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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

SEMIANNUAL TECHNICAL PROGRESS REPORT FOR PERIOD: JULY 1 - DECEMBER 31, 1984

Mechanical Technology Incorporated

November 1985

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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

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SEMIANNUAL TECHNICAL PROGRESS NARRATIVE REPORT

FOR PERIOD OF JULY 1 - DECEMBER 31, 1984

November 1985

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INTRODUCTION

In March 1978, a Stirling-engine development contract, sponsored by the Department of Energy (DOE) and administered by National Aeronautics and Space Administration (NASA)/Lewis Research Center, was awarded to Mechanical Technology Incorporated (MTI) for the purpose of developing an Automotive Stirling Engine (ASE) and transferring Stirling-engine technology to the United States (U.S.) 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 U.S., and was directed at stationary and marine applications. The ASE Development Program was directed at the establishment and demonstration of a base of Stirling-engine technology for the automotive application by September 1984. The high-efficiency, multi-fuel 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 System Design (RESD) to serve as a focal point for all component, subsystem, and system development within the program. The RESD was defined as the best-engine design generated within the program at any given time. The RESD would incorporate all new technologies with reasonable expectations of development by 1984, and which provide significant performance improvements relative to the risk and cost of their development. The RESD would also provide the highest fuel economy possible while still meeting other program objectives.

A schedule was defined within the ASE Program to design two experimental engine versions of the RESD. The first-generation engine system, the Mod I, was designed early in the program, and has been on test since January 1981. The second-generation engine, designated the Mod II, was originally scheduled to be designed in 1981 to demonstrate the final program objectives. However, it was postponed due to Government funding cutbacks. As a result, the Mod I has been the only experimental engine in the program.

Through the course of the program, the Mod I has been modified and upgraded wherever possible, to develop and demonstrate technologies incorporated in the RESD. As a result, the program followed a "proof-of-concept" development path whereby the Upgraded Mod I design emerged as an improved engine system, proving specific design concepts and technologies in the Mod II that were not included in the original Mod I design. 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 Mod II.

Nevertheless, some of the new technology incorporated in the RESD has been successfully transferred to the Upgraded Mod I engine. Iron-based materials were used in place of costly Cobalt-based materials in the Hot Engine System (HES) which was designed to operate at 820°C heater head temperature (the Mod I was tested at 720°C). Smaller, lighter designs were incorporated into the upgraded engine to optimize for better fuel economy and to reduce weight (the Upraded Mod I engine was 100 lb lighter than the Mod I).

The RESD has been revised periodically throughout the course of the program to incorporate new concepts and technologies aimed at improving engine efficiency or reducing manufacturing cost. The RESD was last revised in May 1983. Emphasis of this most recent update of the RESD was to reduce weight and manufacturing cost of the ASE to within a close margin of that for the spark-ignition engine, while exceeding the fuel mileage of the spark-ignition engine by at least 30%.

This May 1983 RESD configuration was changed substantially from previous designs to achieve these goals. The new design uses a single-shaft V-drive, rather than the two-shaft U-drive system of the Mod I; an annular heater-head and regenerator rather than the previous cannister configuration; and a simplified control system and auxiliary components. By these measures, the projected manufacturing cost of the May 1983 RESD was reduced by more than 25% and total engine system weight was reduced by 47% in comparison to the Upgraded Mod I engine, while engine efficiency and power remained approximately the same. This 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 lb.

Since the RESD update in May 1983, a preliminary Mod II design phase has been aimed at translating the new RESD concepts into preliminary Mod II design drawings. Casting drawings of the annular heater head and single-piece V-block were made and reviewed with vendors; the lower end drive system was designed for a durability rig to test the life and operational behavior of the bearings, oil and seal systems, and gas passages. An analysis was performed on the Mod II engine vehicle system to select the vehicle and matching drive train components such as transmission, gear ratios, and axle ratio.

During the second half 1984, preliminary design of the Mod II was concluded with a Technology Assessment, which selected specific technologies or configurations from competing contenders for each component of the Mod II engine. Component and technology development was continued, with continuing emphasis on main seal and piston ring evaluation, and on combustion gas recirculation (CGR) combustor development. Initial detail design of the Mod II was started. Analytical models and algorithms for all aspects of the engine, auxiliaries and parasitics were formulated and integrated into computer analyses for optimization of the Mod II configuration. Testing of Upgraded Mod I engine No. 8 in the Spirit vehicle was continued after return from the General Motors (GM) Industry Test and Evaluation Program (ITEP) evaluation. This vehicle testing was intensified and focused on early development and evaluation of new controls and auxiliaries concepts to be incorporated in the Mod II. Upgraded Mod I engine No. 9 was built and delivered to John Deere and Company (Deere) for their test evaluation under the ITEP Program. Testing and analysis of the Mod I engine system was continued, although focused more narrowly on providing specific input to guide the Mod II design effort.

Listed below are some of the work tasks and achievements undertaken during the July-December 1984 period.

- The tubular-to-elliptical (TTE), CGR combustor was developed to the point where it appears to be a leading contender for incorporation into the Mod II.

- A successful test was performed on an improved ceramic preheater section, which survived 300 start/stop thermal cycles with leakage less than 2%.

- A total of 27 investment castings were made of the prototype XF-818, two-manifold, Mod II heater head during development of the manufacturing process.

- Tests of the cycle-gas flow distribution in the prototype Mod II heater head were completed, with favorable results.

- Tests were completed to define the fatigue strength of XF-818 at room temperature.

- A detail design was completed for a shrink-fit joint between the Mod II piston rod and piston base, and a fatigue test of a prototype assembly was completed.

- Continued progress was made in demonstrating extended life for the PL main seals.

- A new uncut single-solid piston ring design was tested and shown to have potential for improving engine performance, with good life expectancy.

- The power control valve (PCV) design was improved to reduce the operating deadband by two-thirds.

- Further development of the Digital Air/Fuel Control (DAFC) and Digital Engine Control (DEC) was conducted, achieving idle conditions of 400 rpm, 2 MPa mean pressure and fuel flow as low as 0.18 g/s.

- A thorough Technology Assessment was conducted in September, and specific technologies or configurations were selected for each subsystem or major component of the Mod II engine.

- The Mod II cast-iron V-block was redesigned to improve castability and reduce weight.

- Thorough analyses of Mod II crankshaft deflection and roller bearing life were made, and specific design criteria were finalized for these components.

Seven Mod I engines are in active test service at the close of 1984. As of December 31, 1984, the cummulative total operating time for all Mod I engines has reached 8681 hr. Upgraded Mod I engine No. 9 was assembled and delivered to Deere, where it performed reliably during Deere's evaluation tests. Upgraded Mod I engine No. 10 was assembled and prepared for shipment to NASA/Lewis Research Center for NASA testing programs.

The ASE Program is proceeding on schedule, with increased emphasis on Mod II engine design requirements. During the January-June 1985 period, this focus on Mod II design will be more intense, with the objective of completing the Mod II Basic Stirling Engine (BSE) manufacturing drawings and the BSE Design Review by the end of March. Design concepts for controls and auxiliaries will receive concerted attention after that date, in preparation for the Stirling Engine System (SES) Design Review late in July.

I. SUMMARY

Since the inception of the ASE Program in 1978, 13 Quarterly Technical Progress Reports were issued under NASA Contract No. DEN3-32, "ASE Development Program". However, reporting was changed to a Semiannual format in July 1981. This report, the seventh Semiannual Technical Progress Report issued under the contract, and covering the period of July 1 through December 31, 1984, includes technical progress only.

Overall Program Objectives

The overall objective of the ASE Program is to develop an ASE system 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.*
- Show the potential for emissions levels less than: NO $\leq$ 0.4 g/mi, HC $\leq$ 0.41 g/mi, CO $\leq$ 3.4 g/mi, and a total particulate level $\leq$ 0.2 g/mi after 50,000 miles.

In addition to the previous objectives, which are to be demonstrated quantitatively, the following system design objectives were 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.
- Noise/safety characteristics that meet the currently legislated or projected 1984 Federal Standards.

Major Task Descriptions

The major ASE Program tasks are described below:

Task 1 - Reference Engine - This task, intended to guide component, subsystem, and engine system development, involves the establishment and continual updating of 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 1987) that is judged to provide significant improvement relative to the risk and cost of its development.

Task 2 - Component/Technology Development - Guided by the RESD, this task 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 sub-

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

system 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 drive train, controls, and auxiliaries.

Task 3 - Technology Familiarization - The USAB P-40 Stirling engine, which was available at the beginning of the program, was used as a baseline for familiarization, and 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 were built and delivered to the U.S. team members - one was installed in a 1979 AMC Spirit. A fourth P-40 engine was built and installed in a 1977 Opel sedan for testing in Sweden. The baseline P-40 engines were tested in dynamometer test cells and automobiles. Test facilities were constructed at MTI to accommodate the engine test program and required technology transfer.

Another activity under Task 3 during 1984 was the Industry Test and Evaluation Program (ITEP). Major purposes of ITEP are to extend familiarity with and interest in Stirling engine technology into industry, to provide an independent evaluation of ASE technology, to broaden the base of engine test experience, and to give automotive/engine manufacturers an opportunity to make recommendations for improvements in design and manufacturability. Two Mod I engines were procured, assembled, and tested for delivery to automotive/engine manufacturing companies for test and evaluation.

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

The Mod I engine represented an early experimental version of the RESD, but was limited by the technology that could be confirmed in the time available. The Mod I was not intended to achieve any specific fuel economy improvements. Rather, it was meant to verify concepts incorporated in the RESD, and to serve as a stepping stone toward the Mod II, thus providing an early indication of the potential to meet the final ASE Program objectives.

Three Mod I engines were manufactured by USAB and tested in dynamometer test cells to establish their performance, durability, and reliability. Continued testing and development was necessary to meet preliminary design performance predictions. One additional Mod I engine was manufactured, assembled, and tested in the U.S. by MTI. A production vehicle was procured and modified to accept one of the engines for installation. Tests were 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 was upgraded through design improvements to provide a "proof-of-concept" demonstration of selected advanced components defined for the RESD and Mod II.

Task 5 - Mod II Engine - The Mod II (the second-generation ASE design) is based on the RESD, the development experience of the Mod I engine system, and technologies and components developed under Task 2. The goal of the Mod II is to demonstrate the overall ASE program objectives in an engine/vehicle system. Although postponed in 1981, this task was reinstated during the first half of 1984 as the preliminary design of the Mod II engine system was activated. In the latter half of 1984, this has transitioned into a preliminary detail design of the Mod II.

Task 6 - Prototype Study - Postponed.

Task 7 - Computer Program Development - Analytical tools have been developed that are required to simulate and predict engine performance. This effort included the development of a computer program specifically tailored to predict SES steady-state cyclical performance over the complete range of engine operations. Using data from component, subsystem, and engine system test activities, the program was continuously improved, and will be verified throughout the course of the ASE 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. This task also provides for an extensive Quality Assurance Program.

Program Overview, Status and Plans

During 1984, the ASE Program has undergone a notable shift in emphasis from the Mod I to the Mod II engine and final attainment of the overall program goals. During the second half of 1984, this shift in emphasis has been even more focused, culminating in the Mod II Technology Assessment late in September, and the commencement October 1 of detail design of the Mod II engine. Component/technology development has also been focused more specifically toward providing input data and technology required for the Mod II engine design.

RESD

The RESD was revised in May 1983, with a major departure from earlier versions of the engine concept. This revision was motivated primarily by the goal of achieving a manufacturing cost competitive with that of internal combustion (I.C.) engines, while still meeting other program objectives. As reported in the previous Semiannual, (MTI 84ASE408-SA6), an independent manufacturing cost study of the May 1983 RESD was completed during the first half of 1984 by Pioneer Engineering and Manufacturing Company. This RESD, to a substantial measure, is the foundation on which the Mod II engine design is based. Consequently, no further work was done on the RESD during the second half of 1984, with program resources being reserved for execution of the Mod II design. The next concerted effort to update the RESD will be in 1986.

Component and Technology Development

Component and technology development is directed toward advancing Stirling-engine technology and/or components in terms of performance, cost, manufacturability, durability, and reliability as defined by their application in the RESD.

This development involves a sizeable effort, as it includes virtually all technological areas in the engine (i.e., combustors, heat exchangers, materials, seals, mechanical drive systems, controls, and auxiliaries).

FUEL NOZZLE AND COMBUSTOR DEVELOPMENT

Efforts intensified during the second half of 1984 to advance the development and evaluation of the conical fuel nozzle and various alternative forms of combustion gas recirculation (CGR) combustors. This was done in anticipation of the need to make a selection of combustor and fuel nozzles for the Mod II engine late in September. A series of tests was completed on various modifications of the conical nozzle (shown in Figure 1-1) in conjunction with three variants of exchaust gas recirculation (EGR) combustors. Test results indicated that improved atomizing air distribution within the nozzle resulted in lower circumferential ΔT at the heater-head tubes. However, the nozzle produced relatively high soot in the high end of the fuel-flow range. The tests further indicated that: soot levels were similar when either K-Jetronic or Digital Air/Fuel Control (DAFC) fuel controls were used; the P-40 combustor turbulator yielded lower soot levels with the nozzle than did the Bill of Material (BOM) Upgraded Mod I turbulator; and fluctuations in air-throttle position (causing swings in air/fuel (A/F) ratio) were a major contribution to high CO and soot during these tests. The conical nozzle appeared to be more sensitive than the BOM nozzle to such fluctuations. Development testing of the conical nozzle will continue in 1985, as the benefits of lower atomizing airflow and virtual freedom from plugging are still viewed as strong recommendations for adoption into the Mod II engine.

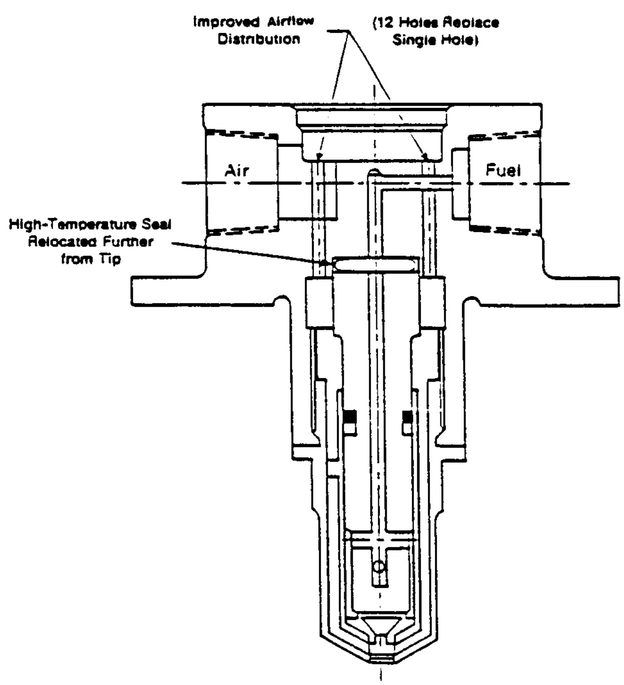

Figure 1-1 Conical Fuel Nozzle Modifications

In a continued effort to improve reliability and durability of the Upgraded Mod I External Heat System (EHS), tests were conducted to evaluate several alternative forms of EGR combustor turbulators. These included the Upgraded Mod I BOM turbulator, as well as the P-40 turbulator. It was concluded from the test data that the P-40 turbulator was better suited for operation with both the BOM fuel nozzle and the conical nozzle than was the BOM turbulator or any of the six alternative turbulator designs tested. This judgement was made on the basis of CO and soot emissions and heater-head rear tube ΔT.

One of the most significant developments during this report period was the results of continued testing of the 12-tube TTE, CGR combustor, illustrated in Figure 1-2. Minor modifications to this CGR combustor produced substantial improvements in performance when tested on engine No. 10 with the BOM fuel nozzle. Excellent soot and CO emissions levels with acceptable NO_x level was obtained over the entire range of fuel flow up to 4.5 g/s.

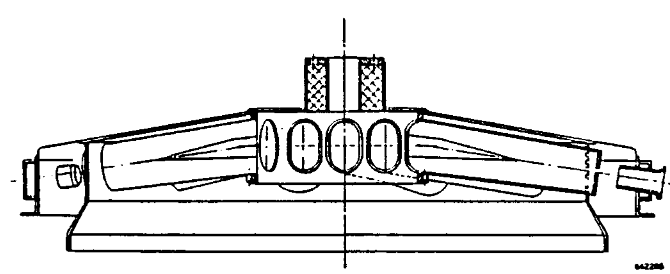

Figure 1-2 12-Tube TTE CGR Combustor

Extensive cold-flow and performance-rig tests were conducted on several design variations of 16-tube, 42-tube, and ribbon-type CGR combustors in an effort to identify a parallel CGR combustor equal or superior to the TTE configuration. However, none of these combustor design variations performed as well as the TTE did. Therefore, it was decided that CGR combustor development in 1985 will be concentrated on the TTE design.

In another area of combustor development, an Upgraded Mod I BOM combustor liner was treated in four separate sectors with four different oxidation-resistant coat-

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ings. This combustor liner is slated for evaluation testing during the first half of 1985.

CERAMIC PREHEATER DEVELOPMENT

The engine preheater is a recuperative heat exchanger that recovers heat from combustor exhaust gas and delivers this heat to fresh inlet air before it enters the combustor. The Mod I preheater is an annular, counterflow recuperator made from over 1000 embossed or stamped stainless steel plates, with a thermal effectiveness in the range from 0.9 to 0.95. The individual plates, each 0.1-mm thick, must be edge-welded on alternate pairs to form leaktight, adjacent, counterflow passages for exhaust gas and incoming air. A small sector of a Mod I welded, metallic preheater is illustrated in Figure 1-3.

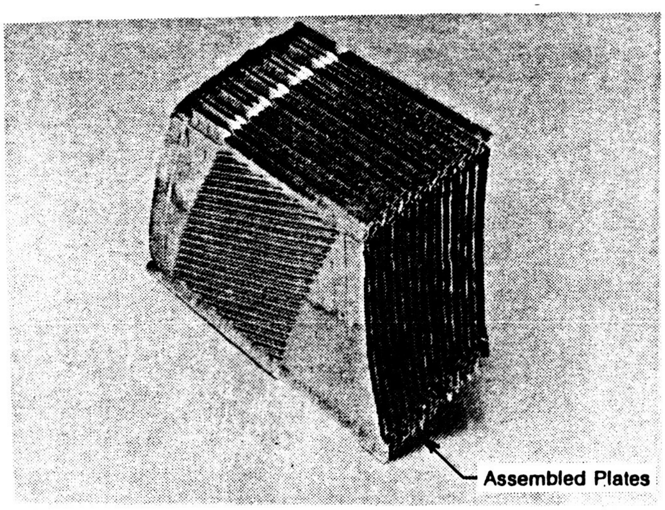

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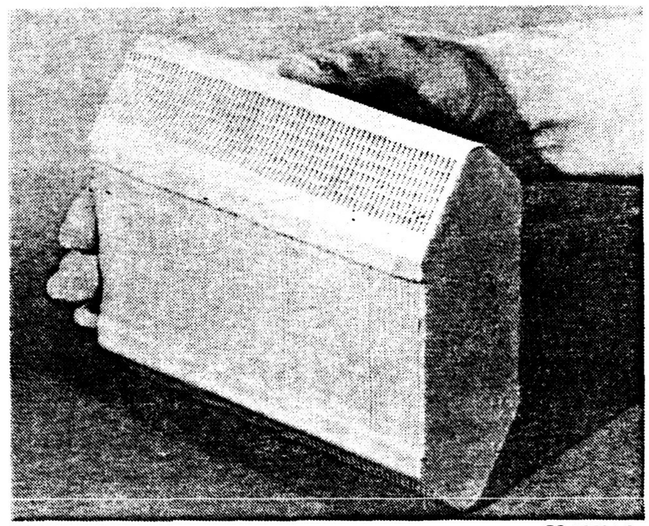
Figure 1-3 Sector of Welded Metallic Preheater

The requirement for such a large number of leaktight welds on a very thin material, as well as the relatively high cost of the thin stainless steel materials, has been a strong incentive to pursue less expensive, alternate designs. As reported in the previous Semiannual, a ceramic design being developed by Coors Porcelain Company appears to have some promise as an alternative. The design consists of blocks built up from 0.1-mm thick corrugated ceramic walls and fused together. One such block or segment is shown in Figure 1-4.



BS-19664-2

Figure 1-4 Ceramic Preheater Segment

The initial trial unit, made from cordierite material, suffered excessive leakage after only 16 thermal cycles simulating engine start-up and shutdown. A later test section, made from mixed-oxide ceramic material, exhibited leak rates of only 1 to 1-1/2% when first received. This test section survived extensive evaluation tests over a range of temperatures up to 820°C, and 300 thermal cycles, after which the leakage rate remained at 1 to 1-1/2%.

Testing of additional mixed-oxide samples during this report period resulted in unacceptable leakage rates. It was decided jointly by MTI and Coors that tooling changes and manufacturing of eight additional mixed-oxide test sections would be performed early in 1985, allowing continued development tests.

