date (Figure 4-28). This work was an extension of the seals endurance testing conducted for the

*See MTI Report No. 84ASE364TP61

Mod I engine on P-40 engines No. 4 and 9; as a result, similar seals and endurance cycles from the P-40 testing were used. The opportunity of developing seals for a Mod I engine on an available Mod I engine where second-order, or long-term, factors would be in play was actively undertaken.

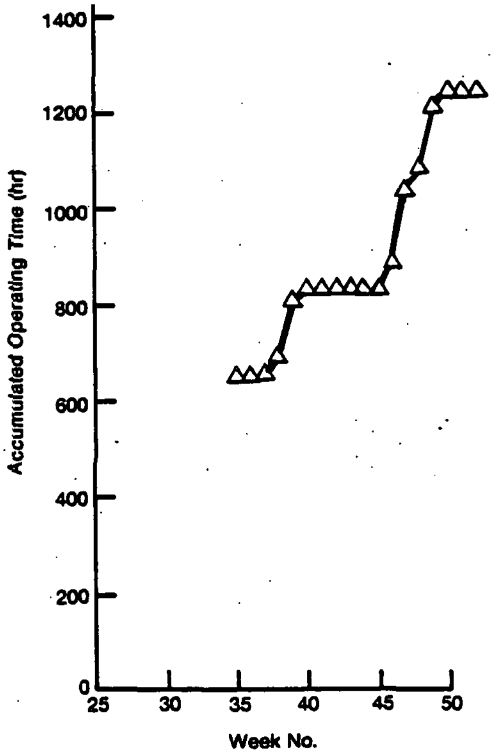


Fig. 4-28 Mod I Engine No. 7 Main Seal Test - Accumulated Operating Time Through 12/31/83

Mod I engine No. 7 was initially fitted with two units each of the HABIA and Rulon J material PL seals, and a modified seal house venting system that separated leakage from the two pairs of seals.

Accelerated duty cycle testing that began in September, 1983 was stopped after 180.15 hours when a crack was discovered in the crankcase housing. Because of the necessity of a major rebuild, and the disturbance to the seal houses, this seals test was considered terminated. A new crankcase was supplied from MTI's ASE stores.

Data from dimensional comparisons between the P-40 and Mod I engines revealed that the crosshead guide clearance differences between the two designs may be contributing to the shorter life of the PL seals on the Mod I engines (shown in Figure 4-29). The P-40 engine uses steel crossheads, which allow for smaller operating clearances with the piston rod slipper than is found on the Mod I engine, which uses aluminum crosshead guide bodies.

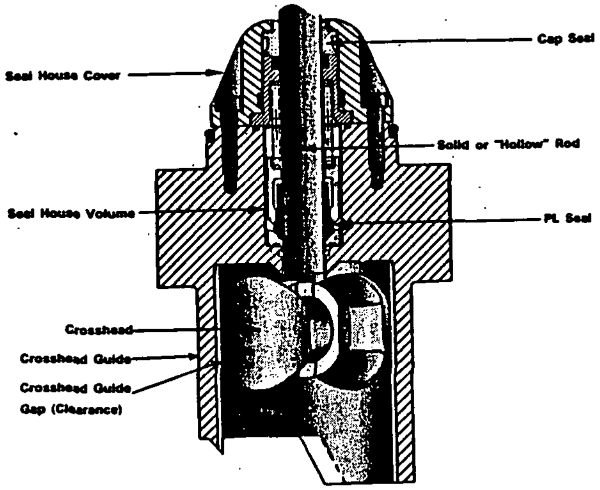


Fig. 4-29 Mod I Crosshead Casting

To explore this theory further, Mod I engine No. 7 was rebuilt with properly configured steel crosshead guides with gaps ranging from 40 to 51 μ, two Rulon J seals and two HABIA PL seals, and a full four-filter seal house venting system (shown in Figure 4-30).

Endurance testing with this configuration began in November and continued through the end of December. Late in December Cycle #4's seal house filter suffered a marked increase in oil capture weight, so the test was stopped for inspection of this area. The seal house was found to be contaminated with oil (failed Rulon J PL seal), with the seal seat (cone piece) condition being a contributing factor. The seat and seal were replaced. The other seal houses were inspected and found to be acceptable, although some changes were made to the EHS to improve durability in this area.

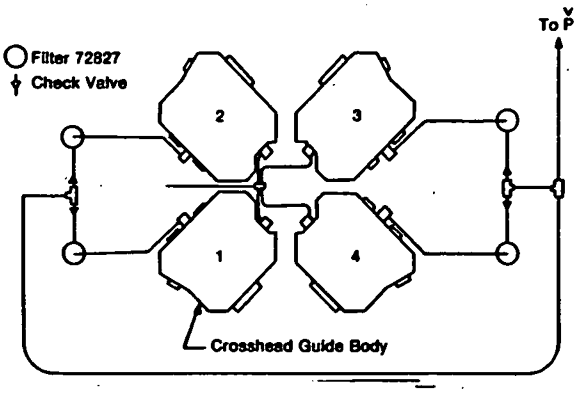


Fig. 4-30 Four-Filter System

The engine was prepared at the end of December for continued seals testing with the accelerated P-40 test cycle, which was to be used until the remaining three seals failed. Component times as of January 1, 1984 are 515.54 hours on all piston rings and cycles 1, 2, and 3 PL seals. Only 3.85 hours were accumulated on the newly installed replacement seal in cylinder #4.

Upgraded Mod I Engine No. 8

Upgraded engine No. 8 has been built for eventual installation in a Spirit vehicle for driving cycle testing and evaluation at General Motors. Initial engine calibrations were obtained during this report period. The data is shown in Figures 4-31 and 4-32 compared to upgraded Mod I engine No. 5. Both power and efficiency are better for upgraded engine No. 8 for the same operating configuration.

Upgraded Mod I Engine Performance Analysis

Development testing of the upgraded Mod I engine continued during the last half of 1983, with emphasis placed on understanding the discrepancies between measured performance and analytical projections. Several tests were performed, and are detailed below.