HEATER HEAD DEVELOPMENT

The heater head encloses the high-pressure gas-expansion portion of the engine cycle, and incorporates the high-pressure, high-temperature finned tubes that effect heat transfer from combustion gas to the cycle working gas. The Mod II annular design also places the regenerator within the heater head housing, as illustrated in Figure 1-5.

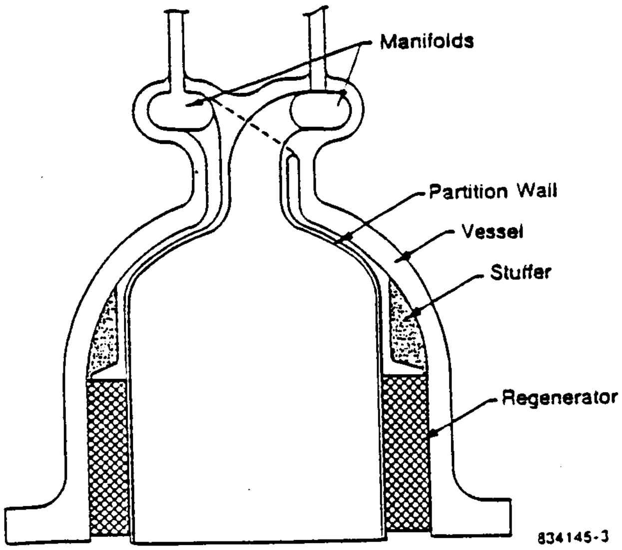


Figure 1-5 Annular Heater Head Housing Configuration

This also creates the necessity of having two manifolds on the heater-head housing, with concentric gas flow paths through the neck connecting the manifolds to the cylindrical portion of the housing as illustrated in Figure 1-5. In addition, the expansion space or cylinder must be isolated from the regenerator by a partition wall. This wall must be gas-tight and should prevent heat flow from the hot expansion space or cylinder radially outward to the regenerator. The preliminary design for Mod II also departs from Mod I heater-head design in tube size and fin geometry, as illustrated in Figure 1-6. These aspects of the Mod II design concept were perceived to pose three risks, namely:

1. Will the heater-head housing with manifolds be manufacturable?

2. Will flow of combustion gas over the new finned-tube geometry give predictable heat transfer performance?

3. Can radial heat conduction losses through the partition wall be kept to a sufficiently low level?

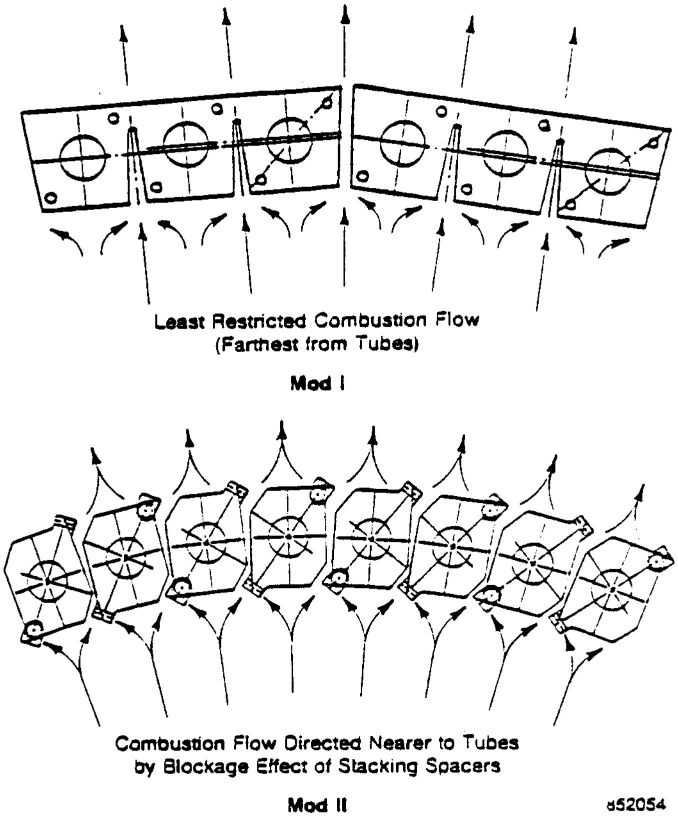


Figure 1-6 Comparison of Mod I and Mod II Fin Configurations

Each of these risks was addressed by a component-development subtask. A preliminary manufacturing effort was initiated earlier with Howmet Corporation to develop an investment casting of the dual-manifold heater-head housing in XF-818 alloy. A first version of this casting is shown in Figure 1-7. The ceramic core used to form the interior cavities of the manifolds is shown in Figure 1-8. Close-tolerance control of size, shape, and location of the core during casting must be exercised in order to maintain proper thickness of the cast walls of the manifolds, which must contain high internal gas pressure at high temperature during engine operation. During this report period, a total of 27 heater-head castings were poured in a continuing process-development effort. Significant improvement in dimensional control of the casting was achieved.

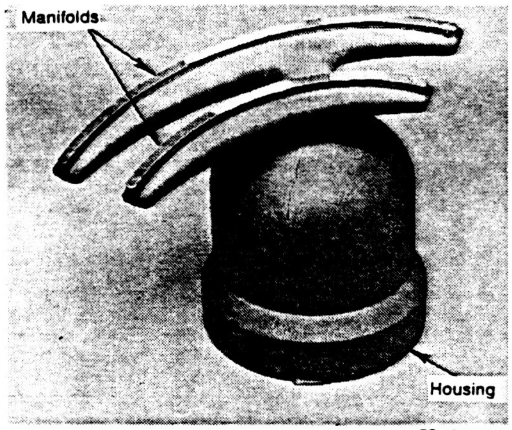


Figure 1-7 Mod II Heater Head Casting (XF-818 Material)

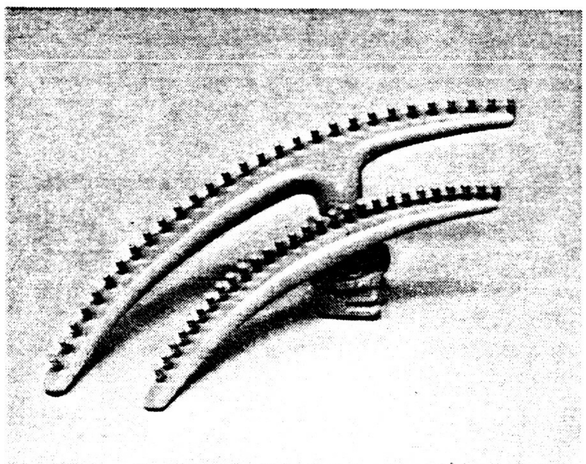


Figure 1-8 Ceramic Cores to Form Internal Flow Passages in Manifolds of Mod II Heater Head

Plastic models of the heater-head manifolds were made, using wax cores from Howmet, and room-temperature gas-flow tests were conducted to determine uniformity of flow distribution among the 30 tubes per quadrant. Flow distribution was found to be better than that of the Upgraded Mod I heater head. The heat-transfer characteristics of the new fin geometry will be determined by gas-flow/temperature-measurement tests of test sections of the new finned tubes in a double mantle rig (DMR) at USAB early in 1985. During this report period, DMR test sections were fabricated for both Mod I and Mod II finned rear-row and unfinned front-row tube geometries. Figure 1-9 shows a representative DMR test section for the Mod I rear row. In addition, an Upgraded Mod I heater head was fabricated using Mod II tubes and fins, to allow testing of the Mod II finned geometry on an engine early in 1985.

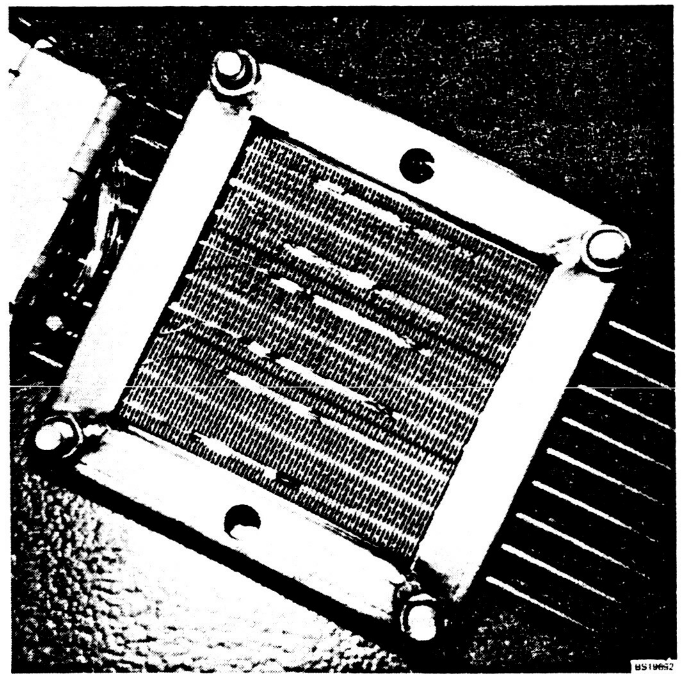


Figure 1-9 DMR Test Section of Mod I Rear Row Finned Tubes

MATERIALS AND PROCESS DEVELOPMENT

Both the housing and the tubes of the heater head are subject to high internal gas pressure and high operating temperatures. These conditions, plus the need for minimum engine weight, impose severe requirements on construction materials for high-temperature strength. Alloys initially employed for these components contained relatively high percentages of the strategic material cobalt, and were high in cost. As reported in the previous Semiannual, a Materials Evaluation Program was initiated in 1981 to rank candidate replacement alloys, and identify the best choice for the Mod II engine heater head tubes and housing. Candidate materials were tested by manufacturing actual heater head quadrants from each material and subjecting them to operation on an HTP-40 Stirling engine. These tests were completed during the first half of 1984. During the later half of 1984, the test data were reviewed and a summary report was completed. The test results ranked the candidate heater head casting alloys in the following order: 1) XF-818; 2) CRM-6D; and, 3) SAF-11. The XF-818 heater head casting did not leak after 1600 hr of test operation. Heater tube materials were ranked as follows: 1) CG-27; 2) Inconel 625; 3) 12RN72; 4) Sanicro 30; and, 5) Sanicro 32H. In addition to completing this comparative evaluation, tests were completed to determine the room-temperature fatigue properties of alloy XF-818. The test data indicated a fatigue endurance strength of $\sim$375 MPa at a stress ratio of R=0.5, and 180 MPa at R=-1.

As reported in the previous Semiannual, an effort was started in the Spring of 1984 to evaluate a ceramic partition wall. The motivation for this effort is the low thermal conductivity of ceramics, which could reduce the radial heat loss from the expansion space through the partition wall to the regenerator. During this report period, fabrication was completed of a Nilcra TS-Grade, partially stabilized zirconia sleeve, assembled inside a stainless steel can with an interference fit. This will be tested in the P-40R annular engine during 1985.

Fatigue tests were completed on the first version of an interference-fit joint between the piston rod and piston base for the Mod II engine. Failure occurring after 3.4 million cycles at 100% of full load provided guidance for a redesign of the joint. Fabrication and test of the redesigned joint will be accomplished during the next report period.

MAIN SEAL DEVELOPMENT

The main seal for the ASE engines seals the reciprocating piston rods, separating high-pressure hydrogen gas within the engine cycle from lubricant oil in the crankcase. A Pumping Leningrader (PL) configuration made from either HABIA or Rulon J material is currently the preferred type for Mod I engines, and is also planned for use in the Mod II engine. During this report period, the influence of bore diameter on PL seals was checked by testing two seals with .015-mm undersize bores in the exploratory and test rig. Friction and leakages of these seals were similar to those of standard PL seals. The influence of the low idle speeds and pressures planned for the Mod II engine on PL seal operation was also explored. Seal friction force and operating temperature were apparently unaffected by operation at 400 and 500 rpm, at gas pressures from 1.0 to 3.5 MPa. Seal friction force was found to be virtually constant throughout the speed range from 400 to 3000 rpm, with lube oil temperature of 32° and 50°C. The higher oil temperature resulted in a slightly lower friction force. These tests give reassurance that PL seals in the Mod II engine will suffer no adverse effects due to low engine idle speed. PL seals were also tested in the exploratory rig under simulated engine start-up conditions after a 15 hr shutdown period, with oil supply shut-off. The seals operated successfully for $\sim$25 sec with low friction force at 400 rpm. This verified that sufficient oil is trapped and retained in the seal/rod interface after extended shutdown periods to support operation for substantially longer than the period required for the engine oil pump to deliver oil to the seals.

Continued tests were conducted on the pumping ring seal, another alternative to the PL type. Four pumping ring seals were tested in P-40 engine No. 9 for a total engine run time of 1342 hr. Engine crosshead clearance ranged from 18 to 65 μm, the same range employed in a previous test of PL seals in engine No. 9. Pumping-ring seal performance was similar to that of PL seals, with the seal in No. 2 cycle (41 μm crosshead clearance) operating the full 1342 hr without failure.

Extended engine tests were also run on PL seals in Mod I engine No. 7. The engine was built with steel crosshead guides, BOM crosshead clearances, using two HABIA PL seals and two Rulon J PL seals. The test was run for 2158 hr on the accelerated duty cycle. One HABIA and one Rulon J seal operated successfully throughout this entire period. The other two seals suffered oil leakage due to dimensional discrepancies. This successful, long-life operation of properly manufactured PL seals is an encouraging substantiation of Stirling engine operating life.

PISTON RINGS

Piston ring development during this period was centered on evaluation of the single-solid configuration shown in Figure 1-10.

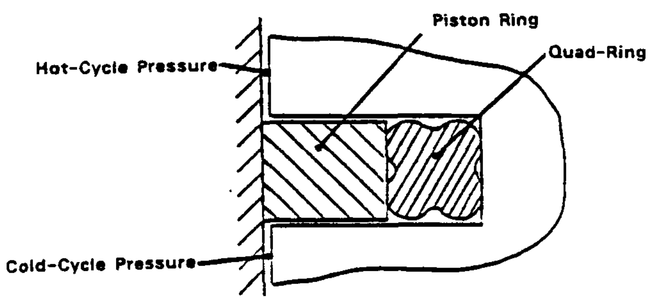


Figure 1-10 Single-Solid Piston Ring

Three sets of single-solid rings were tested successfully in the Mod I motoring rig, and gave consistent results for motoring power and cycle-to-cycle pressure differences. Test results were comparable to or slightly better than those for the BOM split-solid piston rings. A 500 hr test of the split-solid rings was completed in the motoring rig, to form a baseline for evaluation of the single-solid rings. The 500 hr test of single-solid rings was begun during December 1984. The single-solid rings were also evaluated in Mod I engine No. 5. Back-to-back tests of split-solid and single-solid rings were run, with the single-solid rings giving slightly higher engine power at high speeds and pressures. Single-solid rings were left in the engine, have completed 90 hr of consistent operation, and the engine-test evaluation continues during early 1985.

CONTROLS AND AUXILIARIES

The controls and auxiliaries are expected to be major contributors to the achievement of engine/vehicle fuel mileage goals for the Mod II engine. Therefore, development effort was focused on those areas of improvement which are slated for incorporation in the Mod II engine. Effort was also devoted to modifications to effect a reduction in idle fuel flow. After conducting an evaluation test of a Mod I powered Spirit vehicle, General Motors (GM) had previously identified high idle fuel flow as an area where significant improvement in engine fuel economy might be made. New control algorithms were formulated and successfully tested to allow stable idle operation at 400 rpm, 2.4 MPa mean pressure. These were also tested in engine No. 8 in the Spirit with good results. The PCV design was changed to reduce the valve deadband by two thirds. This will improve the transient response of the valve. Further improvements in the DAFC were made, with successful test operation achieved on engines No. 3 and 8. An improved igniter power-supply circuit with better reliability and durability was developed and put into service.

Preliminary performance requirements and system design were formulated for electric-motor drive for the combustion air blower for the Mod II engine. A companion system was defined for a high-efficiency alternator to provide electric power to the blower motor, as well as to all other vehicle electrical systems. These initial design studies indicated that this electric power transmission system from engine to blower will be more efficient than the variator belt system employed on the Upgraded Mod I engine.

ITEP

The purpose of the ITEP was to provide an independent evaluation of ASE technology, and to provide a forum for U.S. automotive and/or engine manufacturers to input recommendations relative to improving the design and manufacturability of Stirling engines. It was also recognized that such a program would provide a larger engine test base from which Stirling engine performance and durability could be assessed.

Two industrial companies participated in ITEP, namely; GM and John Deere and Company. The GM portion of ITEP was completed in June 1984, with their testing of Upgraded Mod I engine No. 8 installed in a 1981 AMC Spirit vehicle. These tests were reported in the previous Semiannual.

The Deere ITEP testing phase was completed in October 1984, with their test of Upgraded Mod I engine No. 9 at Deere's Product Engineering Center in Waterloo, Iowa. Deere is currently preparing a test report to summarize their evaluation of the engine. Engine No. 9 was built with a reinforced crankcase assembly, steel crosshead guides and high-temperature (820°C) heater heads. The engine was tested at MTI prior to delivery at Deere at tube temperatures of 720°, 770°, and 820°C. Testing at Deere included operation on gasoline, JP-4 and No. 2 diesel fuel.

Mod I Engine Test Program

At the end of 1984, ASE Mod I engines have accumulated a total of 8681 hr of test operation, as shown in Figure 1-11. A total of 2263 hr were accumulated during this report period. Figure 1-12 shows the accumulation of test hours during all of 1984 by individual engines. Mod I engine No. 7 (devoted to main seals development) has been the most active, accumulating 960 hr during this report period. This has been predominantly operation under an "accelerated duty cycle" to simulate both the steady-state and transient effects of the EPA driving cycle.

Mod I engine No. 3 accumulated 180 hr during this report period, devoted to development of the EHS, and evaluation of both EGR and CGR combustors.

Mod I engine No. 1 has not been active during this report period. Upgraded Mod I engine No. 5 is tested in the MTI engine test cell, and accumulated 199 hr for a variety of development test purposes during this report period. Upgraded Mod I engine No. 6 is located at USAB and accumulated 349 hr of operation, devoted primarily to endurance testing at a heater-head temperature of 820°C.

Upgraded Mod I engine No. 8 (the ITEP GM engine) has been installed in the Spirit vehicle at MTI, and has accumulated 287 hr of operation during this report period. This has been predominantly in vehicle transient tests, with evaluation of controls and auxiliaries during transients a primary purpose. Upgraded Mod I engine No. 9 started operation with this report period, and has accumulated 218 operating hours. After initial tests at MTI, the engine was shipped to Deere for evaluation in their test cell. Further testing was done at MTI after the engine was returned from John Deere. Upgraded Mod I engine No. 10 was built from Mod I engine No. 1. It was tested at USAB at then delivered to NASA-LeRC for use in NASA testing programs, and has accumulated 81 hr this report period.

Mod II Engine Development and Technology Assessment

Preliminary design of the Mod II engine continued during this report period, culminating in a Technology Assessment conducted September 25-26, 1984. Design of the Mod II durability test rig was completed and procurement/fabrication was also virtually completed. The durability rig is a first build of the Mod II Cold Engine and Drive System (CEDS) based on a preliminary version of the Mod II cast-iron V-block, using dummy heater heads and no EHS. It is intended for motoring tests to evaluate the roller bearings and other new aspects of the Mod II CEDS. Difficulty was encountered in obtaining a sound casting of the V-block. Castings have been flawed by porosity in a high-pressure gas-containing wall. Development of the casting process was con-

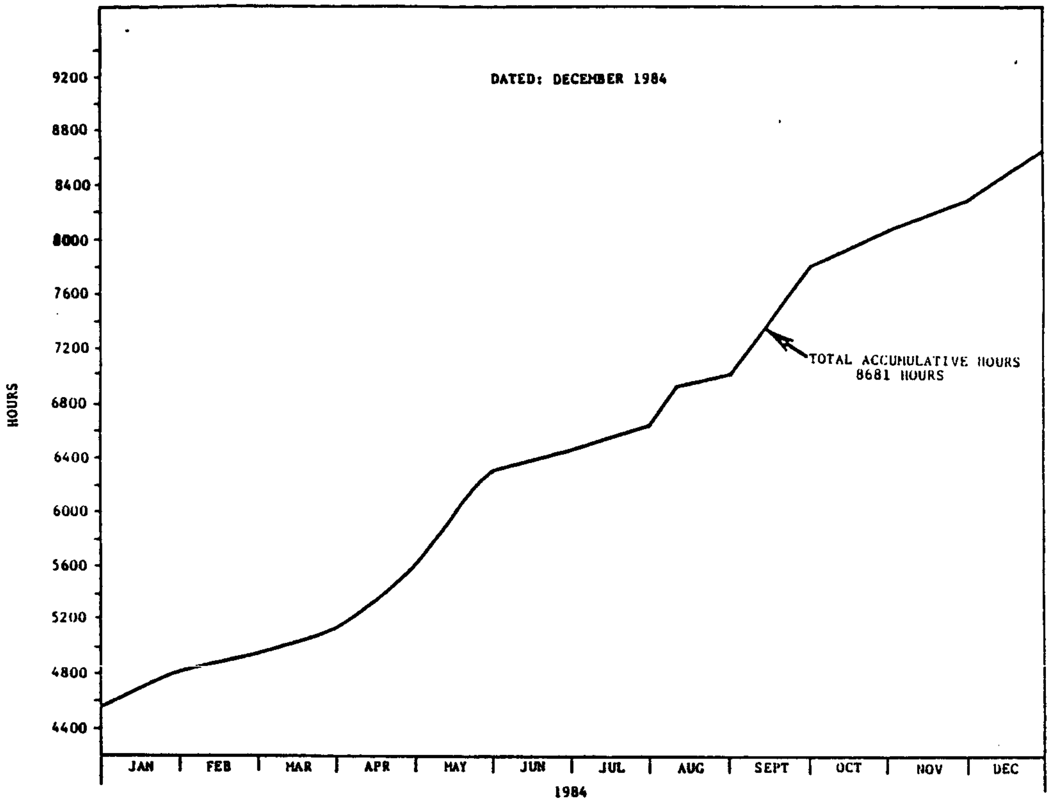


Figure 1-11 Total Mod I Engine Test Hours as of December 31, 1984

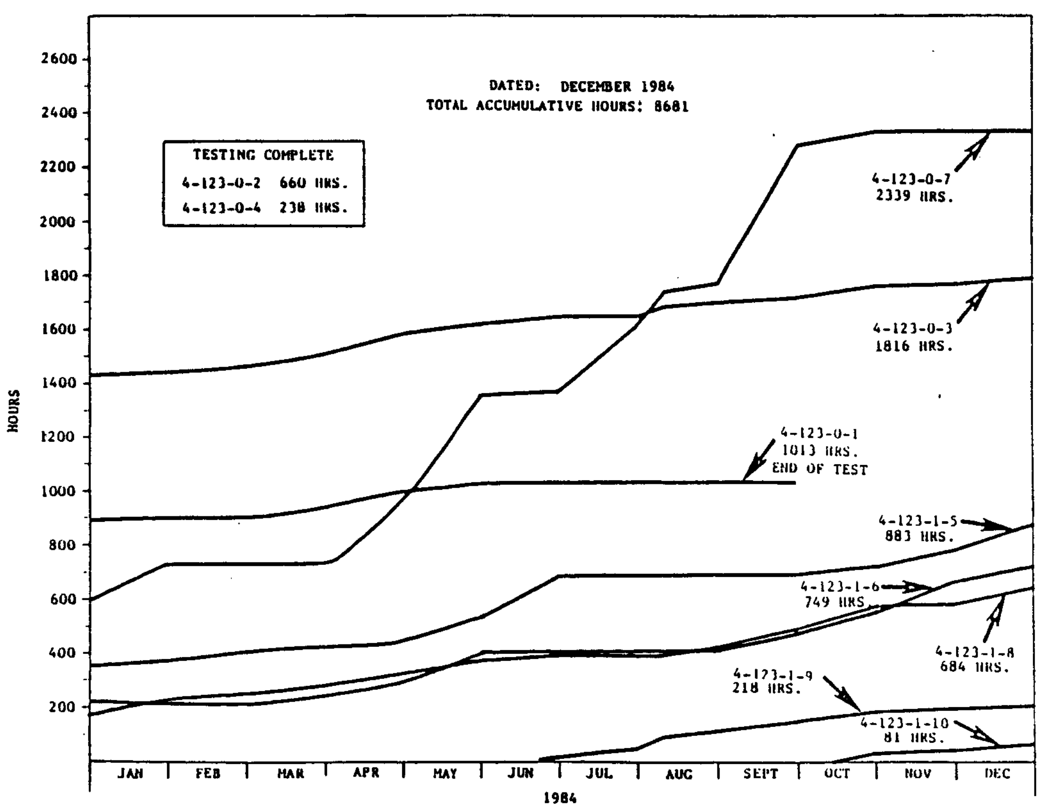


Figure 1-12 Total Mod I Engine Test Hours by Engine as of December 31, 1984

tinued, in an effort to correct this condition.

The Technology Assessment conducted September 25-26 examined the alternative technologies available for each area of the Mod II engine, and selected the preferred alternative for each item. The dominant criteria for selection were: potential for achieving Mod II engine/vehicle performance goals, and development/procurement risk. The selection of Mod II engine configuration and technologies made at that time is summarized below:

EHS: CGR combustor (ribbon configuration); conical fuel nozzle and separate side igniter; metallic preheater; and, Kanthal flamestone.

HES: CG-27 tube (Inco 625 back-up), 2.5-mm I.D./4.0-mm O.D.; dense-pack, single-separate fins of Inco 601 material; and, two parallel heater-head configurations:

- Configuration No. 1 - (low risk for manufacturing lead time); XF-818, two manifolds.
- Configuration No. 2 - (high performance potential); HS-31 material, one or zero manifolds.
- Mesh-type regenerator with no outer can or shell.
- Metallic partition wall; cooler with thin, high-strength wall.

CEDS: Engine V-block to be an interim machined, multi-piece configuration (casting considered high-risk for meeting Mod II assembly schedule); roller bearings; single-solid piston rings of Rulon LD; PL rod seals of Rulon J, with cap seal.

Controls and Auxiliaries: Mean-pressure control with 400 rpm idle speed; electrically driven PCV, 4 ℓ H_2 storage bottle; two-stage H_2 compressor; improved check valves. DAFC with optimized λ control; micropump fuel metering pump; and, Hitachi air-measuring unit. DEC to be the Texas Instruments type 9995 microprocessor with five boards. High-voltage/high-speed alternator; centrifugal combustion air blower and atomizing air compressor to be electrically driven; single 24 V battery.

The Mod II vehicle definition was also finalized during this report period to be a 1985 Chevrolet Celebrity with a four-speed manual transmission, 3125 lb inertia-weight class. The vehicle system will include dual, staged radiator fans, power steering pump, and no air conditioning.

Mod II Engine Final Design

After selection of the Mod II configuration at the Technology Assessment, initial detail design of the Mod II commenced October 1, 1984. The engine block was redesigned to further reduce weight and to improve castability. An "analog" of the cast V-block configuration was designed to be machined from forged AISI 4130 material. This will be employed for initial build of the Mod II engines, to avoid the perceived high risk attached to the cast block design, which will progress in parallel.

Extensive analysis of the roller bearings for crankpin and crankshaft main bearing locations was performed. It was determined that journal diameters must be increased from 35 to 38-40 mm to avoid excessive misalignment in the roller bearings due to crankshaft bending deflections. Bearing dynamic load capacity must also be increased above that of preliminary bearing selections, to ensure meeting bearing life goals.

Algorithms for all engine parasitic losses (e.g., piston-ring losses, blower power, waterpump power, etc.) and other engine modeling functions were formulated in November and a first optimization of the Mod II engine was commenced in December.

Work Planned for January-June 1985

During the period January 1 through June 30, 1985, the major activities listed below are scheduled for completion.

- Final optimization of the Mod II engine and detailed definition of engine configuration.
- Generation of engine performance maps for the Mod II engine.
- Completion of detail drawings for for the Mod II BSE.
- Completion of the BSE Design Review, April 2-3, 1985.
- Definition of SES component design and preliminary execution of detail drawings for SES components, preparing for SES Design Review early in August 1985.

II. REFERENCE ENGINE SYSTEM DESIGN

The 1983 update to the RESD was completed during the first half of 1984 with development of the manufacturing cost for the V-4 concept (Figure 2-1).

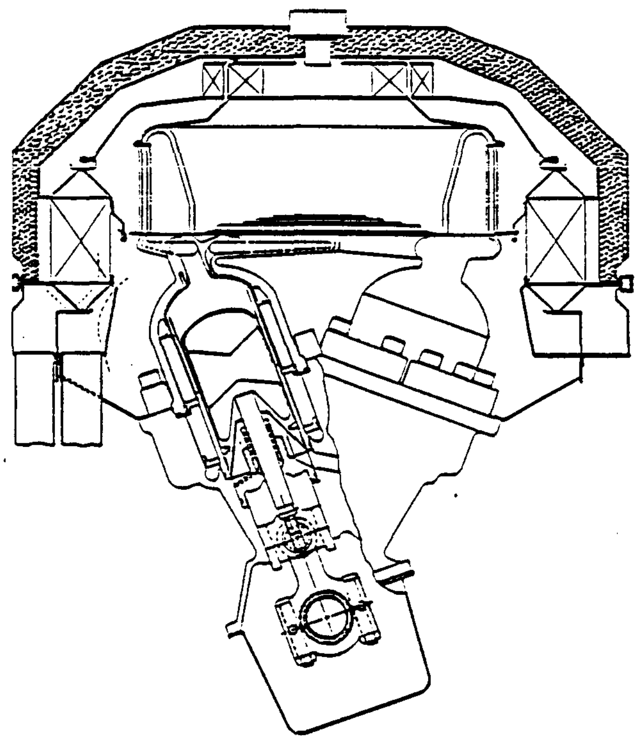


Figure 2-1 V-4 Mark V RESD

During the second half of 1984, all available program resources were devoted to the Mod II engine design effort. Therefore, no further work was done on the RESD during this report period. The next concerted effort to update the RESD will occur in 1986.

III. COMPONENT AND TECHNOLOGY DEVELOPMENT

EHS Development

Activity during July-December 1984 concentrated on completing the evaluation of CGR combustor designs and the conical nozzle in order to meet the deadline for the Mod II combustion component decision in January 1985. The testing of alternate EGR turbulator designs was also completed during July-December.

Several ceramic preheater test sections were evaluated for performance and durability in the preheater rig. Unexpected problems with the conical fuel nozzle were uncovered during the early part of 1984 and considerable effort focused on understanding and seeking solutions to these problems.

The analysis of constant volume sample (CVS) cycle emissions was extended to include predictions of CO emissions.

An attempt to relate CO emissions to tube temperature was also made. Refinements to these predictions will be made in 1985.

FUEL NOZZLE DEVELOPMENT

Development of the conical fuel nozzle continued during July-December with initial efforts concentrating on verifying the effectiveness of the nozzle design improvements (Figure 3-1).* A number of problems arose during the testing, including increased soot levels and later, increased rear-row heater-head ΔT. A concentrated effort to understand and solve these problems ensued.

Tests on engine No. 8 with the improved nozzle and BOM Upgraded Mod I EGR combustor showed no signs of fuel leakage. This implied that the new seal arrangement was performing as desired. There were no indications of melting of the seal which had previously plagued the old seal system. The slight improvement in heater-head ΔT illustrated in Figure 3-2 seemed to demonstrate the benefit of the improved atomizing airflow distribution. Soot levels, however, appeared higher than previously experienced. Soot increased uncharacteristically with load to the extent that excessive preheater plugging resulted and the engine could not be run at maximum power. While relatively high soot levels had been measured during earlier testing, they occurred at mid-fuel flows ($1.0 \leq$ fuel flow ≤ 3.0 g/s) and were not severe enough to cause excessive preheater plugging as was now the case.

An extensive test program was conducted to determine the cause of the sooting and investigate means of reducing the soot to acceptable levels. The investigation was intended to determine the contributions of the A/F control (DAFC versus K-Jetronic), fuel nozzle design and combustor design on soot levels. The tests were conducted at USAB in both the combustion rig and engine No. 3. The rig used a K-Jetronic A/F control while the engine was run with both the K-Jetronic and DAFC.

The rig tests are summarized in Table 3-1 (shown on page 3-18). As noted in the table, several combustor/turbulator and fuel-nozzle configurations were run. Figure 3-3 compares the various combustor/turbulator designs and Figure 3-4 defines the conical nozzle changes. The results showed that:

- Soot levels with the BOM EGR turbu-

*Figures can be found at the end of the section beginning on page 3-18.

lator were consistent with those measured on engine No. 8.

- Changes to the conical nozzle or even damage to the atomizing air sleeve made little difference in the soot or emissions performance.

- The more intense, concentrated swirl of the P-40 turbulator resulted in improved performance.

- Establishing an intense combustion zone close to the nozzle, as with the MTI combustor (Figure 3-3C), while beneficial to ΔT and soot, results in severe over-heating of the combustor and nozzle parts.

Engine No. 3 tests with the conical nozzle were run to determine the effect of A/F control on the soot level and to verify the improvement in soot with the P-40 turbulator. The configurations tested with the nozzle were:

- P-40 turbulator/DAFC
- BOM turbulator/DAFC
- BOM turbulator/K-Jetronic.

A summary of the soot emissions from these tests is shown in Figure 3-5. It was concluded from the test results that:

- Soot levels were improved with the P-40 turbulator to acceptable levels.

- Soot and gaseous emissions were similar with either the K-Jetronic or DAFC.

- Fluctuations in the air throttle position which, in turn, caused large swings in A/F ratio, was a major contributor to high CO and soot levels. Neither A/F control could counter these fluctuations.

- The conical fuel nozzle is more sensitive to fluctuations than the BOM nozzle. However, the sensitivity was reduced when the BOM turbulator was replaced with the P-40.

Finally, it was concluded that the soot levels produced by the conical nozzle/ Upgraded Mod I BOM combustor combination have always been unacceptable. The reason they did not appear as severe in earlier tests on engine No. 5 (previously designated Upgraded Mod I engine No. 10) may have been that fluctuations of the air throttle were minimized during these tests by limiting the air throttle position. λ variations were also minimized before data was taken.

The performance evaluation of the conical nozzle with the P-40 turbulator was continued in engine No. 5. As Figures 3-6 and 3-7 indicate, CO and soot emissions were consistent with those measured on engine No. 3. The straight-slot atomizing air sleeve also resulted in acceptable, although slightly higher, CO and soot levels (Figures 3-8 and 3-9).

The heater head temperature distribution, however, began to deteriorate during the test series. Figure 3-10 summarizes the tube temperature spreads recorded on both engines No. 3 and 5. Engine No. 3 ΔT is based on four thermocouples (T/Cs) while engine No. 5 spread is based on eight T/Cs. Of concern was the increase in ΔT at 3.5 and 4.5 g/s fuel flow from 55° and 75°C, respectively to 120° and 150°C. In general, the temperature spread with the conical nozzle is somewhat higher than with the BOM nozzle at all conditions. Surprisingly, the temperature spread of the combustion gas after the rear row (Figure 3-11) was better with the conical fuel nozzle at fuel flows above 2.0 g/s. The peak values were, however, higher than those with the BOM nozzle.

The sudden increase in ΔT is still not understood. Changing atomizing air swirl sleeves or nozzle stackup did not significantly change the level of the temperature spread. Efforts to understand and reduce the ΔT to acceptable levels will continue in 1985. The benefit of the lower atomizing airflow of the conical nozzle is still significant. Lower maintenance requirements, as a result of the nonplugging feature of the nozzle, are also desireable.

EGR DURABILITY AND AERODYNAMICS

As part of a program to improve the reliability and durability of the Upgraded Mod I EHS, several alternate EGR turbulators (Table 3-2 and Figure 3-12), and the aerodynamically modified turbulator (Figure 3-13) were evaluated during July-December. Only alternate No. 5 of Table 3-2 exhibited as good or better performance in terms of CO, soot, and ΔT than the P-40 turbulator. Alternate No. 5 performed somewhat better than the P-40 with the BOM nozzle. Overall, the P-40 appeared more suited to both conical and BOM nozzles as indicated in the CO and soot emissions of Figures 3-14 and 3-15.

TABLE 3-2
EGR TURBULATOR DESIGNS

	Alternate No.					
	1	2	3	4	5	6
Ri (mm)	0	15.00	20.00	0	15.00	20.00
D (mm)	70.00	70.00	70.00	60.00	60.00	60.00
t (mm)	1.00	1.00	1.00	1.00	1.00	1.00
b (mm)	4.91	4.96	5.00	5.13	4.91	5.00
N	36.00	32.00	30.00	30.00	24.00	20.00
B	10.00	36.60	46.85	12.00	45.81	59.53
A_2 (cm^2)	30.00	30.00	20.00	20.00	20.00	20.00
A_3 (cm^2)	38.48	38.48	38.48	28.27	28.27	28.27
A_3 (cm^2)	38.48	37.41	25.90	28.27	21.21	15.71
H (mm)	17.00	19.00	20.00	13.00	17.00	20.00

In view of the excellent performance achieved with the previous turbulators, further EGR aerodynamic development was deemed unnecessary and the recommendation to replace all Upgraded Mod I BOM turbulators with the P-40 design was made. The EGR combustor would still be used as a test bed for oxidation-resistant coatings as well as alternative material evaluation.

Figure 3-16 identifies four oxidation-resistant coatings applied to a single Upgraded Mod I BOM combustor. The combustor is scheduled for test during the first half of 1985. A preliminary investigation into the fabrication of a combustor shell out of an oxidation-resistant alloy, Cabot 214, was begun and will be completed in 1985.

CGR COMBUSTOR DEVELOPMENT

Considerable progress was made during July-December in CGR combustor development, leading to the definition of a CGR system for Mod II. Minor modifications to the 12-tube, TTE combustor (Figure 3-17) greatly improved performance. Initial test results indicated excellent CO and soot levels (Figures 3-18 and 3-19) while yielding acceptable NO_x emissions (Figure 3-20). Extensive cold flow and performance rig testing of the 16-tube, 42-tube, and ribbon combustors were performed during July-December. Initial tests of the 16- and 42-tube combustors with the internal turbulators (Figure 3-21) indicated that back pressure produced by the turbulator reduced recirculation to much lower levels than were obtained during cold-flow tests with single ejectors. After this discovery, considerable effort was made to increase the level of recirculation, and to reduce soot levels with the conical nozzle.

Table 3-3 (shown on page 3-24) summarizes the modifications which were made to both the 42-tube and ribbon combustors, and summarizes the subsequent test results. The combustors are illustrated in Figures 3-22a through 3-22e.

While none of the combustors performed nearly as well as the TTE/BOM nozzle combination, the data from configuration No. 3 was encouraging. As the soot emissions of Figure 3-23 illustrates, the reduction in turbulator diameter improved the emissions in a similar manner as the changes to the EGR turbulators previously mentioned, by increasing swirl intensity. The increased soot level at lower fuel flows was probably a result of the fuel nozzle location. As Figure 3-22c illustrates, the nozzle is far above the mixing tube exit, which results in poor mixing and rich zones as the air and fuel flow are reduced. The NO_x emissions of this configuration were increased as a result of the reduced recirculation caused by the decreased diameter. Figure 3-24 clearly demonstrates this effect. The temperature spread (Figure 3-25) was also very good and, in fact, better with the conical nozzle than the BOM fuel nozzle.

The data from the ribbon combustor tests showed low NO_x emissions, but unacceptable soot and CO emissions. This was primarily due to the large combustor exit diameter. While the external turbulator was sufficient to stabilize the flame (the mixing channels of the ribbon combustor are radial and thus, impart no swirl) it did not improve soot emissions to acceptable levels.

In view of the excellent preliminary result with the TTE combustor, CGR development in 1985 will concentrate on extensive evaluation of this combustor for Mod II. Other CGR combustor development will proceed as an RESD effort.

TRANSIENT EMISSIONS

Continued analysis of the CVS urban-cycle vehicle-emissions data led to a refinement of the HC emissions predictions as well as a prediction of CO emissions. As with unburned hydrocarbons, most of the CO is formed during start-up and the initial vehicle accelerations of the urban cycle. CO emissions during this subphase could be related to a drop in heater head tube temperature. The distribution of CO emissions as determined from the data of Table 3-4 is:

$$g/mi = 0.43\left(\frac{Yct + Ys}{Dct + Ds}\right) + 0.57\left(\frac{Yht + Ys}{Dht + Ds}\right)$$

where:

Yct = Mass emissions as calculated from the transient phase of the cold start test (80 sec warmup + Phase I) in g/test phase.

Ys = Mass emissions as calculated from the stabilized phase of the cold start test (Phase II) in g/test phase.

Yht = Mass emissions as calculated from the transient phase of the hot start test (30 sec warmup + Phase III) in g/test phase.

Dct, Ds, Dht = Measured distances during the respective phase of the test.

TABLE 3-4

CO EMISSION FROM GM CVS URBAN-CYCLE TESTING OF SPIRIT STIRLING VEHICLE

Run No.	g/mi	Yct	Ys	Yht	Yct (%)	80 s Warmup (%)	Ys (%)	Ygt (%)	30 s Warmup (%)
6008	1.87	26.16	0.88	3.81	78.6	46.5	6.2	15.2	2.8
6015	1.86	23.49	0.66	5.91	71.5	43.5	4.7	23.8	5.0
6035	1.94	23.65	1.29	5.79	68.8	42.4	8.8	22.4	5.4
6038	1.89	24.01	0.65	6.06	71.5	42.2	4.5	24.0	5.0

These terms are graphically defined in Figure 3-26.

Figure 3-27 illustrates the initial attempts to quantify the effect of tube temperature on CO. Parametric engine testing is planned in 1985 in which steady-state data will be obtained and used to refine the correlation.