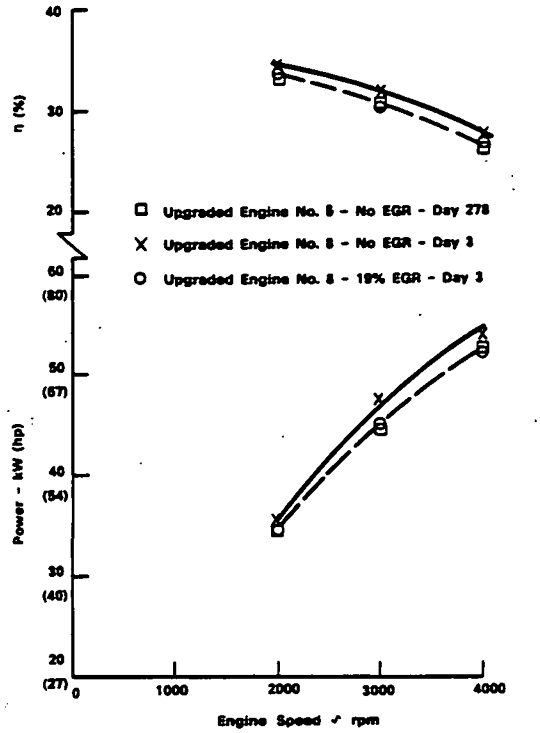


Fig. 4-31 Upgraded Mod I Performance (T_{set} = 720°C/15 MPa) - Power/η Versus Engine Speed

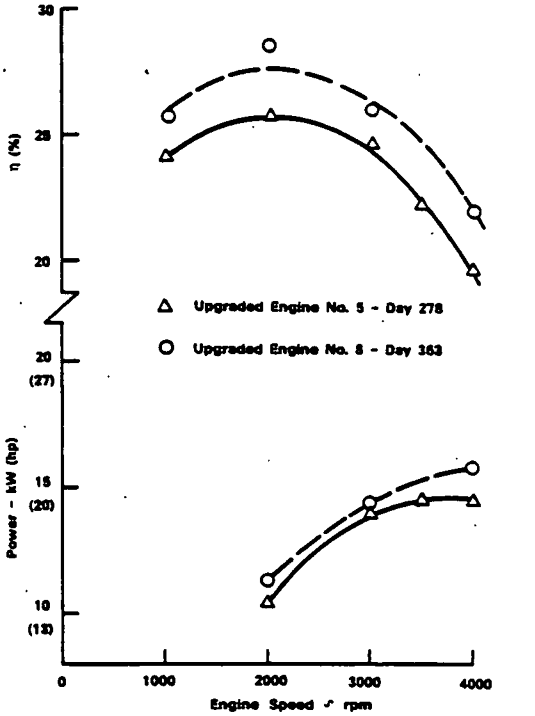


Fig. 4-32 Upgraded Mod I Engines No. 5 & 8 Performance (T_{set} = 720°C/5 MPa) - Power/η Versus Engine Speed

HYBRID HEATER HEAD TESTS

Review of the upgraded Mod I design indicated that the cylinder housing could be a cause of the reduced performance. The heater head material (XF-818) has a higher coefficient of thermal expansion than the Inconel 718 piston dome material.

Under hot operating conditions, the appendix gap (clearance between the dome and cylinder housing wall) is larger than in a Mod I engine, and it is anticipated that the losses associated with the appendix would then be higher than that predicted by the code.

In addition to this problem, the different neck geometry from the manifold of the housing to the housing itself was suspected of causing a nonuniform flow distribution within the cylinder housing (Figure 4-33). This was confirmed by localized melting of the cylinder liner O-ring and backup ring, and by extensive thermocouple installation on the housing (Figure 4-34). This nonuniform flow entry, with resultant nonuniform cylinder wall heating, can be expected to cause further performance degradation.

Two approaches were taken to correct this problem. In order to salvage existing hardware, weld material was added to the neck to straighten out the flow. Tests of two heater heads modified in this manner produced drastically different results (shown in Table 4-6).

Since the weld material addition was done by hand, it is not surprising that the results were quite different. A second approach taken was to assemble a "hybrid" heater head using an upgraded Mod I regenerator housing and a Mod I cylinder housing. A back-to-back test comparing the original to the upgraded Mod I heater heads produced substantial improvements (Figure 4-35).

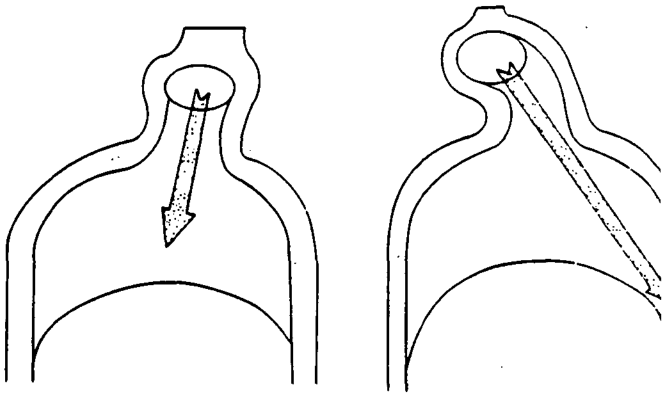


Fig. 4-33 Cylinder Housing with Piston at Bottom Dead Center: Gas Flow from Manifold

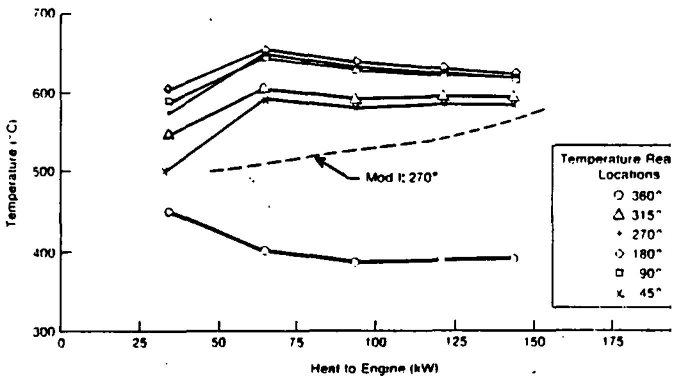


Fig. 4-34 Upgraded Mod I Cylinder Housing Casting Temperatures

Table 4-6
Performance Change After Heater Head Weld Modification

	Max Power		Max Efficiency
	kW	(hp)	% Points
Engine No. 5	+2.5	(+3.4)	+4.3
Engine No. 6	-1.0	(-1.3)	-0.3

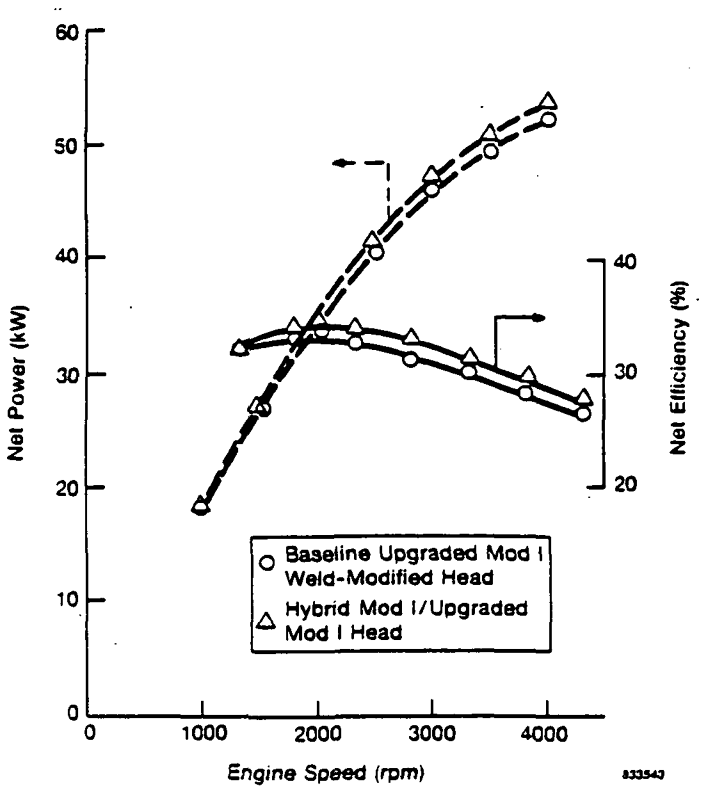


Fig. 4-35 Performance Improvement Measured With Hybrid Mod I/Upgraded Mod I Heater Heads

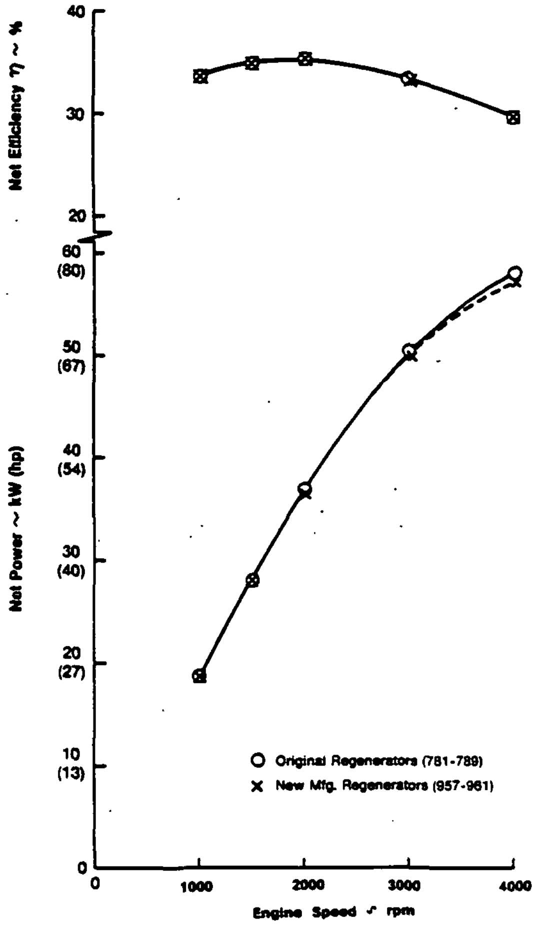


Fig. 4-36 Upgraded Mod I Engine No. 6 Net Power/Efficiency Versus Engine Speed (15 MPa/720°C Tset)

REMANUFACTURED REGENERATOR TESTS

The regenerators used for early upgraded Mod I testing were destructively examined and found to be incorrectly manufactured. The dome-shaped regenerator is manufactured correctly by stacking decreasing diameter screens to achieve the dome shape. The original regenerators were formed by stacking constant-diameter discs, and then forming the dome shape by a heavier press on the OD portion of the regenerator, which would create a nearly solid material mass at the OD. In addition, the braze between the regenerator screens and shell was found to be inadequate, resulting in the screen matrix being loose inside the regenerator, and creating the possibility of bypass leakage between the regenerator matrix and shell. A set of correctly manufactured regenerators were tested back-to-back with the original type (Figure 4-36). As can be seen, no significant performance difference was noted at either 5 or 15 MPa.

820°C SET TEMPERATURE TESTING

Initial testing of the upgraded Mod I was conducted at 720°C rear-row set temperature as an interim step in the development process. Testing was conducted on upgraded engines No. 5 (SES) and 6 (BSE) to evaluate the performance impact of the higher set temperature. Similar results were obtained for both engines (Figure 4-37). The data is presented in the form of differences in power and efficiency in Figure 4-38). Upgraded engine No. 6 showed the greater increase in power, which was anticipated since the increased blower power required for 820°C operation slightly decreases the power improvement in the full SES configuration of upgraded engine No. 5.

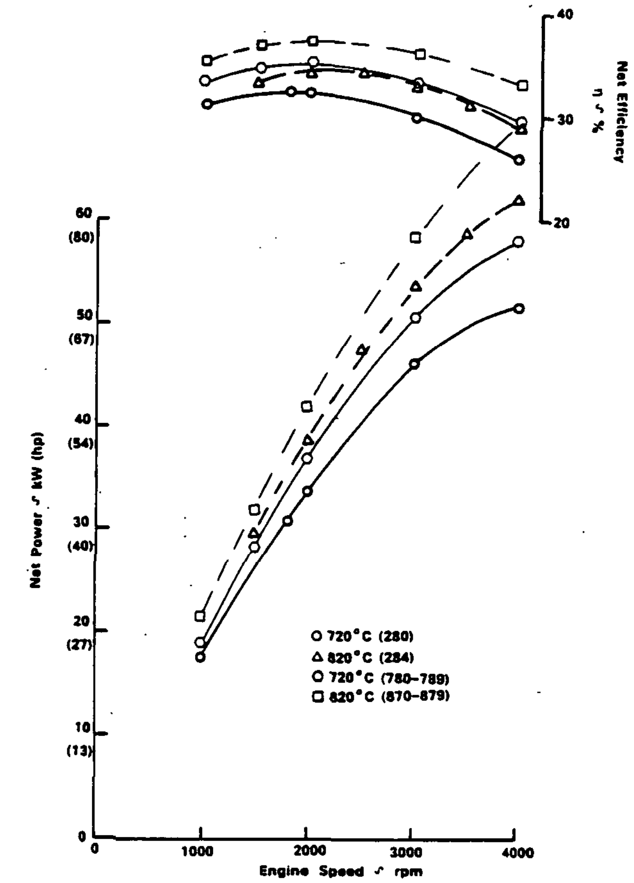


Fig. 4-37 Upgraded Mod I Engines No. 5 & 6 Net Power/Efficiency Versus Engine Speed