CERAMIC PREHEATER

Ceramic preheater test sections manufactured by Coors were evaluated for performance and durability. Results of performance tests, which were published in the previous Semiannual, showed performance comparable to the metallic preheater. The durability and repeatability of these test sections need further development; however, the basic material (the Coors proprietary mixed-oxide material) and the design have been validated by a very successful test section. This test section was received with only a 1 to 1-1/2% leak rate, was then performance tested and subjected to thermal cycling tests simulating engine start-up and shutdown. After a total of 300 of these cycles, the leak rate remained at the low 1 to 1-1/2%. Other test sections made from the same material did not result in acceptable leakage. A meeting was held at Coors to discuss this repeatability problem and plan future efforts. It was decided to fabricate eight additional test pieces made with the mixed-oxide material and the process as close as possible to the one used to fabricate the one successful block. Also, some tooling change will be made in order to improve the producibility of the ceramic plates. A purchase order was approved and sent to Coors for eight additional blocks and the needed tooling changes. Delivery of the new tooling and equipment purchased by Coors to expedite production of the blocks is expected in April 1985, with the first of eight blocks to be delivered in June 1985.

HES Development

The primary goals of this task are to develop a manufacturable annular heater head that performs with high efficiency, develop heat transfer correlations to predict the performance of the Mod II heater head, and to complete a ceramic partition wall for testing on an annular P-40 engine.

One component critical to the success of the Mod II engine is the annular heater head. In order to reduce the risk of this component, a decision was made to begin development of this component at an early stage so as to allow sufficient time to solve any problems that might arise. These prototype heater heads will be used for fatigue testing and for flow distribution evaluation. In order to reduce the risk of the new size tubes, front row gap, and new style fins, both rig and engine tests of the Mod II geometry will be conducted. Also, an additional partition wall made from a ceramic and metallic component was fabricated in an effort to reduce heat conduction losses. This wall is to be evaluated in the P-40R engine.

Efforts during the second half of 1984 centered on prototype heater head manufacture, heater head flow testing, manufacture of heat transfer rig test sections of both Mod I and Mod II tube and fin geometry, fabrication of an Upgraded Mod I heater head with Mod II size tubes and Mod II style fins, and fabrication of the ceramic partition wall for the P-40R. Primary objectives for the first half of 1985 are the completion of the prototype heater head, rig testing of Mod I and Mod II tube and fin geometry, engine tests of the Mod II tubes and fins on the Upgraded Mod I engine, and design input to the Mod II heater head.

HEATER HEAD DEVELOPMENT

The purchase order with Howmet Turbine Components Corporation for the prototype heater head castings was placed late last year. After approximately six months of trial manufacture, the first metal casting was poured in July. By the end of December 1984, a total of 27 castings had been poured with successive improvements in the tooling and process. Significant improvements in dimensional control of the casting were made. Also, a casting was successfully fabricated without tube-hole core posts. Breakage of these core posts on previous engine heater-head castings (both Mod I and Upgraded Mod I) during the casting process has been a consistent problem, particularly for the Mod II heater head due to the long manifold arms. Work continues on these castings with fatigue samples expected early in 1985.

Fabrication of both front and rear heat-transfer test sections for the Mod I and the proposed Mod II geometry were completed. These test sections are being sent to USAB for testing in their DMR heat-transfer rig. This rig simulates the heat transfer process from the combustion gas to the heater tubes by using a double-wall tube arrangement (concentric tubes) with water flowing through the inner tube and a stagnant gas layer in the annulus. The composition of the gas (and associated thermal conductivity) determines the heat flux for a given tube temperature. Therefore, by controlling the gas composition (ranging from air to helium), the heat flux can be increased for a given heater-tube set temperature. Testing on this rig will take place early in 1985.

Fabrication of an Upgraded Mod I heater head utilizing tubes of the smaller Mod II diameter and Mod II-style fins was completed. Testing of the heater head, which will take place in early 1985, will be back-to-back with a standard Upgraded Mod I heater head, thereby determining the impacts of front row tube diameter, tube pitch, and the Mod II style fins in an engine environment. Existing heat transfer correlations will be used to compare performance, thereby validating the Mod II design.

Plastic models of prototype Mod II heater-head manifolds were fabricated using wax cores from Howmet identical in geometry to those used for metal casting. Results indicated good flow distribution between tubes, approximately equal to that of the Mod I heater head and considerably improved over the Upgraded Mod I. These results are presented in Figure 3-28.

PARTITION WALL DEVELOPMENT

The ceramic partition wall was completed and is ready for testing on the annular P-40R engine at USAB.

Materials and Process Development

The goal of this task is the utilization of low-cost, low-strategic-element content materials in the ASE, capable of surviving 3500 hr of automotive duty cycle exposure. Additionally, this task functions to provide materials-related support to the Mod II design and RESD component-development activities. Accomplishments during the second half of 1984 included:

- Completion of the report on HTP-40 alternative materials testing.
- Design and procurement of ceramic (PSZ) heater head partition walls for P-40R testing.
- Verification of heat treatment and joint design for Mod II piston rod/ piston base joint.
- Determination of room temperature fatigue properties of XF-818.

Activities planned for the next reporting period include:

- Hydraulic fatigue testing of Mod II heater head casting of XF-818.
- Determination of stresses in the XF-818 Mod II heater head castings via strain gages.
- Hydraulic fatigue testing of Mod II cast iron V-block simulating an actual engine duty cycle.
- Analysis of Mod II design for hydrogen compatability by Dr. Anthony Thompson.

HTP-40 TESTING (820°C) OF ALTERNATIVE HEATER HEAD MATERIALS

This task, which began in 1980, was completed in October 1984. The main purpose of the test was to rank the candidate heater head casting alloys, and to validate the selection of the heater tube alloy. This was to be accomplished by first accumulating engine operating exposure and then running an accelerated duty cycle to attempt to break the castings and heater tubes. The test results are reported in detail in the 1984 Contractors Coordination Meeting (CCM) and are summarized below.

Eight quadrants were manufactured, two of each candidate alloy, and one of the referenced material (HS-31). Heater tubes of the five candidate materials were used to build the quadrants according Table 3-6.

The test results ranked the candidate heater head casting alloys in the following order:

1. XF-818
2. CRM-6D
3. SAF-11

TABLE 3-6

HEATER HEAD HOUSING AND TUBE MATERIAL COMBINATIONS

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

One casting each of alloy CRM-6D and SAF-11 failed. CRM-6D sprung a leak while SAF-11 failed catastrophically by breaking into two pieces. The XF-818 casting, while developing several nearly-through-the-wall cracks, did not leak after 1600 hr of operation. The composition of each alloy is indicated in Table 3-5 (shown on page 3-25).

The heater tube selection of CG-27 for the Upgraded Mod II was continued as the best choice followed by Inconel 625. Both of these alloys performed satisfactorily in the hotter-than-normal combustor. Sections of heater tubes operated above 900°C while a maximum of 870°C was required. Alloy 12RN72 showed a severe susceptibility to oxidation and failed by oxidation-assisted creep. Sanicro 31 and 32 H both have low creep strength.

ROOM TEMPERATURE FATIGUE STRENGTH OF CAST XF-818

The room temperature fatigue properties of cast alloy XF-818 were determined at R=.5 and R=-1. This data will compliment previously determined elevated temperature tensile and fatigue property data for cast alloy XF-818.

The results of the room temperature fatigue testing are shown in Figures 3-29 and 3-30, compared to the elevated temperature fatigue strength. The fatigue endurance limit at maximum stress was found to be 375 MPa (54.4 ksi) for R=.5 and 180 MPa (26.1 ksi) for R=-1 at room temperature. These data are being used to optimize the design of the lower end of the heater head castings.

HEATER HEAD CERAMIC PARTITION WALL

This activity focused on the selection and subsequent procurement of insulating ceramic heater head partition walls to be performance evaluated on a P-40R engine. A Nilcra TS-Grade, partially-stabilized zirconia (PSZ) was selected, based on considerations of thermal conductivity, resistance to thermal shock, coefficient of thermal expansion and component availability. In all, eight partition-wall assemblies were procured with each assembly consisting of an interference fit PSZ cylinder insert inside of a stainless steel canister. These will be tested in 1985.

PISTON ROD/PISTON BASE JOINT DEVELOPMENT

This task had as its goal the development of a fatigue and leak resistant joint between the piston rod and piston base. It also served to establish the heat treatment procedure required to ensure the proper hardness conditions in various sections of the piston rod.

The piston rod and the piston base are mechanically held together by an interference fit. The seam between the two parts which results from this fit must be sealed off to prevent leakage of the hydrogen working gas. Both brazing and welding techniques were investigated as a means of creating this seal. Based on parametric evaluations of various techniques for forming this seal, Tungsten Inert Gas (TIG) Welding was found to be most effective.

The heat treatment procedure specified for the piston rod was shown to effectively produce the required core and surface hardness conditions in the piston rod. A surface hardness value of Rockwell C70 on the running surface of the rod is achieved by gas nitriding, while a core hardness value of Rockwell C50 in the crosshead-end of the rod is achieved through induction hardening.

A fatigue test program to verify the integrity of the joint design was initiated. Failure of a test specimen after 3.4 million cycles at 100% of full load was found to have been caused by high stresses, accompanied by fretting in the joint. A redesign of this high-stress area and continued design verification fatigue testing are scheduled for the next report period.

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 1984 was directed at the evaluation of PL seals and alternative rod seals in engines, in the exploratory test rig, and at the testing and development of a single-piston-ring system in the motored engine.

Rod seal and piston ring development will continue during 1985, although testing will be mainly carried out in engines.

MAIN SEALS

A summary of the seal testing conducted in the exploratory rig is given in Table 3-7.

TABLE 3-7
SUMMARY OF SEAL TESTS

Seal Set	Seal Type/ Material	Preload (lb)	Test Hr	Test Cycle
49	PL/HABIA PTFE[1]	80	100	RLDC #4[2]
50	PL/HABIA PTFE	80	392	RDLC #4
51	PL/HABIA PTFE	80	506	RDLC #4
52	PL/HABIA PTFE	80	34	Low Pressure/ Speed
53	DA[3]/Vespel SP21	80	4	RLDC #4
54	DA/Vespel SP21	80	17	RLDC #4
55	DA/Rulon J	80	51	RLDC #4

[1]Polytetrafluorolethylene
[2]Reduced life duty cycle #4
[3]Double angle

As reported in the previous Semiannual, two sets of BOM HABIA PL seals (seal sets 46 and 47) were tested in the exploratory rig using reduced life duty cycle No. 4 as shown in Figure 3-31. With seal set 46, failure due to oil leakage occurred after 236 hr. Seal set 47 completed 501 hr with essentially no oil leakage. Seal set 49, also BOM HABIA PL seals, was basically a repeat of the previous tests. Initially, both seals gave very high hydrogen leakage. This did not change significantly with operating time and the test was abandoned after 100 hr. There was no damage to the rod or seals. However, the inner surfaces of the seals which conform to the rod were not uniform and appeared to be shorter in axial length than normal; this could account for the high gas leakage. During the test there was no oil leakage past the seals.

To investigate the influence of bore diameter on seal performance, two HABIA PL seals were selected which had bores 0.015 mm smaller than the minimum specified diameter for BOM seals (seal set 50). After the initial break-in period, friction and hydrogen leakage were similar to those seen with other PL seals. The test was terminated after 392 hr when oil was detected in the seal housing. Subsequent inspection showed that the oil leakage was from the bottom seal only and that it had leaked between the seal and its seat.

In seal set 51 the seal bores were within the specified manufacturing range. The objective of the test was to determine whether the seals could complete 500 hr of testing under reduced life duty cycle No. 4. The test was terminated after 506 hr. During this period hydrogen leakage was low and no oil leakage was detected. Post test inspection showed that some oil had entered the seal housing past both seals and was present on the loading sleeves and seal seats, although the amount of oil leakage was small and not sufficient to form a pool. It appeared that the oil had leaked between the seals and the seats.

Up to this point, rig testing of main seals had concentrated on speed and pressure conditions which the seals would experience during normal Mod I engine operation. In the Mod II engine design, there is an intent to reduce the pressure and speed at idle, which raised the question of whether this might have an adverse effect on the operation of the PL seal. To investigate this a pair of HABIA PL seals were tested in the exploratory rig (seal set 52). The results are summarized in Figures 3-32 through 3-34.

At a constant speed of 400 rpm, measurements of friction force for the two seals and the seal temperature were recorded, with hydrogen pressures ranging from 150 to 500 psi. Over this pressure range, the friction force was virtually constant at ~13 lb. The variation in seal temperature was on the order of 1°C and averaged ~32°C. The low friction force and seal temperature indicate no adverse effects under these conditions.

The test was repeated at 500 rpm (this time with different oil inlet temperatures) with results very similar to those at 400 rpm. The oil inlet temperature had virtually no effect on the friction force as shown in Figure 3-33 For each case, the seal temperature was virtually constant at 3-5° above the oil inlet temperature.

In a third test, the hydrogen pressure was maintained constant at 150 psi and the speed varied up to 3000 rpm. With an oil inlet temperature of 39°C, there was little change in friction force as shown in Figure 3-34. At a higher inlet temperature (52°C), the friction force was generally lower, although more variable.

The data does not provide a complete picture of the interactions of speed, pressure, and oil inlet temperature with friction force and seal temperature. However, it does indicate that the PL seals are capable of operating normally at low speeds and pressures.

One further test was carried out to represent engine start-up conditions. The oil supply to the seals was shut off and the rig allowed to stand for ~15 hr. The rig was then started and the speed set at 400 rpm. For ~25 sec the seal friction remained at a low level similar to that seen with oil supply on. After this, the friction started to increase rapidly and there was a corresponding increase in seal temperature. The test was stopped at that point. This indicated that even after an extended stationary period, there was sufficient oil trapped in the rod/seal interface to allow the seal to operate satisfactorily for a period of time much longer than it would take for the oil supply to reach the seals during an engine start.

Seal set 53 was a potential alternative rod seal design based on the double-angle principle as shown in Figure 3-35. The seal was made from Vespel SP21. The seals gave low gas leakage; however, the test was abandoned after 41 hr of testing under reduced life duty cycle No. 4 when oil was detected in the seal housing. Inspection revealed that there had been a large leakage of oil past both seals.

In an attempt to eliminate the oil leakage the double-angle seal design was modified by increasing the inlet angle on the low pressure side of the seal. These modified seals (seal set 54) were also tested using reduced-life duty cycle No. 4. Initially, both seals gave low hydrogen leakage; however, the leakage past the bottom seal increased with time. The test was terminated after 17 hr when oil was detected in the seal housing. At this time hydrogen leakage past the bottom seal exceeded 10 cc/min while that from the top seal was only 0.25 cc/min. Inspection showed that the rod was scored and discolored where it passed through the bottom seal. This was consistent with the increase in friction force which occurred during the test. Although oil had leaked into the seal housing it appeared that the modified design had prevented an adequate oil film from being formed between the rod and bottom seal leading to failure.

The seals in seal set 55 were also double-angle seals similar to those shown in Figure 3-35. However, in this case, the seals were made from Rulon J. After 51 hr of testing under reduced-life duty cycle No. 4, oil was detected in the seal housing and inspection revealed that there had been a substantial oil leakage past both seals.

As discussed in the previous Semiannual, there was reason to believe that premature PL seal failures experienced in the Mod I engine were caused by excessive crosshead clearances which resulted from differential expansion of the aluminum crosshead guides and steel crossheads. The validity of this was demonstrated in P-40 engine tests where different crosshead clearances were used in each cylinder. These tests were carried out using the accelerated duty cycle shown in Figure 3-36 and initially using HABIA PL seals. In the first test the crosshead clearances were 40, 82, 100, and 120 µm, respectively. The test showed a definite correlation between seal failure (oil leakage) and crosshead clearance. After 270 hr of testing there was no oil leakage past the seal which had a crosshead clearance of 40 µm. However, the other three seals had leaked oil and the amount of oil leakage increased with crosshead clearance.

In the second test, a smaller range of crosshead clearances was used as shown in Table 3-8. In this test Rulon J, PL seals were used with the accelerated duty cycle. The test was continued until all four of the original seals had failed. The last seal to fail was in cylinder No. 3; the failure was due to excessive gas leakage into the crankcase. In cylinder No. 1, high gas leakage occurred after 438 hr. The rod and seal had both been scored by some unknown foreign material. A new rod and seal were installed and completed 909 hr without failure before the test was terminated. In cylinder No. 2 the first seal failed after 756 hr; in this case the failure was due to oil leakage. The seal had deformed, producing a convergent inlet on the low- pressure side of the seal. A second seal was installed and completed 592 hr without failure. In cylinder No. 4 the first seal failed after 1285 hr due to excessive gas leakage into the crankcase. A replacement seal was installed and ran for 62 hr without failure before the test was terminated.

TABLE 3-8
P-40 ENGINE NO. 9 CROSSHEAD CLEARANCES

	Cylinder			
	1	2	3	4
Crosshead Clearance (µm)	18	41	52	65

As previously reported, an alternative rod seal design (a pumping ring seal shown in Figure 3-37) demonstrated good performance in rig testing. To give a comparison with the PL seals, four pumping ring seals made from Rulon J were installed in P-40 engine No. 9 with the same crosshead clearances given in Table 3-8 and run to the accelerated duty cycle. The total duration of the test was 1342 hr during which time the seal in cylinder No. 2 did not fail. In cylinder No. 1 oil leakage was detected after 407 hr. This was the result of damage to the lower external O-ring incurred during assembly and was not a primary seal failure. The same fault occurred with the replacement seal after an additional 744 hr. In cylinder No. 4 oil leakage was detected after 407 hr due to wear of the brass washer located under the base of the seal, which was probably the result of the large crosshead clearance. The washer and seal were replaced and ran for an additional 666 hr before oil leakage occurred due to wear of the washer. In cylinder No. 3 oil leakage occurred after 1073 hr; however, in this case there was no obvious explanation. In comparison, the performance of the pumping ring seal and the PL seal are essentially the same.

To confirm that excessive crosshead clearance was the major cause of premature seal failures in Mod I engines, Mod I engine No. 7 was assembled with steel crosshead guides (instead of aluminum) to eliminate the effects of differential expansion. Apart from the change in material, the crosshead guide dimensions conformed to the BOM dimensions (actual crosshead clearances are given in Table 3-9). HABIA PL seals were installed in cylinders No. 1 and 2. Rulon J PL seals were installed in cylinders No. 3 and 4 and the engine was run with the accelerated duty cycle. The test was terminated after a total of 2158 hr when a regenerator failure caused extensive contamination of the engine and damage to components. Prior to this the seals in cylinders No. 1 and 3 had operated without failure. In cylinder No. 4 oil leakage was detected after 411 hr. The oil leakage was the result of deformation of the seal where it had been extruded between the piston rod and the seal seat. Subsequent inspection revealed that the bore of the seal seat was oversize. This had accelerated the extrusion process and therefore the oil leakage cannot be classified as a primary seal failure. A new seal and seat were installed and the replacement seal ran for an additional 766 hr before oil leakage occurred again. This was classified as a primary seal failure since there were no apparent extraneous factors involved. A new seal was installed and ran for an additional 980 hr before the test was terminated. In cylinder No. 2 the original seal ran for 1590 hr before oil leakage was detected. Inspection indicated that this failure was due to an oversize seal seat. The replacement seal completed the remainder of the test period, 658 hr, without failure.

TABLE 3-9

CROSSHEAD CLEARANCES FOR MOD I ENGINE NO. 7 WITH STEEL CROSSHEAD GUIDES

	Cylinder			
	1	2	3	4
Crosshead Clearance (μm)	40	51	45	43
PL Seal Material	HABIA	HABIA	Rulon J	Rulon J

PISTON RINGS

Piston ring testing during the second half of 1984 was concentrated mainly on the Mod I motoring rig. The major effort was devoted to evaluating a single-solid piston ring design shown in Figure 3-38. The solid uncut piston ring made from Rulon LD is mounted in the groove on a single quad ring which applies an outward radial load to the piston ring and seals the potential leak path behind the piston ring. The piston/quad-ring system was designed to fit into grooves with the same dimensions as the Mod I split-solid piston ring grooves. In the assembled condition the nominal quad-ring squeeze was 11%. At all speeds and pressures greater than 3 MPa the motoring power with the single-solid ring was significantly less than the baseline (split-solid rings) as shown in Figure 3-39. The maximum difference in recorded mean cycle pressures was 0.33 MPa, although for the majority of cases, it was less than 0.2 MPa as shown in Figure 3-40. Overall, the differences in cycle mean pressures with the single-solid ring were less than with the baseline split-solid piston rings (see Figure 3-41). After the initial performance measurements the rig was motored at 2000 rpm/9 MPa for 50 hr and the measurements were repeated. There was no significant change in performance over this period. To investigate consistency of performance, the first set of single rings were removed and a second set of rings of the same design was installed. These rings reproduced the performance of the first set over a period of 32 hr of motoring at 2000 rpm/9 MPa. At this point a third set of rings of the same design was installed. Initially these also reproduced the performance of the orignial set of rings. The rig was motored for a total of 250 hr mainly at 2000 rpm/9 MPa with periodic performance checks. During this period there was a slight increase in motoring power as shown in Figure 3-42; however, the cycle-to-cycle pressure differences remained low throughout. When the rig was disassembled, oil was found in two cycles and this had reached the piston rings. The wrist pin in one crosshead had been displaced and the end of the pin had been rubbing against the crosshead guide which could account for the increased motoring power.