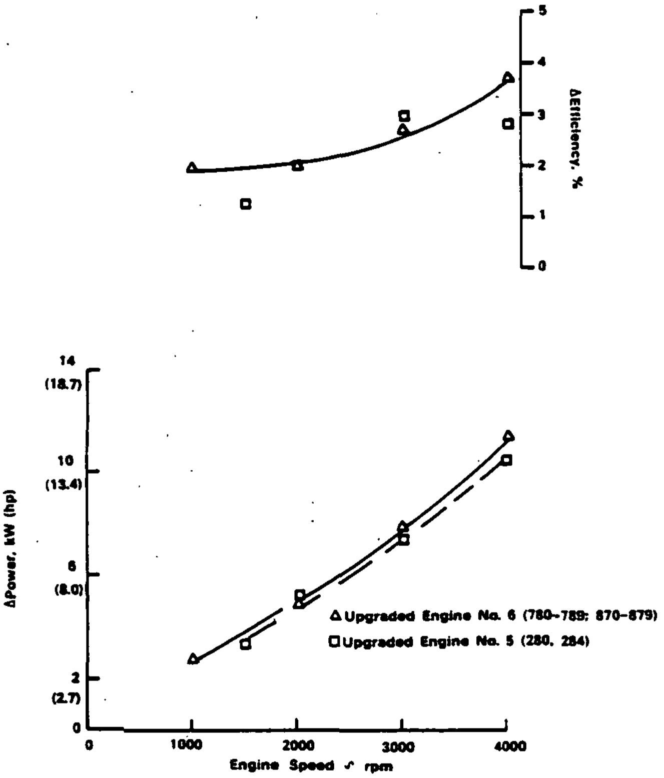


Fig. 4-38 Upgraded Mod I Engine Testing - Effect of Increased Set Temperature 720–820°C

ROLLING-ELEMENT DRIVE UNIT TESTS

Back-to-back tests were conducted with the journal and rolling-element drive units to evaluate the difference in friction losses between the two units. Figure 4-39 and Table 4-7 present the results of this test. As can be seen, the measured difference in performance is very close to that predicted.

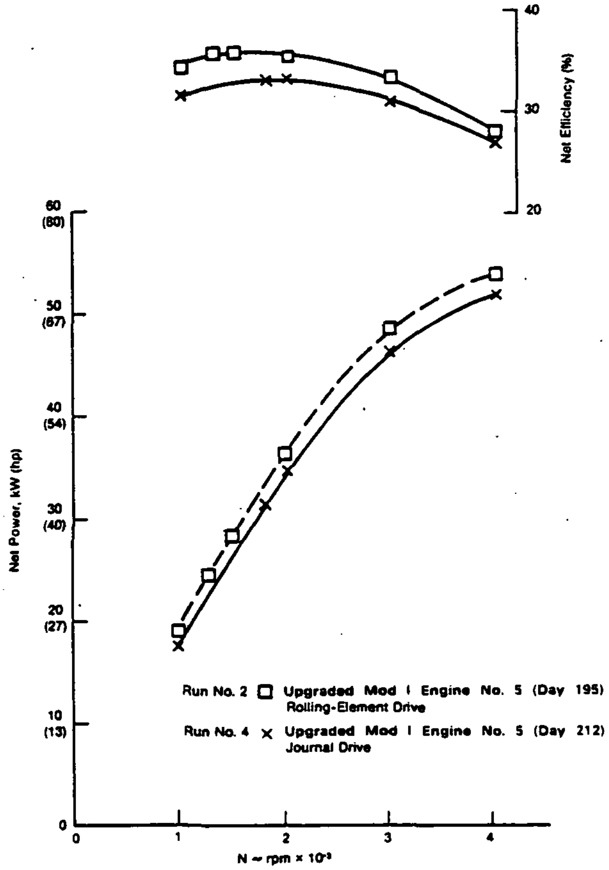


Fig. 4-39 Upgraded Mod I Engine No. 5 Performance - Effect of Rolling-Element Drive (15 MPa/720°C Set Temperature)

Table 4-7
Rolling-Element Drive Unit Test Results

	Max Power Point		Max Efficiency Point
	kW	(hp)	(%)
Predicted	2	(2.7)	1.5
Measured	1.8	(2.4)	2.4

WORKING GAS TEMPERATURE TESTS

Due to the flow distribution in the upgraded Mod I heater head, it has been suspected that working gas temperature levels for the upgraded engine may differ from the Mod I even though both are operated at the same rear-row set temperature. The BOM upgraded Mod I heater head did not include working gas temperature measurement, so instrumentation was added to the regenerator housing manifold. Similar instrumentation will be added to the Mod I, which currently measures working gas temperature in a different location. Preliminary testing with the upgraded Mod I compared to the standard Mod I temperature indicates a substantially lower working gas temperature may in fact exist with the upgraded Mod I (Figure 4-40). A test of the Mod I with the regenerator manifold working gas temperature location is required to confirm this difference.

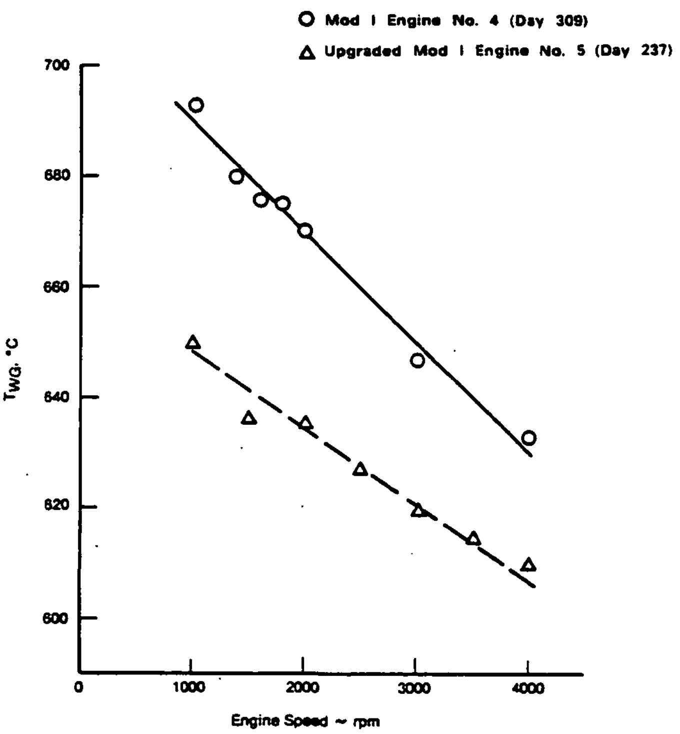


Fig. 4-40 Mod I Engine No. 4/Upgraded Mod I Engine No. 5 Performance (15 MPa/720°C Set Temp.)

FINAL ACCEPTANCE TESTS ON UPGRADED MOD I ENGINES NO. 5 AND 6

Upgraded Mod I engines No. 5 and 6 underwent final acceptance testing during the last half of 1983 as an SES and BSE, respectively. The test results for upgraded Mod I engine No. 5 compared to the final acceptance criteria are summarized in Table 4-8. The power and efficiency levels were below the minimum acceptable; however, all other criteria were acceptable. The engine was therefore recommended for conditional acceptance to permit development testing for correction of these problems.

Upgraded Mod I engine No. 6 underwent final acceptance testing at both 720°C and 820°C (see Table 4-9). Acceptable performance, except for some small deviations in power and efficiency, was achieved. All other criteria, with the exception of HC emissions repeatability, were met. The engine was subsequently recommended for acceptance.

Table 4-8
Upgraded Mod I Engine No. 5 Final Acceptance Test

Parameter	Condition (MPa/rpm)	Spec*	Actual**	Δ	ΔAdj***
Power (kW)	15/1000	18	18.2	+ .2	- .1
	15/2000	35	34.5	- .5	- .9
	15/3000	48	46.6	-1.4	-2.2
	15/4000	53	52.3	- .7	-1.8
Efficiency (%)	15/max η	33	33.1	+ .1	- .3
	5/2000	27	26.2	- .8	-1.5
Average Emissions (g/kg fuel)	EI_{HC}	<3.56	∿0		
	EI_{CO}	<29.6	1.12		
	EI_{NOx}	<8.69	7.17		

*See MTI Report No. 83ASE311TP49
**Average all data points.
***Power adjusted for atomizer air compressor.

Table 4-9
Upgraded Mod I Engine No. 6 Final Acceptance Test

Parameter	Condition (MPa/rpm)	720°C Spec	720°C Actual	720°C Δ	820°C Spec	820°C Actual	820°C Δ
Power (kW)	15/1000	19	18.8	- .2	21	21.5	+ .5
	15/2000	37	37	0	42	42	0
	15/3000	51	50.6	- .4	58	58.6	+ .6
	15/4000	58	58.2	+ .2	68	69.8	+1.8
Efficiency (%)	15/1500	35	34.9	- .1	38	37.2	- .8
	5/2000	31	30.7	- .3	33	32.3	- .7
Average Emissions (g/kg fuel)	EI_{HC}	<3.56	∿0		<3.56	∿0	
	EI_{CO}	<29.6	1.17		<29.6	.76	
	EI_{NOx}	<8.69	7.35		<8.69	10.31	

Preliminary Design and Analysis

The novel design concepts embodied in the RESD require extensive development before implementation in the Mod II engine. One of the high-risk areas in the design is the manufacturability of the annular heater heads and the V-Block. In order to evaluate the required technology advances, a preliminary design of several components was initiated during the last half of 1983. The procurement and testing of manufacturing samples of these components is planned for PY 1984. The status of these activities is described below.

Heater Head - The annular heater head casting design was completed, and drawings were released for vendor quotes.

V-Block - The casting drawing for this component has also been completed and released for quote.

Piston/Rod Joint - The piston/rod joint for the RESD utilizes a new manufacturing concept*. There has been some concern about the design life of this joint, so a test piece and fatigue test rig adaptor were designed to perform tests on the rod joint. Testing was initiated, and no failures have occurred to date.

Crankshaft Balancing - A computer-aided solids modelling program was utilized to design the crankshaft along with the balance weights.

The solids modelling was required due to the complicated shape of the crankshaft (Figure 5-1).

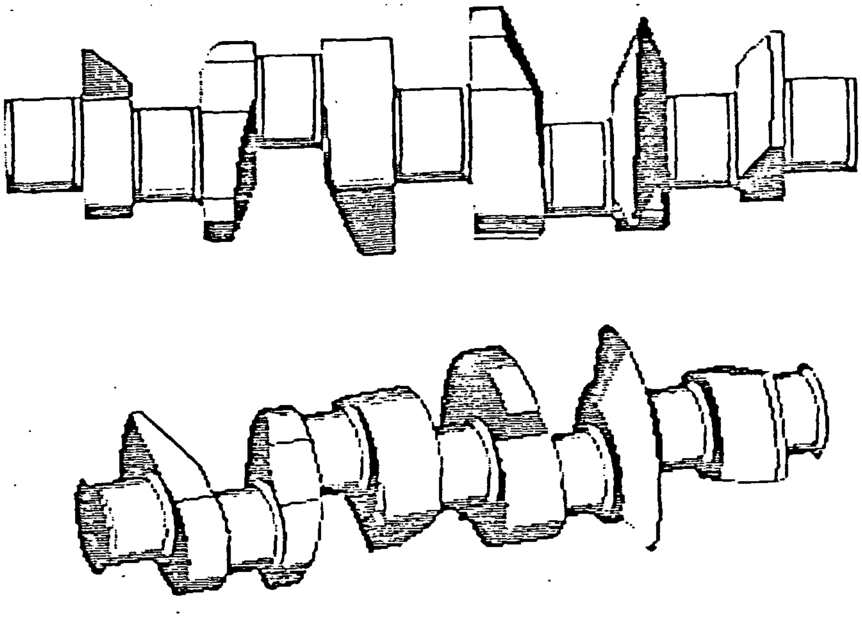


Fig. 5-1 RESD Crankshaft Modeling

*See MTI Report No. 83ASE334SA4

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

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

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

- provide a larger engine test base to assess engine performance/durability.

The ITEP program has progressed considerably during the last half of 1983. Hardware procurement is complete and, except for some spare parts, all hardware is in-house. Two companies have been selected for the program. Upgraded Mod I engine No. 8 has been built, and is currently undergoing functional and characterization testing.

The two companies selected for the industry evaluation are General Motors and John Deere. General Motors requested delivery of a vehicle, while John Deere will first evaluate their engine in a test cell and then in a generator set.