The test data indicated that the single-solid piston rings gave a consistent reduction in motoring power which was maintained in the short term. The next phase of testing was aimed at determining the durability of the single-solid piston rings.

The plan was to first carry out a 500 hr baseline motoring test with the BOM split-solid piston rings measuring performance at intervals during the test. For the majority of the time the rig would be motored at 2000 rpm/13 MPa. The test would then be repeated with single-solid piston rings. Prior to these tests, due to life limitations, it was necessary to completely rebuild the rig, replacing the existing rolling bearing crankshaft system (lightweight reduced friction drive) with conventional journal bearing crankshafts. At the same time, the Data Acquisition System (DAS) was upgraded by adding four pressure transducers, one in each of the dummy heads and a shaft encoder connected to one of the crankshafts. The outputs from these were input to a microcomputer through appropriate interfaces. With this facility the pressures in the individual cycles and their phase relationships relative to the crankshaft reference could be accurately measured and recorded. From this information it would be possible to determine whether changes in motoring power were due to friction only or due to changes in leakage which would effect the cycle pressures.

After rebuilding the rig with split-solid piston rings, a new baseline was established as shown in Figure 3-43. Following the baseline test a 500 hr endurance test was initiated. During this period the rig was motored mainly at 2000 rpm/13 MPa with abbreviated performance checks at regular intervals. 150 hr into the test a crack developed in the crankcase; the rig was rebuilt with a new crankcase, although all the other hardware including the piston rings were retained. A check carried out immediately after indication of the rebuild had no effect on the motoring power. Figure 3-44 compares the motoring power at the end of the 500 hr test with the initial data. For all conditions there was a significant increase in motoring power during the test and in the worst case this was more than 1 kW. The differences in mean cycle pressures are shown in Figure 3-45. During the first 100 hr there was a significant decrease in the pressure differences; however, the differences increased again just prior to the discovery of the crankcase crack. After the rebuild the pressure differences were lower and remained at approximately the same levels up to 450 hr when there appeared to be a trend for increasing pressure difference. There is no obvious reason why a crack in the crankcase should effect the cycle-to-cycle pressure differences. During the test, wear of the split rings was minimal, 2-4% by weight. Wear of the solid rings was greater, 8-13% by weight. A similar 500 hr endurance motoring test of single-solid piston rings has just started.

The single-solid piston ring is also being evaluated in Mod I engine No. 5. To provide back-to-back data a new set of BOM piston rings was installed, run-in, and a complete performance map was generated. The single-solid piston rings were then installed. Except for the necessary change in piston bases, all other hardware was the same as for the previous test. After running in the single-piston rings a new performance map was generated. The power developed with the BOM piston rings and single-solid piston rings are compared in Figure 3-46. At the higher speeds and pressures the engine developed more power with the single-solid rings. At lower speeds and pressure the differences are in the same order as the measurment errors, although there is no indication that the single-solid rings degraded the performance under these conditions. Following the baseline test the single-solid rings were left in the engine while other development testing took place. The rings have now completed a total of 90 hr of operation with consistent performance.

Engine Drive System Development

The primary activities taking place under this task are the durability test rig and the manufacture of a ductile cast-iron engine V-block. The durability test rig is a preliminary version of the Mod II Cold Engine and Drive System (CEDS). It features most of the designs planned for the Mod II actual engine. The cast V-block is important because of its complexity; considerable effort will be required to manufacture a block which will meet the engine requirements.

Procurement of hardware for the durability test rig was started in April 1984. The original plan was to have the test rig up and running by September 1984. However, hardware design was not completed until October 1984. Due to problems with the V-block casting and crankshaft material quality, rig operation will be delayed until March 1985. The original intent of this test rig was to evaluate the design ideas prior to actual Mod II engine design in early 1985. This now is not possible, however, it is believed that rig operation is still important, to reveal problems prior to actual engine test. It is hoped they can be corrected prior to actual engine operation, which is scheduled for early 1986.

The manufacture of the ductile iron V-block was started in March 1984, after completion of tooling and the first pour. Despite the use of "chills" and core coatings, porosity in the center portion of the block could not be removed (see Figure 3-47). In order to furnish a block for the durability rig, an early pour was accepted with the understanding that it might be porous. This was done to gain machining experience on the block. Meanwhile, development of the casting to remove porosity was continued. The problem was not eliminated and activity to find an alternate supplier with more resources available to work on the problem was in progress. Deere has offered to review the existing tooling and, if acceptable, will try to pour a good casting.

If the present vendor is unsuccessful, this offer will be accepted and a new effort will begin early in 1985, on the preliminary block initially, to be followed by the final Mod II engine block.

The third drawback in the rig schedule has been the crankshaft (see Figure 3-48) material properties. Consultations with Mercury Marine originally helped define a specification for the crankshaft which doubles as the inner race for the rolling-element main and connecting rod bearings. The material was ordered and used, but left the final parts with several surface inclusions. It was surmised that these unacceptably large inclusions occurred because the intermediate forging process normally employed by Mercury Marine (which breaks up and reduces inclusions) was not used for these shafts. However, the crankshafts will be used in the rig to determine life characteristics of the rolling element bearings and other design features related to the Mod II.

Controls System/Auxiliaries Development

The major goals of this task include the development of the engine control and auxiliaries sytems. Specific control systems goals include the development of a highly flexible DAFC with a low combustion-air pressure drop and low minimum fuel flow, and development of a simplified mean pressure control that does not require a servo-oil actuator. A specific goal for the auxiliaries task includes the development of a high-efficiency combustion air blower. The hardware designs for both tasks must be compatible with the extremes of an automotive operating environment.

Development during the second half of 1984 focused on integrating the DAFC into the engine system, improving the λ schedule, decreasing head temperature excursions during transients with software improvements and working-gas temperature measurements, and improving idle performance.

PCV improvements were also addressed by decreasing the dead band and beginning the development of an electrically actuated PCV.

Auxiliary improvements are expected to contribute heavily to additional fuel economy. Therefore, a new blower configuration is being implemented, since the blower is a major energy absorber.

COMBUSTION CONTROL

The DAFC was configured into a reliable control in the test cell for Upgraded Mod I engine No. 8. 175 running hours (varied) were completed from the end of August through the beginning of October. Also, the DAFC was installed on USAB's EHS test engine No. 3 by MTI and USAB personnel, and in the Spirit by MTI personnel. Twelve sets of DAFC electronic hardware have been built and tested and the software and hardware have been modified so that the HITACHI airflow meter can be used with the DAFC. A check of the λ curve by emissions was accomplished in August with Upgraded Mod I engine No. 8. With a correct calibration of the pump/nozzle, and the values plugged into the DAFC fuel map, the λ by emissions came within 5% of λ set. The DAFC operated adequately with the BOM and improved prototype conical nozzles in the test cell.

The DAFC λ control and an anticipatory algorithm have been evaluated and found to be repeatable. Tests indicated up to a 94% reduction in CO with the anticipatory algorithm (see Figures 3-49 and 3-50).

Intermittent fuel flow of 0.21 g/s average consumption (on for 24 sec at 0.6 g/s, off for 44 sec) was attained. Operating conditions were 400 rpm, 3.0 MPa, 810°C, and 1.3 kW (1.74 hp) out. The same conditions can be supported by continuous flow with no loss in fuel economy. During subsequent testing, operating conditions were moved down to 400 rpm, 2.4 MPa, 813°C, and 0.9 kW (1.2 hp) out, thus providing a continuous fuel consumption of 0.18 g/s. In addition, the DAFC, nozzle, and combustor operated reliably.

A counter circuit for the DAFC micropump motor was designed, fabricated, installed, and checked out in the Spirit. The counter will allow for use of a totalizer to keep track of the total amount of fuel used during different test runs on a chassis dynamometer or over-the-road.

Transient tests on working-gas temperature response were conducted on engine No. 5. The tests showed that the working-gas T/Cs indicated the temperature droop 30-40% faster than the rear tube T/Cs (Figure 3-51). The proportional control for the speed variator has been successfully tested on the engine in the test cell and in the Spirit using a flapper-type, single-stage servo valve. For good blower speed control, whether by "bang-bang" control or proportional control, the excessive amount of "stiction" in the speed variator must be reduced significantly, and a servo-oil pump must be selected that will supply sufficient oil at idle for the proportional servo valve to operate properly.

MEAN PRESSURE CONTROL

A design modification of the existing Upgraded Mod I PCV was initiated during August to reduce the deadband (neutral zone) of the valve. The current valve exhibits a typical deadband on the order of 30-35% of the total valve travel. A simple change in the design (requiring changes in two pieces) reduced the deadband by two-thirds. The total travel of the valve was also reduced by the same amount, thus maintaining the mechanical gain of the valve (expressed in flow area opened per unit of travel). The modified valve works exactly as designed and is leak tight.

During September, tests were conducted for check-out of the new control algorithms for low-speed and low-pressure idling, pump-down tests, and down-transient tests. The new control algorithm worked as expected, providing the capability for stable idle operation down to 400 rpm and 30-bar P_{mean} (2.4-MPa mean pressure). The steady-state idle fuel flow at this point is 0.18-0.20 g/s, with 820°C tube temperature.

Pump-down tests revealed that the Mod I hydrogen compressor is capable of pumping the engine down to this pressure at 400 rpm, with 200-bar tank pressure; however, the pumping time is lengthy. Down-power transient testing revealed that short-circuiting control of the engine as plumbed in the cell is inadequate above 2000 rpm.

Reduced idle speed and pressure testing was also done in the Spirit. Both steady-state and over-the-road testing were done. An idle speed of 500 rpm and pressures of 30 to 40 bar (P_{mean}) was tested. Idle stability was good and driveability was fair to good. The data from Spirit engine No. 8 steady-state idle testing were reduced, as shown in Figures 3-52 through 3-54.

In October, the Spirit was driven for the first time with a four-speed manual transmission. Response of the controls was unsatisfactory, with no changes to the existing logic, hardware, or setpoints. Performance and driveability of the vehicle with engine speeds less than 2500 rpm are, however, now acceptable, although some improvements in stability at low engine loads is still needed. At engine speeds greater than 3000 rpm, the mean pressure control is unable to completely control the engine, and excessive engine speeds occur during high-power gearshifts. The problem appears to be due to inadequate short-circuiting flow. The reduced deadband, PCV and larger diameter hydrogen plumbing were installed, with significant improvements in controllability and over-speeding.

As a result of the Mod II Technology Assessment, the electrically actuated PCV (improved Upgraded Mod I valve) was selected. An electrically actuated PCV, similar to the USAB rack and pinion design (2-18340) was ordered, and the necessary drive electronics were designed, built, and tested. Testing of the electrically actuated PCV was started and shows promising results.

DEC

Development proceeded for improvements and modifications of the DEC. Final components were selected for the cold junction thermistor circuitry. Modifications to the bench system were completed to interface with LVDT and control circuits used with the electric PCV, and modifications to the cold junction RTD, oil temperature RTD and the +10 VDC reference circuits were detailed.

A DEC was installed in the USAB's EHS test cell on engine No. 3 and was run successfully with that engine.

Software modifications for engine No. 3 at USAB which include two additional engine shutdown guards (water and oil over-temperature) were completed and shipped.

All cabling from the DEC to the engine for Deere was completed in August, and the DEC and all transducers were operating normally.

MOD II ELECTRONICS DEVELOPMENT

An improved igniter power converter circuit has been designed, tested, and installed in the Spirit, combustion test cell and engine test cell. All igniter circuits are currently functioning properly. All components for the printed-circuit-board version of this igniter circuit are on order and on schedule.

Alternate-source pressure transducers have been received from Data Instruments and are currently under test. These pressure transducers are less costly, more readily available, and require no re-adjustments of electronics when replaced, as do the current Bell and Howell transducers.

A Mod II DEC block diagram has been produced and distributed. This diagram is a top level diagram defining the Mod II control system configuration.

MOD II CONTROL SYSTEM ANALYSIS

Documentation of the DEC logic continued with the completion of two more control modules, thus bringing the total of completed numbers to 13 out of 26. This more than completes half of the task since the remaining modules are the smaller and less complex modules.

An upgrade to the PCV module is underway. This upgrade provides additional flexibility in the tuning of the power-control system to the manual transmission.

COMBUSTION AIR BLOWER

The lobe blower was determined not to be sufficiently developed for the first build of the Mod II engine. However, work will continue to complete mapping of the best set of hardware of the prototype design with a reduced noise housing (modified porting), since noise is still a problem with the blower. A frequency analysis of the noise showed that the frequency distribution is independent of speed, and the sound level is a function of speed and stroke (i.e., airflow rate).

A modified housing has been completed and a blower will be rebuilt with this modified housing, lower piston seal preloading, and improved cage-to-housing sealing, and will be remapped to 3000 rpm at full eight-part stroke.

During the Technology Assessment it was decided that the prime Mod II blower will be an electrically-driven centrifugal blower combined with a Mod I impeller for an EGR combustor, and a redesigned impeller (if necessary) for a CGR combustor (which is the prime choice). The backup system will be a direct-drive centrifugal unit to be designed at USAB.

Projected performance characteristics/requirements for the Mod II blower ΔP versus airflow and power requirements have been defined. Blower and auxiliary load requirements data were defined for the Mod II alternator and electric blower designs. Requirements for a 3:1 speed increaser were established. Analysis of vehicle CVS cycle performance at GM Research Laboratory showed that the ratio of blower speed to engine speed varied from 7:1 to 10:1. Minimum electric drive requirements for the blower were set. Aeroflex Laboratories Incorporated has been contacted concerning the possible use of their brushless permanent magnet DC motor for the Mod II blower requirements.

After review of the pros and cons of a 12 versus 24 V electrical system it was decided that a 12 V system will be used and the alternator will supply 12 VDC for the vehicle and control systems, as well as a higher voltage for the blower motor.

The Mod II, 32,000 rpm alternator design entered the drafting stage. A traction drive planetary ball speed increaser design was evaluated. For prototype evaluation purposes a production traction drive planetary ball speed increaser was ordered from Paxton Products Incorporated.

Alternator drawings were completed and are being reviewed. Rotor material and bearings were received and stator laminations stamped. The maximum output voltage of the alternator has been set at 60 VDC to provide an estimated 82% efficiency.

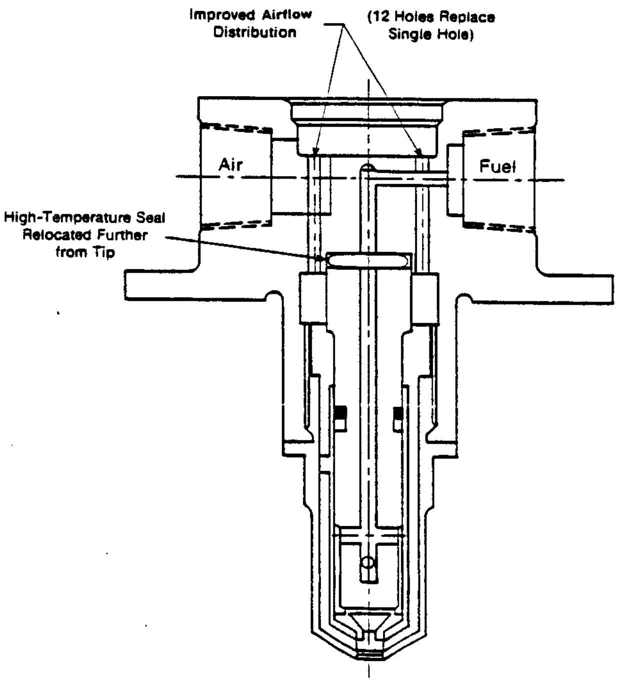


Figure 3-1 Design Improvements to Conical Fuel Nozzle

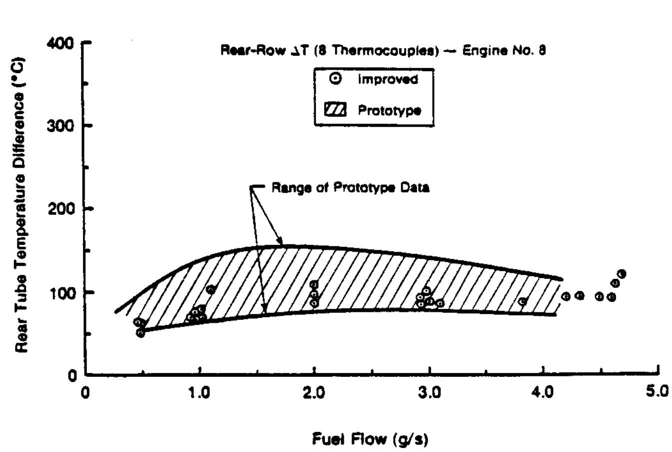
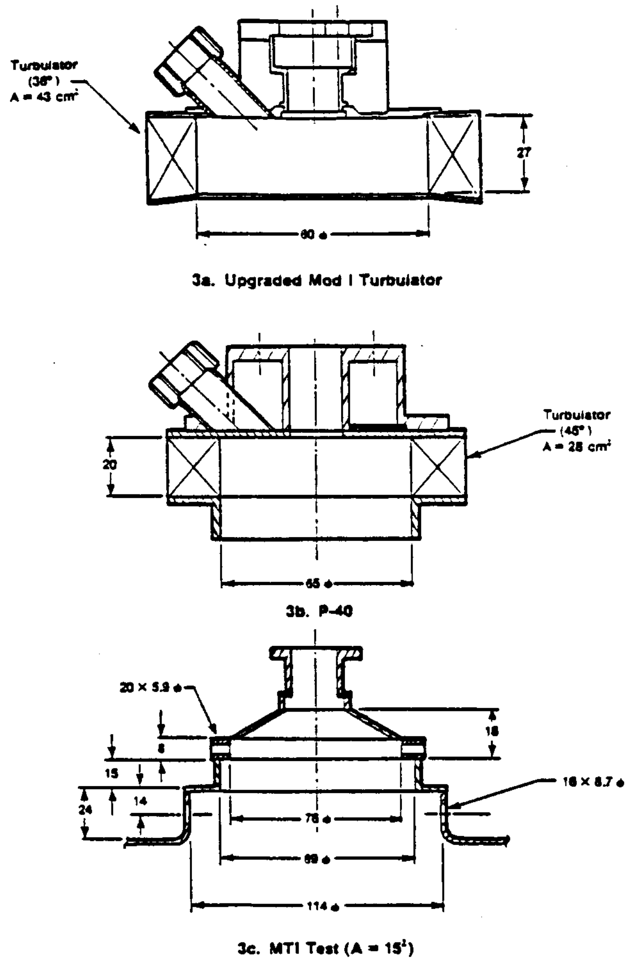


Figure 3-2 Effect on Heater Tube ΔT of Conical Nozzle Improvements

TABLE 3-1
RESULTS OF CONICAL NOZZLE SOOT INVESTIGATION

Date	Rig Test No.	Combustor	Nozzle	Comment
10/03/84	58-89	Mod I P-40	BOM	Soot levels similar to engine - Bacharach = 0-1
10/04/84	90-105	Mod I P-40	Prototype conical -30° cone (conical #1)	Soot level higher than BOM in mid-fuel range: 1.5-2.5 g/s, however, in general, acceptable. Good at high fuel flows. 2.0 g/s at λ = 1.2 only high level. Other emissions good.
10/04/84	106-114	Mod I P-40	Improved atomizing airflow distribution. New seal, otherwise, as above (conical #2)	Similar trend in soot levels. Peak at 2-2.5 g/s, even at high λ. CO and other emissions good.
10/05/84	115-139	Mod I BOM	Conical #1	Soot level higher than with P-40, however, no extremely high levels. At high fuel flows soot increases with decrease λ and low fuel flow without EGR, as expected.
10/08/84	140/155	Mod I BOM	BOM	Similar trend in soot level as engine data. Generally lower than conical for given λ.
10/08/84	156-170	Mod I BOM	Replaced damaged atomizing air swirl sleeve and fuel spin. 35° cone prototype body (more stable spray (conical #3)	Soot levels similar to conical #1. As were other emissions.
10/08/84	171-135	Mod I BOM	Conical #4 - replaced atomizing air swirl sleeve with sleeve having 10 straight slots, otherwise, same as #3	Soot level, again same as before, except for unexplainable high level (Bacharach = 6) at 2 g/s, λ = 1.3.
10/09/84	186-199	MTI	Conical #4	Soot levels initially better than Mod I combustor, however, then increased with time. Nozzle/combustor overheated. Front tube row much hotter than before.
9/10/84	200-203	MTI	Conical #3	Same trend as with conical #4. Combustor found to be severely damaged - overheated and distorted.