The General Motors engine (upgraded Mod I engine No. 8) will be installed in an AMC Spirit vehicle, and delivered to General Motors in April, 1984 following test cell checkout of the engine, and vehicle transient development tests. The John Deere engine (upgraded Mod I engine No. 9) will be assembled and tested at MTI, and then shipped to John Deere by the end of July 1984.

Figure 6-1 shows the current ITEP-related activities scheduled for 1984/1985.

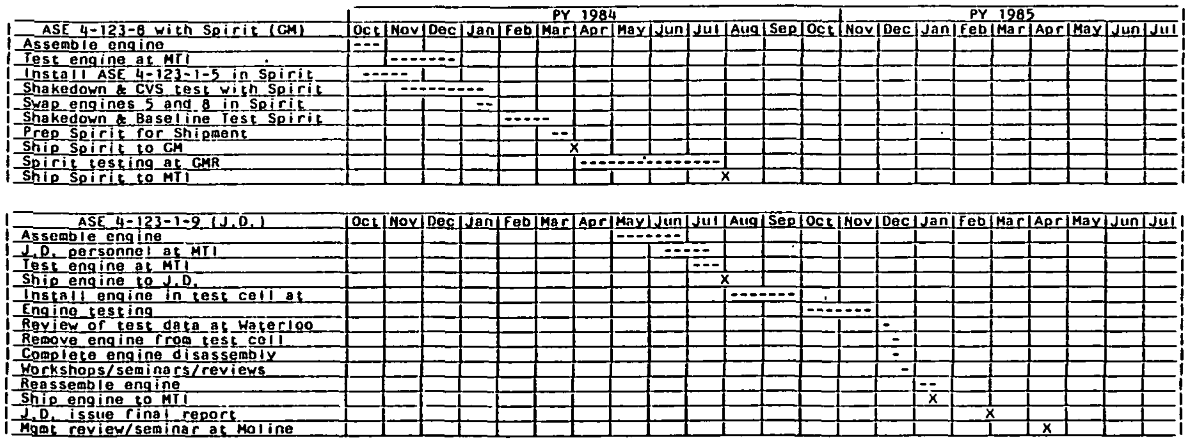


Fig. 6-1 ITEP-Related Activities

VII. PRODUCT ASSURANCE

Quality Assurance Overview

Below is the status of the ASE Program Quality Assurance Reports (QAR's) as of December 31, 1983:

Open QAR's	118
Closed QAR's (Total to Date)	526
P-40 QAR's	144
Mod I QAR's	415
Upgraded Mod I QAR's	85
Total QAR's in System	644

Mod I QAR Experience

A summary of problems documented via the QAR system is presented in Table 7-1 and Figures 7-1 through 7-4.

Problems are defined as items that: 1) cause engines to stop running; 2) prevent engines from being started; and, 3) cause degradation in engine performance. Problems that fall into these categories must be minimized to provide acceptable engine performance and mean time between failures.

Major problems identified for individual units/assemblies that were established prior to June 30, 1983 are shown in comparison with the results of the last semiannual report period.

Table 7-2 is a summary of the operating times versus failures for all ASE Program Mod I/upgraded Mod I engines as of December 31, 1983.

Table 7-1
Major Problem Summary

Established Prior to 6/30/83	From 6/30 to 12/31/83	% of Total
Moog Valve	1 Report	5
Heater Head	3 Reports	19
Check Valves	2 Reports	18
Combustion Blower	6 Reports	33
Fuel Nozzle	7 Reports	35
Igniter	1 Report	17
Preheater	7 Reports	50
Atom. Air Comp./ Servo-Oil Pump	1 Report	20
Combustor	4 Reports	40
Flame Shield	3 Reports	30
PL Seal Assembly	3 Reports	27
Crankcase/Bed Plate	1 Report (New)	
Mod I Hrs Prior to 6/30/83 - 2689 Mod I Hrs Accum. as of 12/31/83 - 2132		
44% of Hrs Accum. from 6/30 to 12/31/83		

Table 7-2
Summary of Operating Times Versus Failures for All ASE Program Mod I/ Upgraded Mod I Engines as of 12/31/83

Mod I Engine	Operation Time	Mean Operating Time to Failure
No. 1	655	50.4
No. 2*	1047	174.5
No. 3*	1452	121
Upgraded No. 5*	345	69
Upgraded No. 6*	203	---
Upgraded No. 8*	36	36

*All classes of failures since initial start of engine are included in the calculation of mean time to failure. Remaining engine's operation times are based on time and failures occurring after acceptance test.

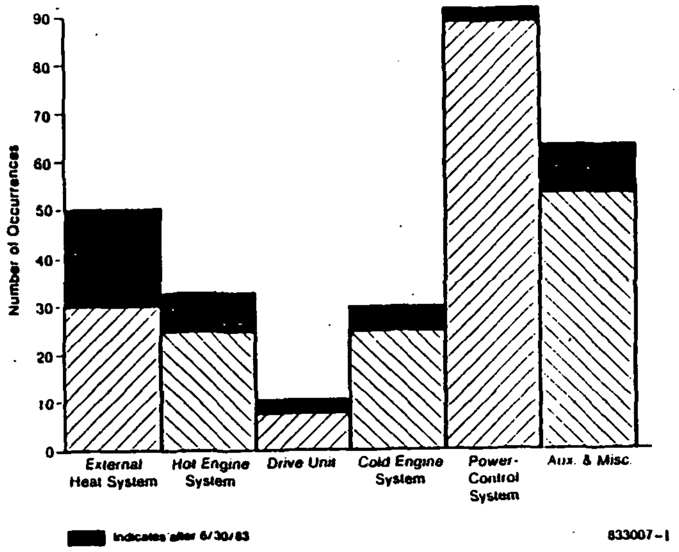


Fig. 7-1 Upgraded Mod I Engine Major Systems' Failures/Discrepancies Through 12/31/83

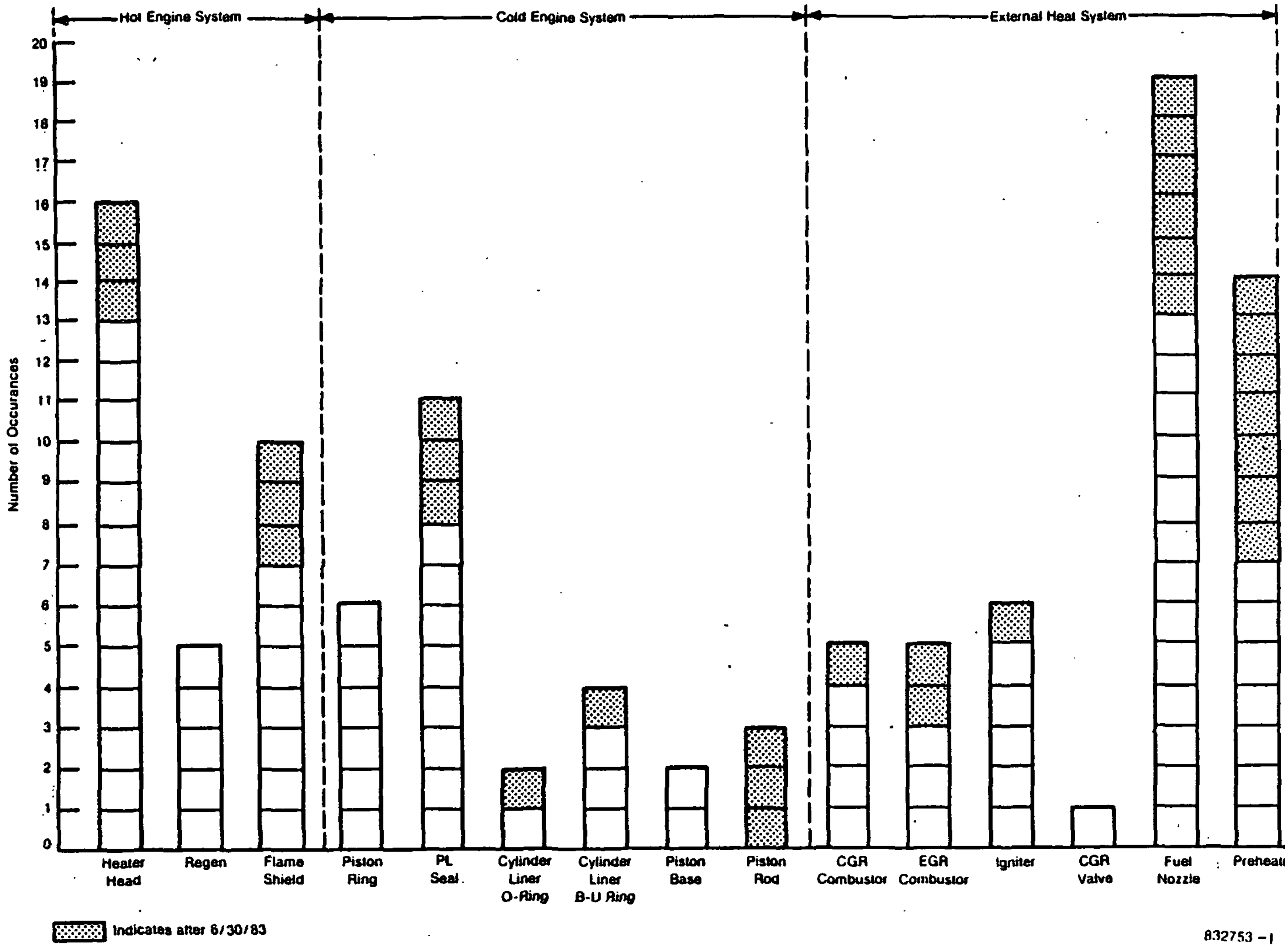


Fig. 7-2 Hot Engine, Cold Engine, External Heat Systems' Failures/Discrepancies Through 12/31/83

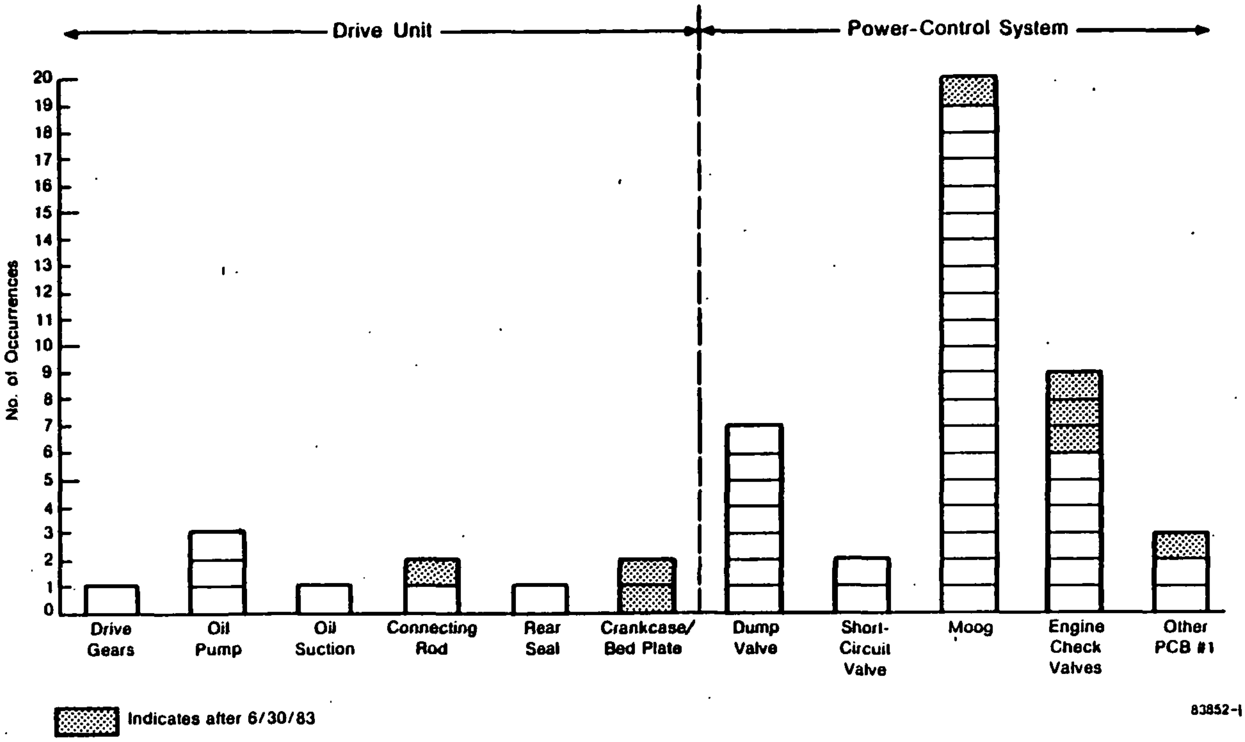


Fig. 7-3 Drive Unit and Power-Control System Failures/Discrepancies Through 12/31/83