3a. Upgraded Mod I Turbulator

3b. P-40

3c. MTI Test (A = 15²)

Figure 3-3 Comparisons of EGR Turbulators Used During Conical Nozzle Soot Investigation

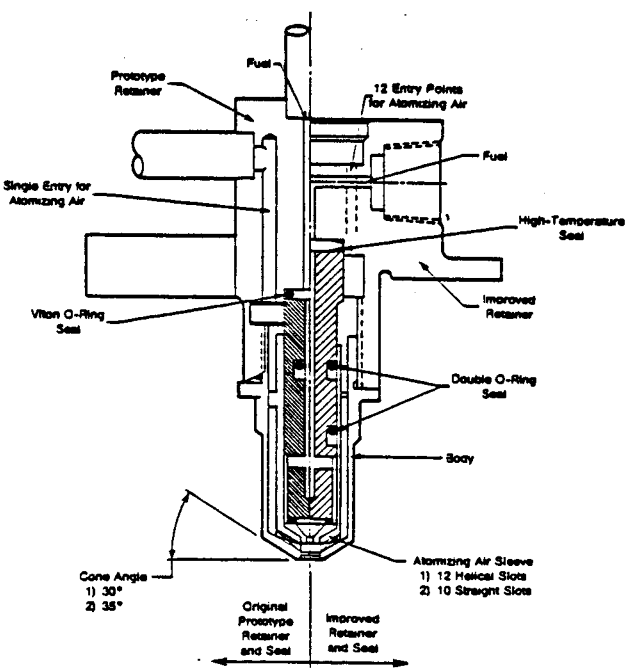


Figure 3-4 Changes to Conical Nozzle During Soot Investigation

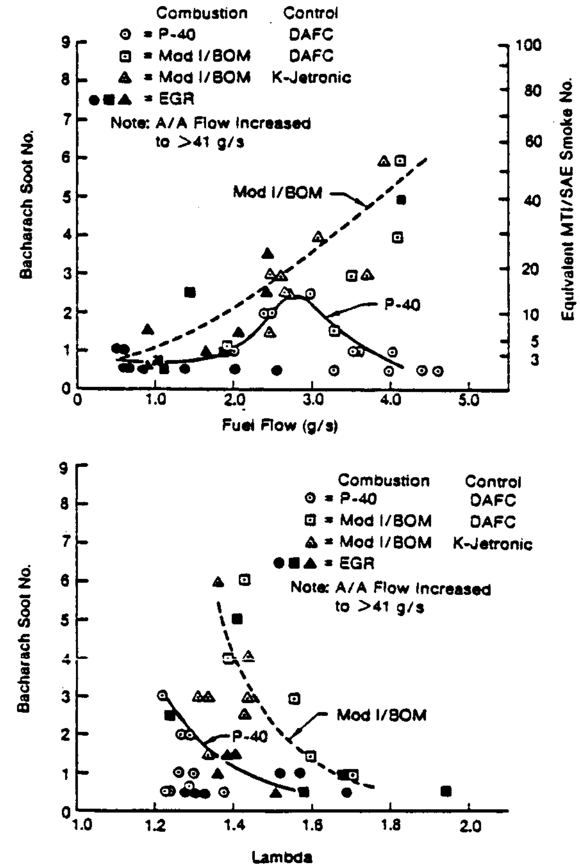


Figure 3-5 Engine No. 3 Conical Fuel Nozzle Soot Level

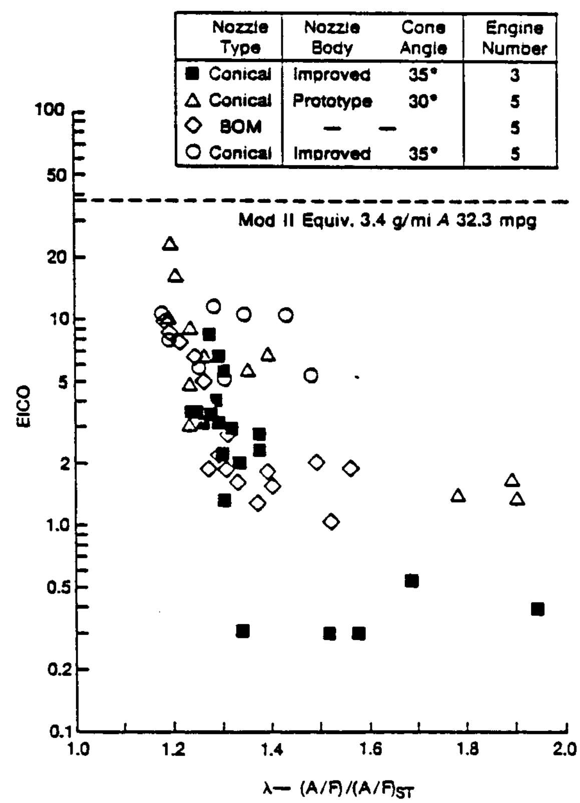


Figure 3-6 Conical Nozzle CO Emissions, Engine No. 5 Tests

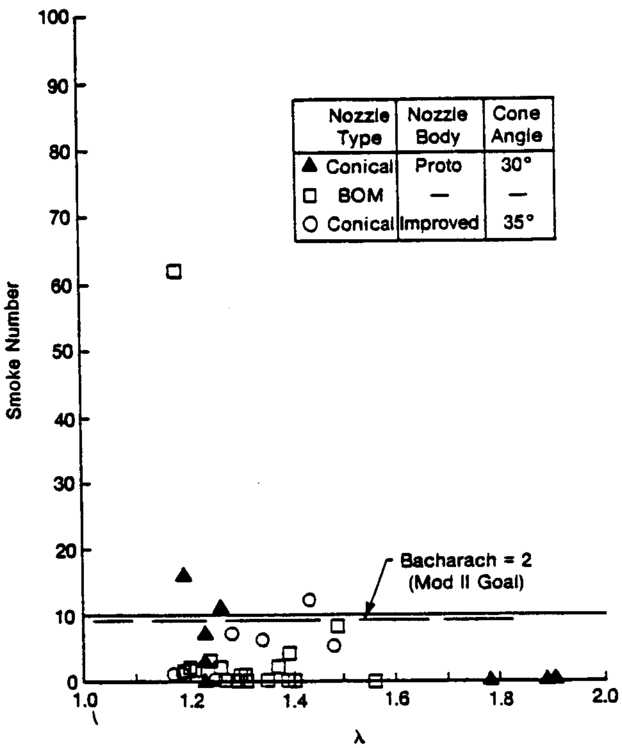


Figure 3-7 Conical Nozzle Soot Emissions, Engine No. 5 Tests

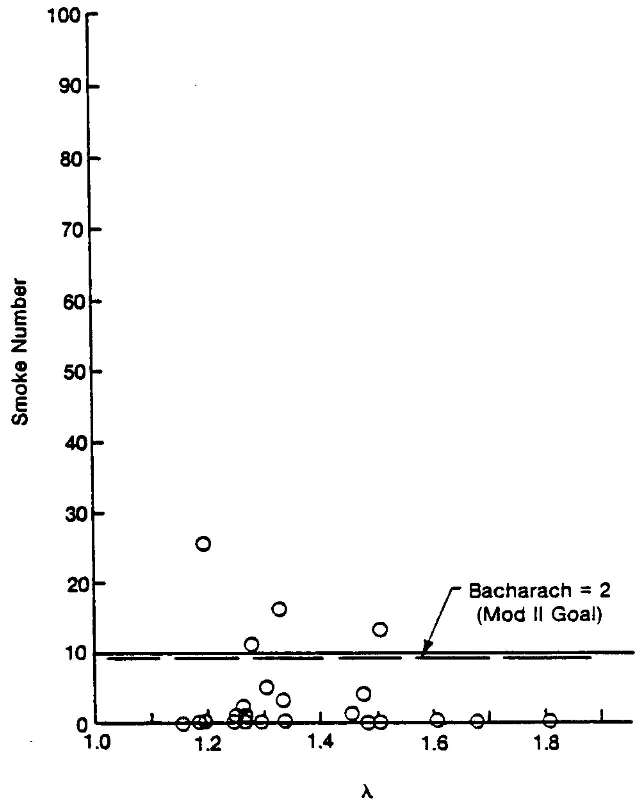


Figure 3-9 Soot Emissions of Conical Nozzle with Straight-Slot Atomizing Air Sleeve

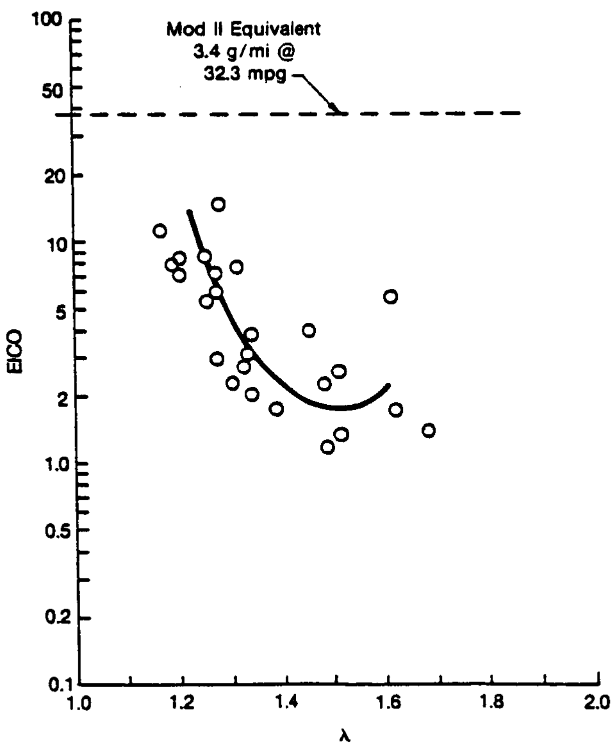


Figure 3-8 CO Emissions of Conical Nozzle with Straight-Slot Atomizing Air Sleeve

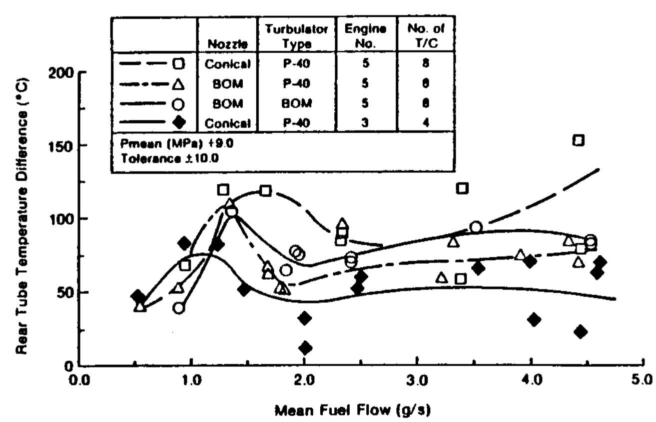


Figure 3-10 Heater Head Rear Row Temperature Spread (T_{max} - T_{min}) During Conical Nozzle Testing

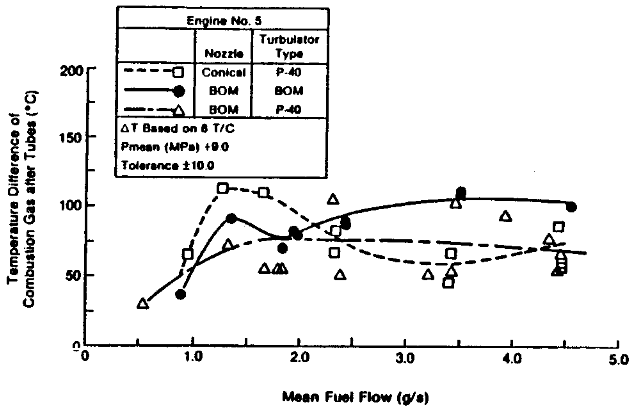


Figure 3-11 Temperature Spread, Combustion Gas After Rear Row, for Conical and BOM Fuel Nozzles

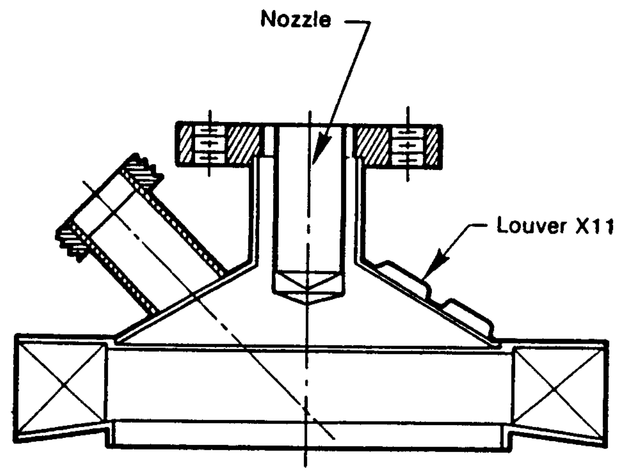


Figure 3-13 EGR Combustor Turbulator, Modified Aerodynamic Design

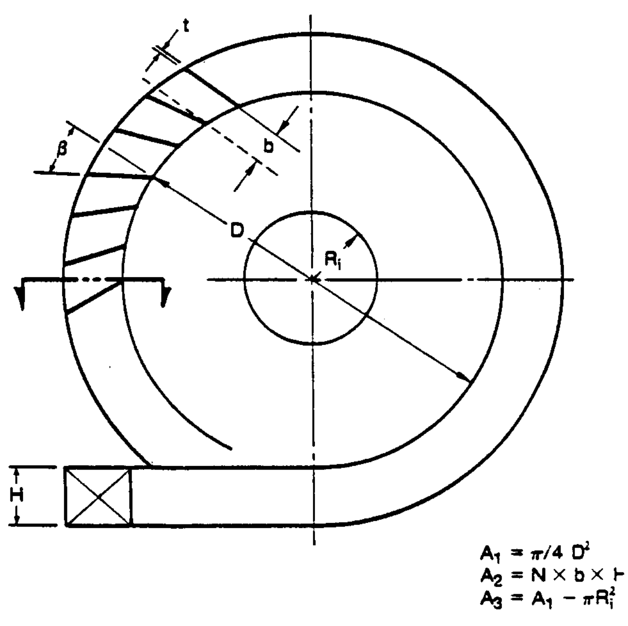


Figure 3-12 Definition of Turbulator Parameters

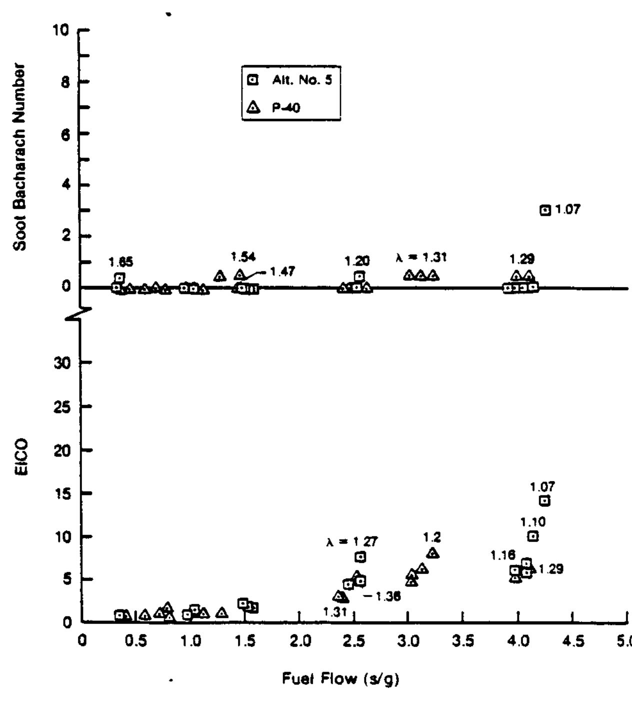


Figure 3-14 Comparison of P-40 and Alternate No. 5 Emissions with BOM Fuel Nozzle (A/A = 0.7 bar)

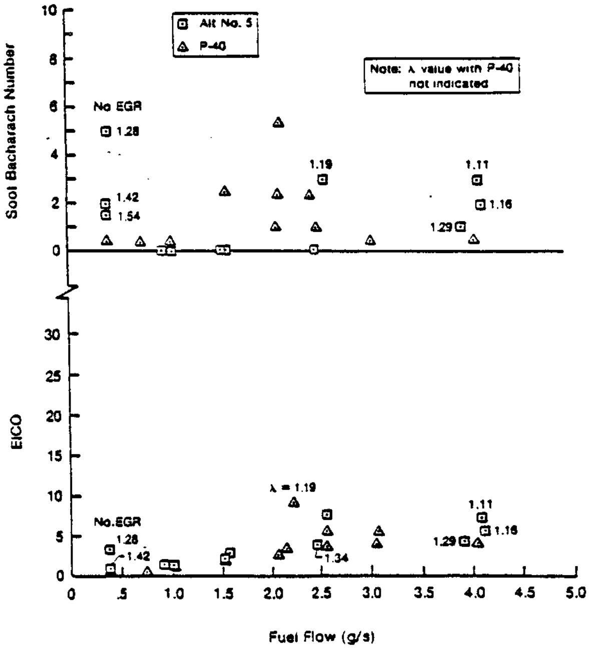


Figure 3-15 Comparison of P-40 and Alternate No. 5 with Conical Fuel Nozzle (A/A = 0.36 bar)

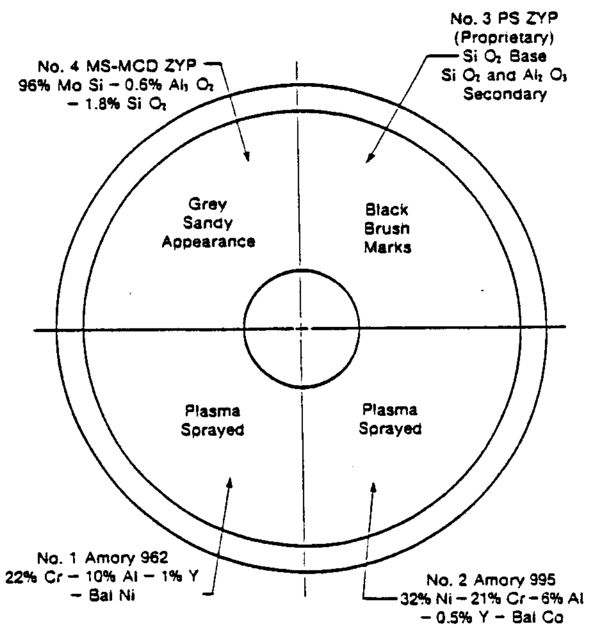


Figure 3-16 Identification of Four Oxidation Resistant Coatings for Combustors (view from bottom)

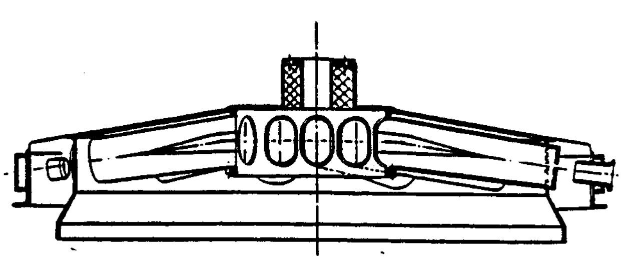


Figure 3-17 Tubular-to-Elliptical CGR Combustor

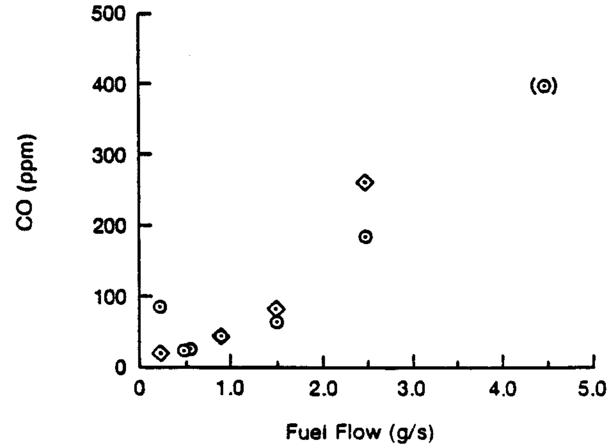


Figure 3-18 CO versus Fuel Flow Engine No. 10 (λ = 1.25) TTE Combustor

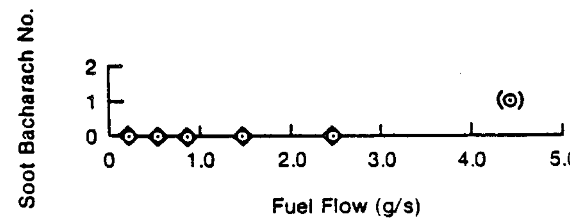


Figure 3-19 Soot versus Fuel Flow Engine No. 10 (λ = 1.25), TTE Combustor

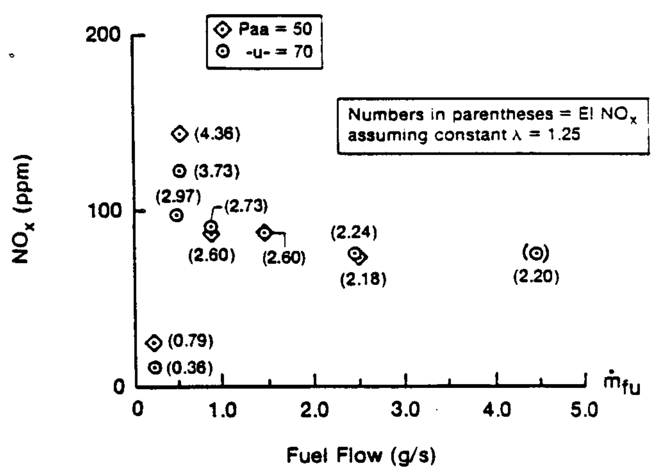


Figure 3-20 NO_X versus Fuel Flow, Engine No. 10 (λ = 1.25), TTE Combustor

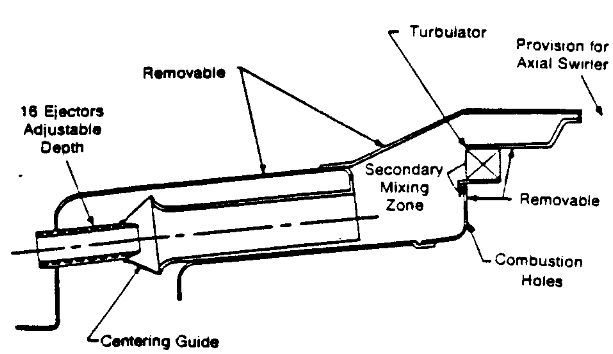


Figure 3-21 Tubular CGR Combustor Concept with Internal Turbulator

TABLE 3-3

PERFORMANCE RIG TEST SUMMARY OF 42-TUBE AND RIBBON CGR COMBUSTOR CONFIGURATIONS

Config. No.	Combustor Designation	Modification	Fuel Nozzle	
1	42-Tube	None (Figure 3-22a)	Conical	High soot, low recirculation
2	42-Tube	Internal turbulator and mixing holes removed (Fig. 3-22b)	Conical	50-60% recirculation - high soot, decreasing with increased airflow
3	42-Tube	21 vanes added at 30° swirl angle. Diameter reduced to 89 mm (Fig. 3-22c)	Conical	Good performance at 2.0 g/s fuel and above. Excellent rear-row ΔT
4	42-Tube	External turbulator, otherwise same as Configuration #3 (Fig. 3-22d).	Conical	Flame confined to external turbulator region causing overheating of combustor in CGR vane region, high soot and ΔT.
5	42-Tube	Same as #3	BOM	Some soot, high ΔTs - combustor flameshield distored.
6	42-Tube	Same as #3	Conical-inserted 0.024" further	Results not as good as in #3. Again, combustor found distorted - major cause of problems.
7	Ribbon	No internal turbulator external turbulator with 45° swirl 192 mm exit diameter (Fig. 3-22e).	Conical	Sooty, however, improved with airflow.
8	Ribbon	Same as #7, however, leak around ejectors repaired.	Conical	Similar to #7. ΔP was lower. Large leakage found around external turbulator.