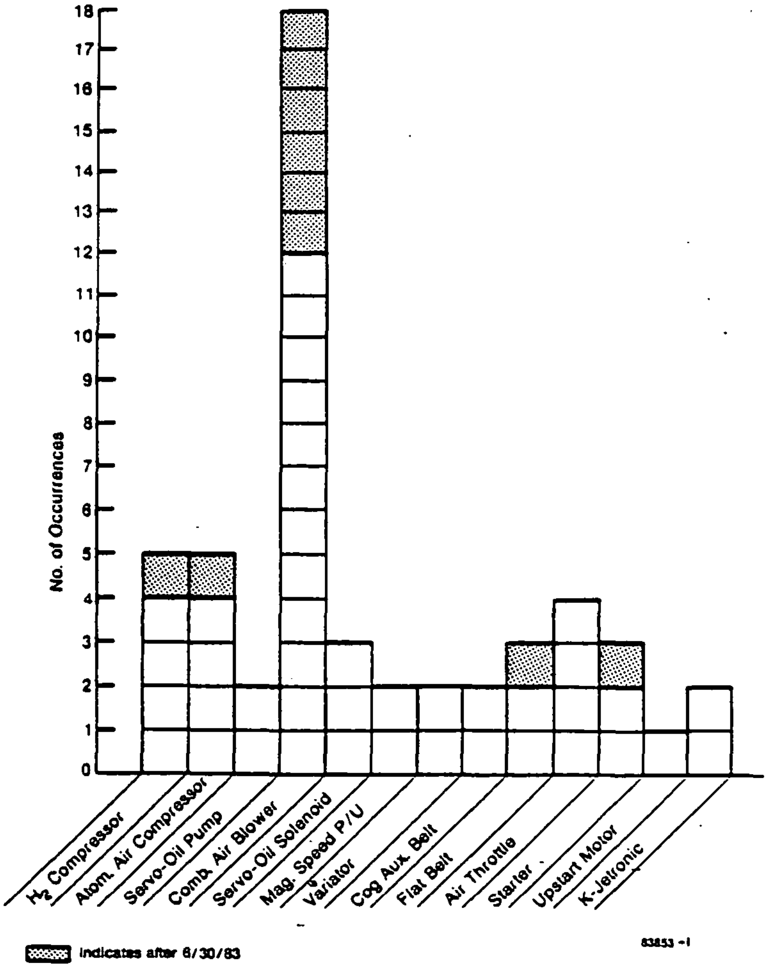


Fig. 7-4 Auxiliaries and Miscellaneous Items' Failures/Discrepancies Through 12/31/83

VIII. FACILITIES

Listed below is a brief summary of the facilities activities performed during the second half of 1983:

- Minor maintenance was performed on all test cells.
- All test cells are currently operational.
- A preliminary design study was completed on a second test cell at MTI.
- A positive-displacement seals test rig is being installed in the High-Pressure Cell.

APPENDIX - TERMS AND DEFINITIONS

Term	Definition
AFC	Air/Fuel Control
Al	Aluminum
AMG	AM General
AOP	Average Operating Point
ASE	Automotive Stirling Engine
B	Boron
BOM	Bill-of-Materials
BSE	Basic Stirling Engine
C	carbon
°C	degrees Celcius
Cb	columbium
CES	Cold Engine System
CGR	combustion gas recirculation
Co	cobalt
CO	carbon monoxide
CO_2	carbon dioxide
Cr	chromium
CRU	Communications Register Unit
CSP	Cold-Start Penalty
CVS	constant volume sample
DAFC	Digital Air/Fuel Control
DAS	Data Acquisition System
DC	direct current
DCC	Digital Combustion Control
DEC	Digital Engine Control
DOE	Department of Energy
EDS	Engine Drive System
EGR	Exhaust Gas Recirculation
EHS	External Heat System
EHSTR	External Heat System Transient Response Code
°F	degrees Fahrenheit
Fe	iron
ft	foot
ft^2	square foot
ft^3	cubic foot
FY	fiscal year
g/mi	grams per mile
g/s	grams per second
HC	hydrocarbon
HES	Hot Engine System
hp	horsepower
HTP-40	High-Temperature P-40 Engine
Hz	hertz
in	inch
kg	kilogram
ksi	thousand pounds per square inch
kW	kilowatt
lbs	pounds
LRFD	Lightweight Reduced Friction Drive
m	meter
mi	mile
mm	millimeter
Mn	manganese
Mo	molybdenum
Mod I	first-generation automotive Stirling engine
MTI	Mechanical Technology Incorporated
MPa	megapascals
MQ	material quote
N	nitrogen
NASA	National Aeronautics and Space Administration
Ni	nickel
NO_x	oxides of nitrogen
NTU	number of transfer units
PL Seal	Pumping Leningrader Seal
P_{max}, P_{mean}, P_{min}	working gas pressures
psi	pounds per square inch
psig	pounds per square inch gauge
PTFE	polytetraflouroethylene
P-V	pressure-volume
QAR	Quality Assurance Report
RESD	Reference Engine System Design
RFD	Reduced Friction Drive
rpm	revolutions per minute
s	second
SES	Stirling Engine System
Si	silicon
S/N	serial number
SS	Stainless Steel
Ta	tantalum
T/C	thermocouple
T_{cgat}	combustion gas temperature after tubes
TTB	Transient Test Bed
USAB	United Stirling of Sweden AB
VE	Value Engineering
W	tungsten
λ	$\frac{\text{air/fuel}}{\text{air/fuel}_{\text{stoichiometric}}}$
η	efficiency
Δ_η	efficiency difference
ΔWt.	weight difference
ΔPower	power difference
ℓ/min	liters per minute

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15. Supplementary Notes

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16. Abstract

This is the fifth Semiannual Technical Progress Report prepared under the Automotive Stirling Engine Development Program. It covers the twenty-second and twenty-third quarters of activity after award of the contract. Quarterly Technical Progress Reports related program activities from the first through the thirteenth quarters; thereafter, reporting was changed to a semiannual format.

This report summarizes activities performed on Mod I engine testing and test results, testing of the Mod I engine in the United States, Mod I engine characterization and analyses, Mod I Transient Test Bed fuel economy, upgraded Mod I performance and testing, Stirling engine reference engine manufacturing and reduced size studies, components and subsystems, and the study and test of low cost casting alloys. The overall program philosophy is outlined, and data and results are presented.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
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