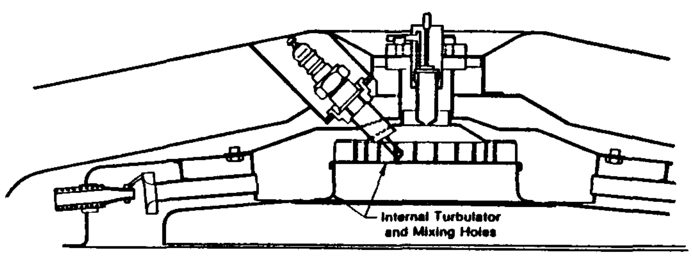


Figure 3-22a Initial Configuration of 42-Tube CGR Combustor

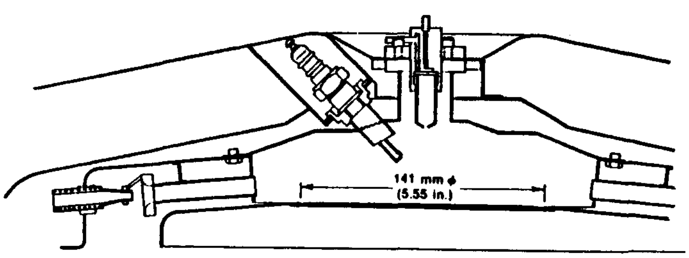


Figure 3-22b Ribbon CGR Combustor with Internal Turbulator and Mixing Holes Removed

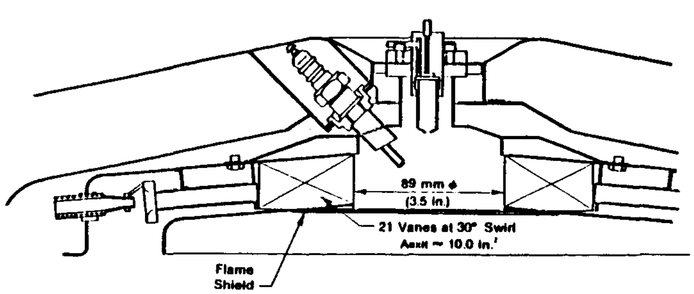


Figure 3-22c 42-Tube CGR Combustor with Swirl Vanes Added at 30° Angle

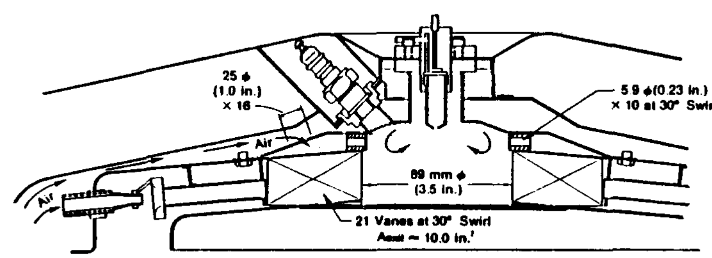


Figure 3-22d 42-Tube CGR Combustor with Swirl Vane and External Turbulator Added

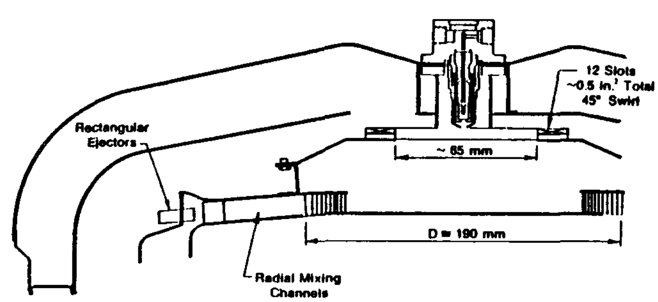


Figure 3-22e Ribbon Combustor with External Turbulator, 45° Swirl Angle

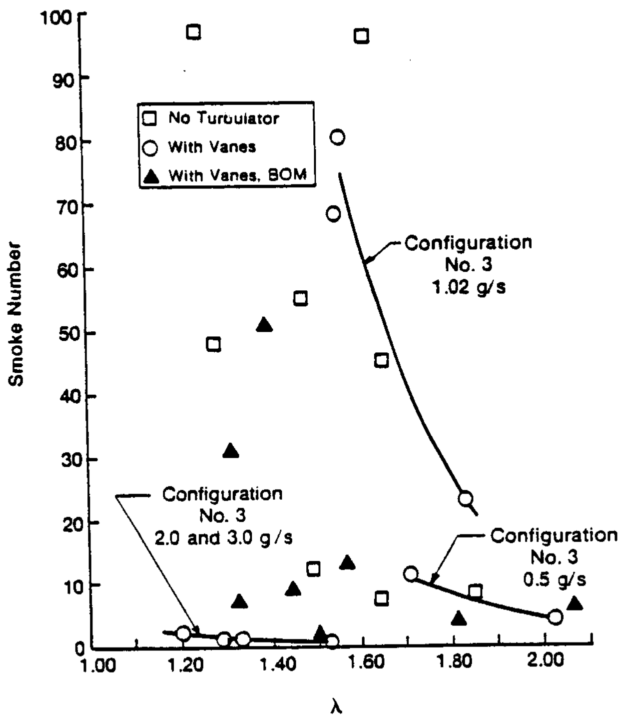


Figure 3-23 Soot versus Lambda (Rig Performance) 42-Tube Combustor, Configuration No. 3

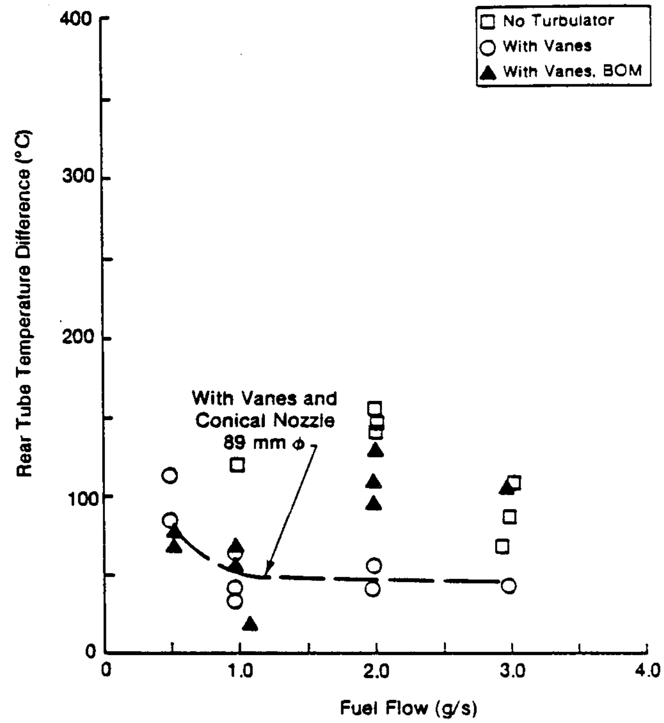


Figure 3-25 ΔT versus Fuel Flow (Rig Performance) 42-Tube Combustor, Configuration No. 3

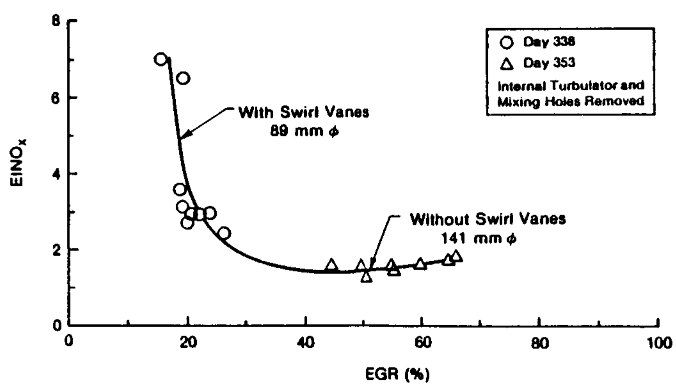
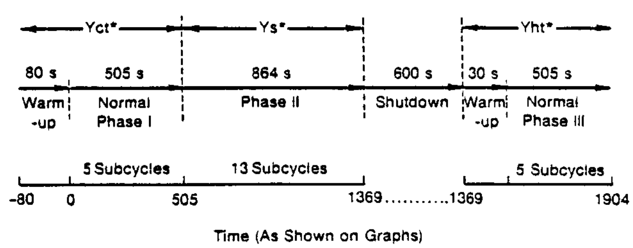


Figure 3-24 Effect of Reduced Recirculation on NO_x Emissions, 42-Tube CGR Combustor, Configuration No. 3.



*See text for definitions
A subcycle is defined as an acceleration, cruise, and deceleration.

Figure 3-26 CVS Urban-Cycle Dynamometer Test as Conducted at GM

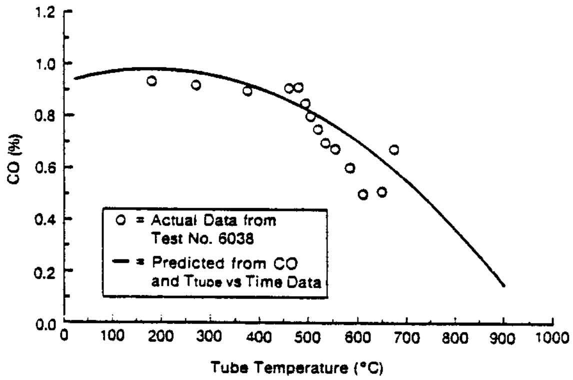


Figure 3-27 Effect of Tube Temperature on CO Emissions

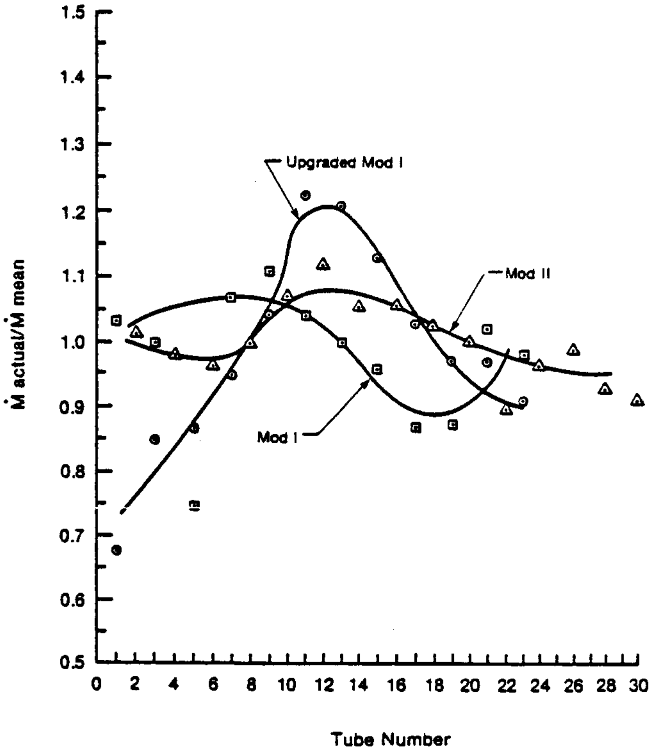


Figure 3-28 Comparison of Measured Pressure-Drop Data for Heater Head Tubes of Mod II, Mod I, and Upgraded Mod I Heater Heads

TABLE 3-5

NOMINAL CHEMISTRY OF HEATER HEAD CASTING ALLOYS AND HEATER TUBE MATERIALS

Heater Head Casting Alloys

	C	Si	Mn	Ni	Cr	Mo	W	Co	Cb	B	N	Fe
HS-31	0.05	0.05	0.05	10.5	25.5	-	7.5	54.0	-	-	-	1.00
CRM-6D	1.05	0.55	4.75	5.0	21.0	1.0	1.0	-	1.0	0.00	-	64.5
SAF-11	0.06	0.07	0.07	16.0	24.0	-	13.0	-	-	0.04	-	44.5
XF-818	0.02	0.03	0.15	18.0	18.0	7.5	-	-	0.4	0.00	0.12	54.0

Heater Tube Materials

	Co	Cr	Ni	Mo	W	C	Al	Ti	B	Cb	Mn	Fe	Si	N
CG-27	-	13.00	38	5.75	-	0.05	1.6	2.5	0.010	0.7	-	38.00	-	
Inconel 625	-	21.50	61	9.00	-	0.05	0.2	0.2	-	-	0.25	2.50	0.2	-
Sanicro 32	-	21.00	31	-	3.0	0.09	0.4	0.4	-	-	0.60	42.80	0.6	-
Sanicro 31H	-	21.00	31	-	-	0.07	0.3	0.3	-	-	0.60	46.13	0.6	-
12RN72	-	19.00	25	1.40	-	0.10	0.5	0.5	0.006	-	1.80	51.80	0.4	-

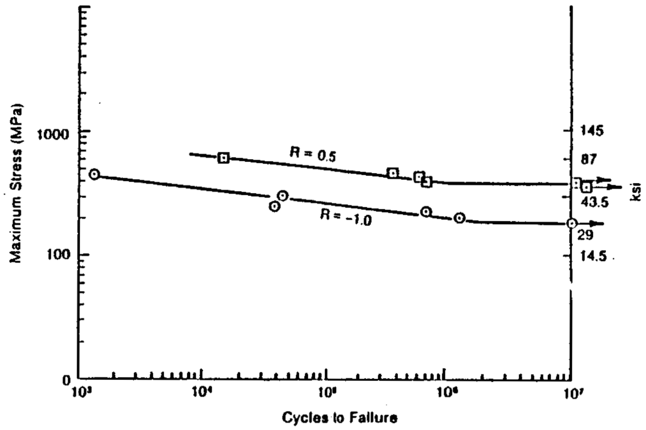


Figure 3-29 Room Temperature Fatigue Test Results, Alloy XF-818

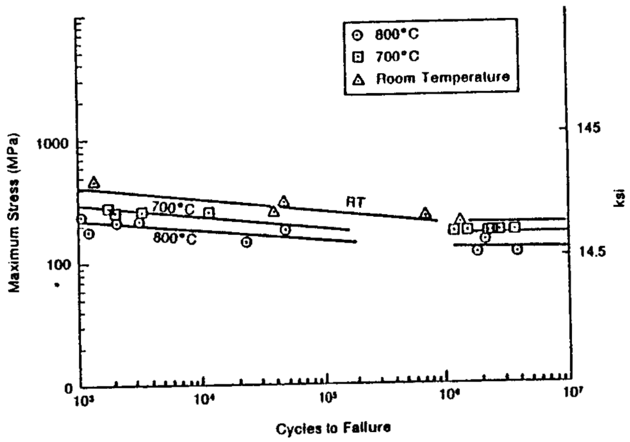


Figure 3-30a Comparison of Fatigue Strength at Room Temperature, 700° and 800°C, at R=-1, for Alloy XF-818

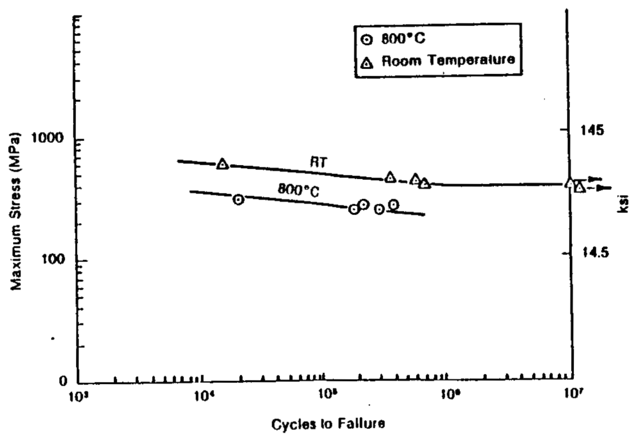


Figure 3-30b Comparison of Fatigue Strength at Room Temperature, 800°C, at R-.5, for Alloy XF-818

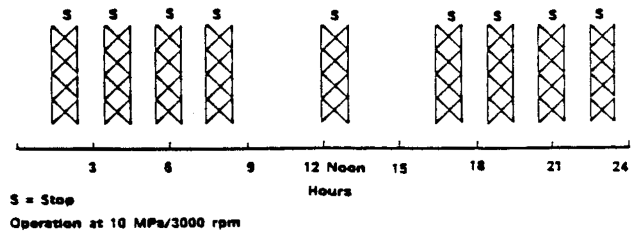


Figure 3-31 Reduced Life Duty Cycle No. 4 for Seal Testing

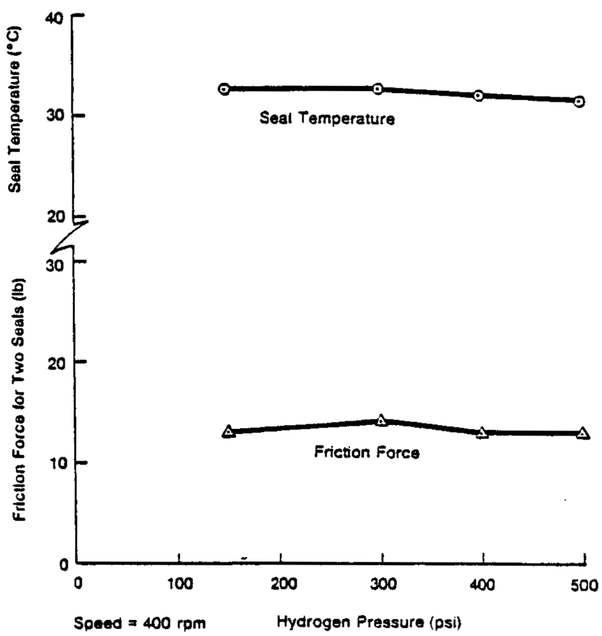
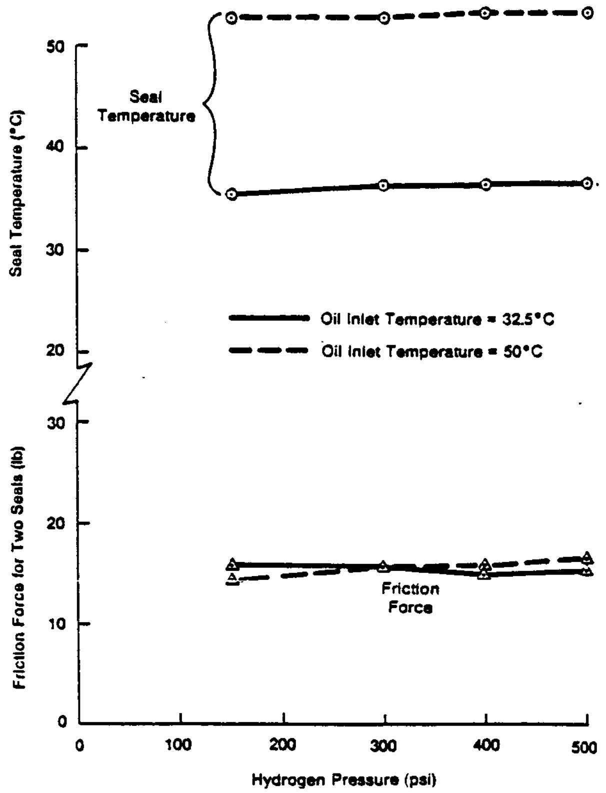


Figure 3-32 Results of Rod Seal Test Under Mod II Low-Idle Conditions



Speed = 500 rpm

Figure 3-33 Results of Rod Seal Test Under Mod II Low-Idle Conditions

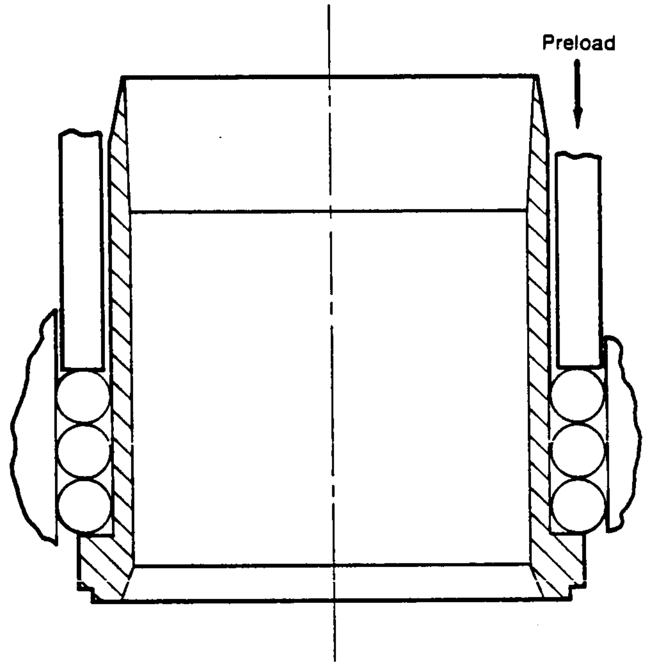
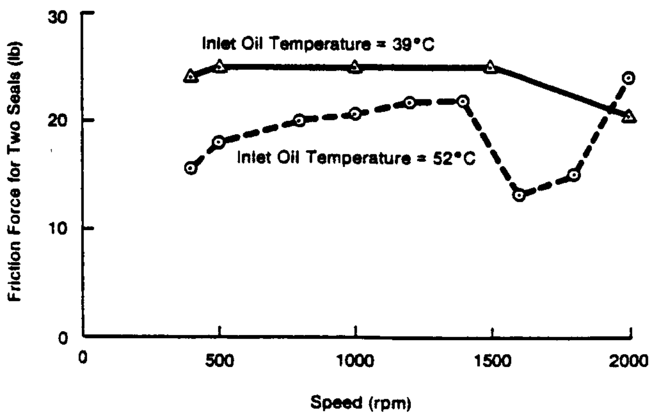


Figure 3-35 Double Angle Seal



Hydrogen Pressure = 150 psi

Figure 3-34 Effect of Engine Speed and Oil Temperature on Rod Seal Friction Force

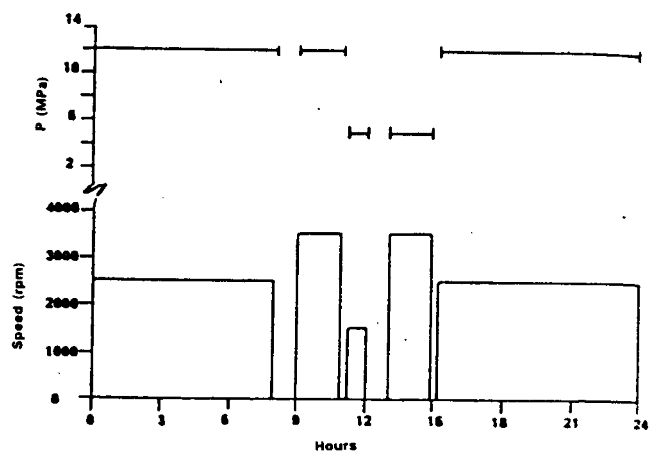


Figure 3-36 Accelerated Duty Cycle for Seal Testing

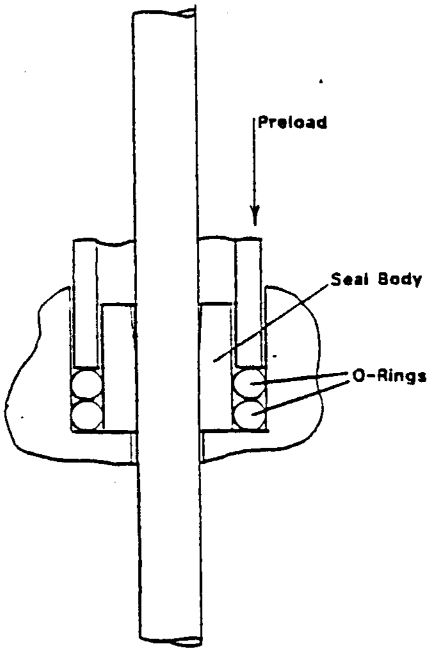


Figure 3-37 Pumping Ring Seal

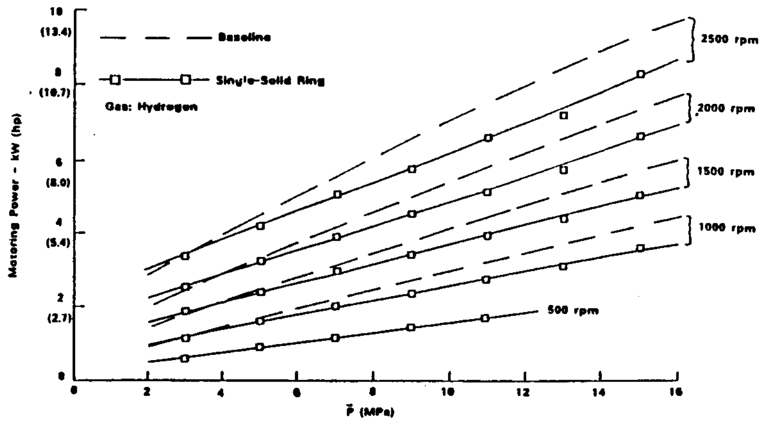


Figure 3-39 Motoring Power - Single Pressure-Loaded Piston Ring

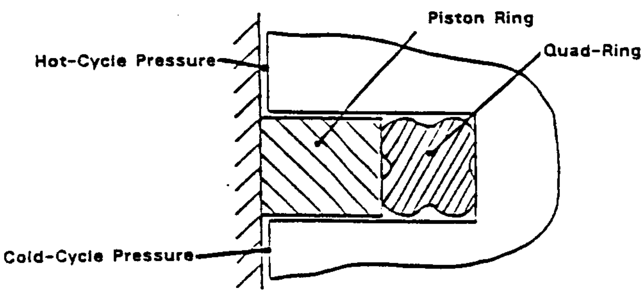


Figure 3-38 Single-Solid Piston Ring

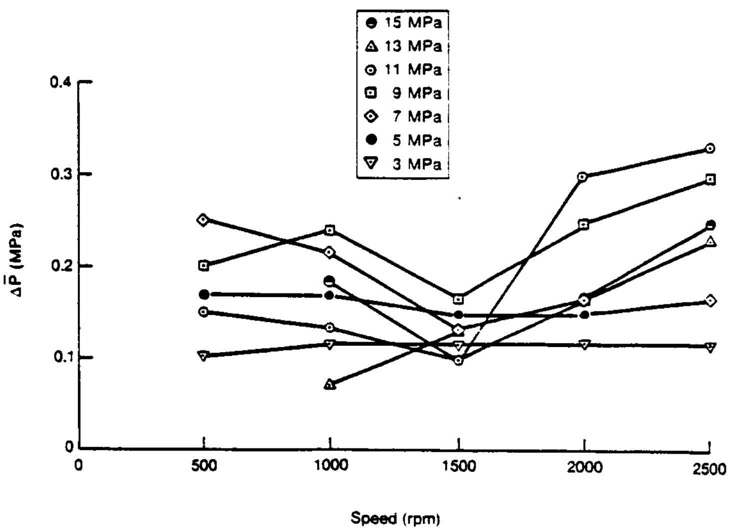


Figure 3-40 Differences in Cycle Mean Pressure with Single-Solid Piston Ring - Motored Engine - Hydrogen

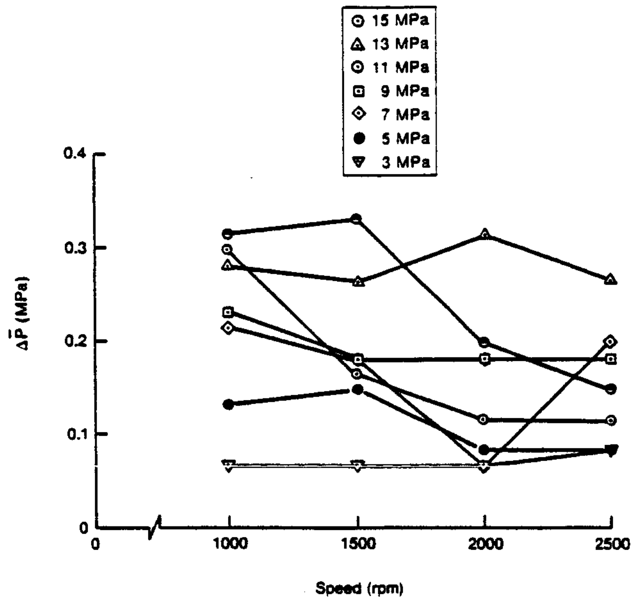


Figure 3-41 Differences in Cycle Mean Pressure with BOM Split-Solid Piston Rings - Motored Engine - Hydrogen

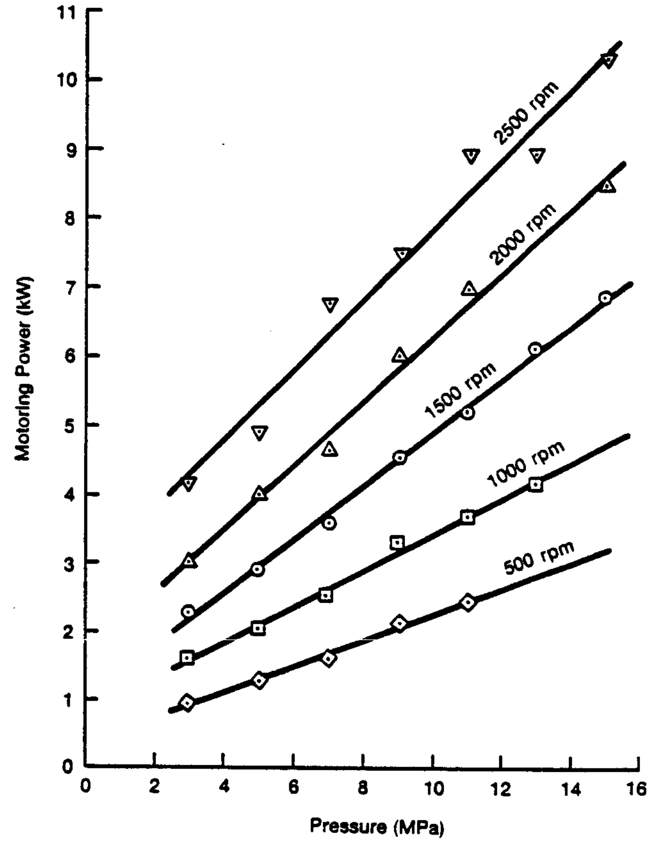


Figure 3-43 New Baseline Data for BOM Piston Rings - Motoring Rig - Hydrogen

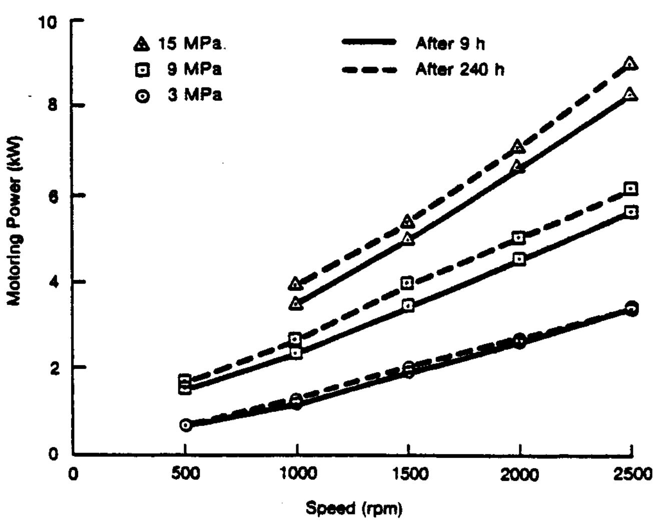


Figure 3-42 Motoring Power - Single-Solid Piston Rings - Hydrogen

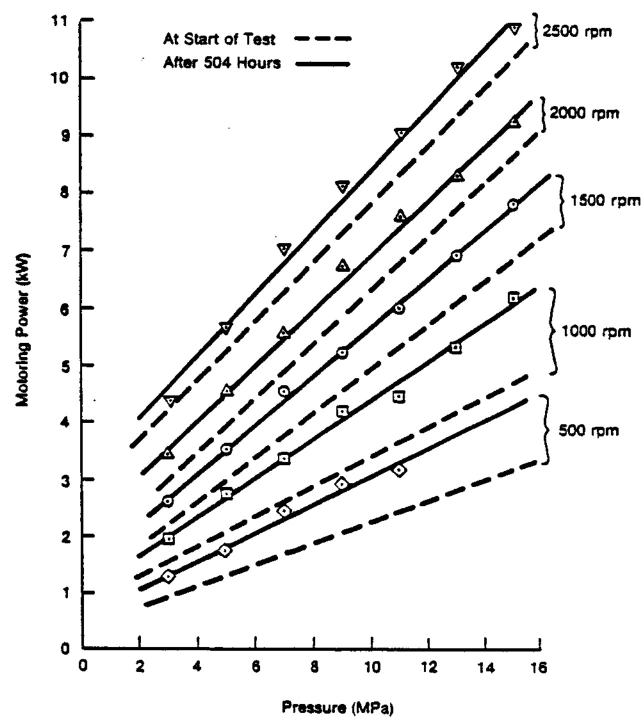


Figure 3-44 Change in Motoring Power During 500-Hour Endurance Test - BOM Piston Rings - Hydrogen

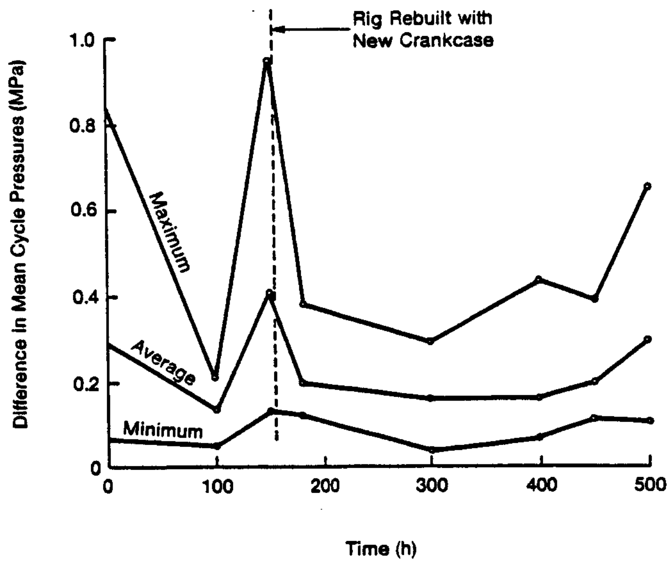


Figure 3-45 Differences in Mean Cycle Pressures During 500-Hour Test - BOM Piston Rings - Hydrogen

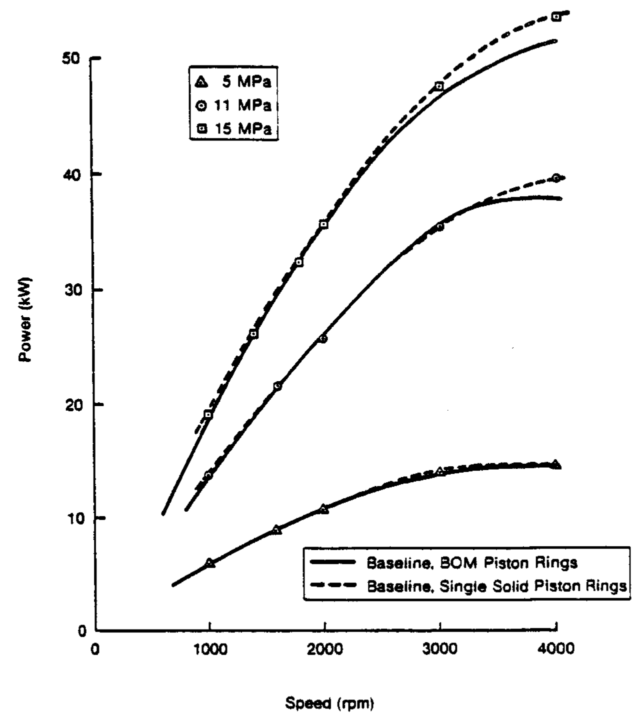


Figure 3-46 Comparison of Power Developed with BOM Rings and with Single-Solid Rings (Mod I Engine No. 5)

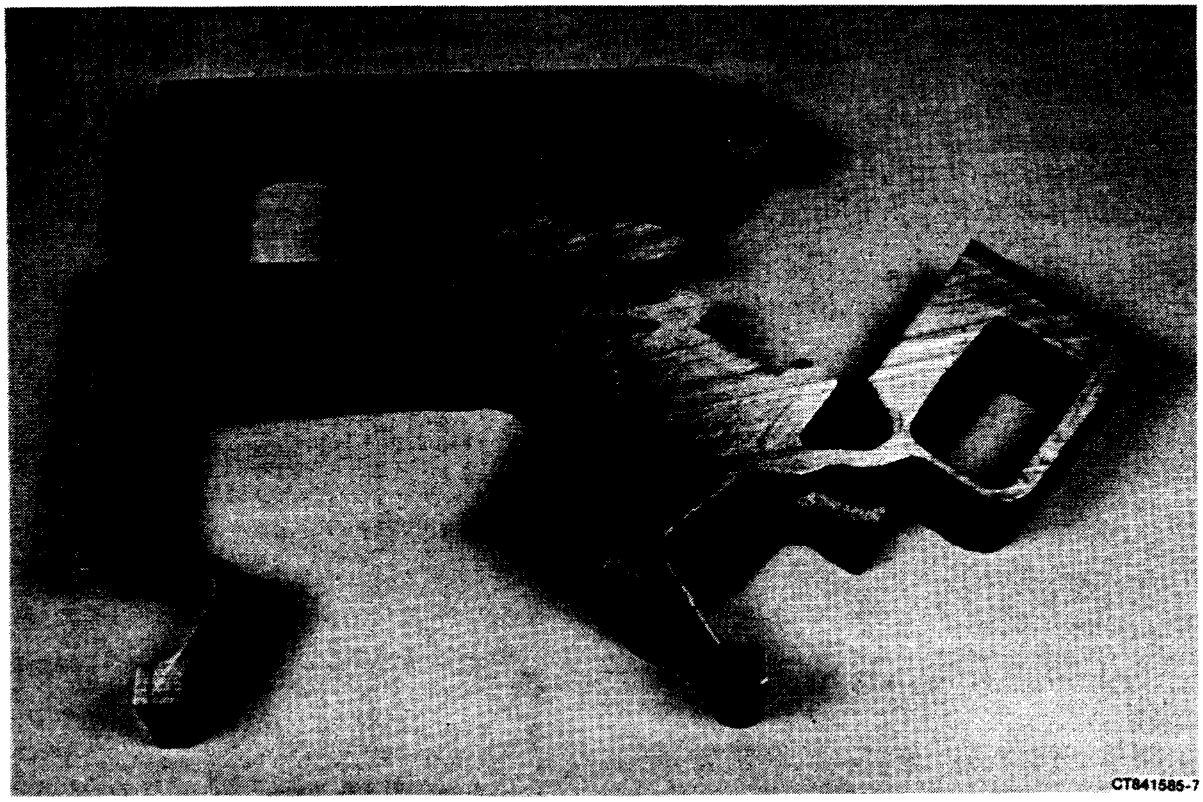


Figure 3-47 Porosity in First Pour of Cast Iron Durability Rig Block

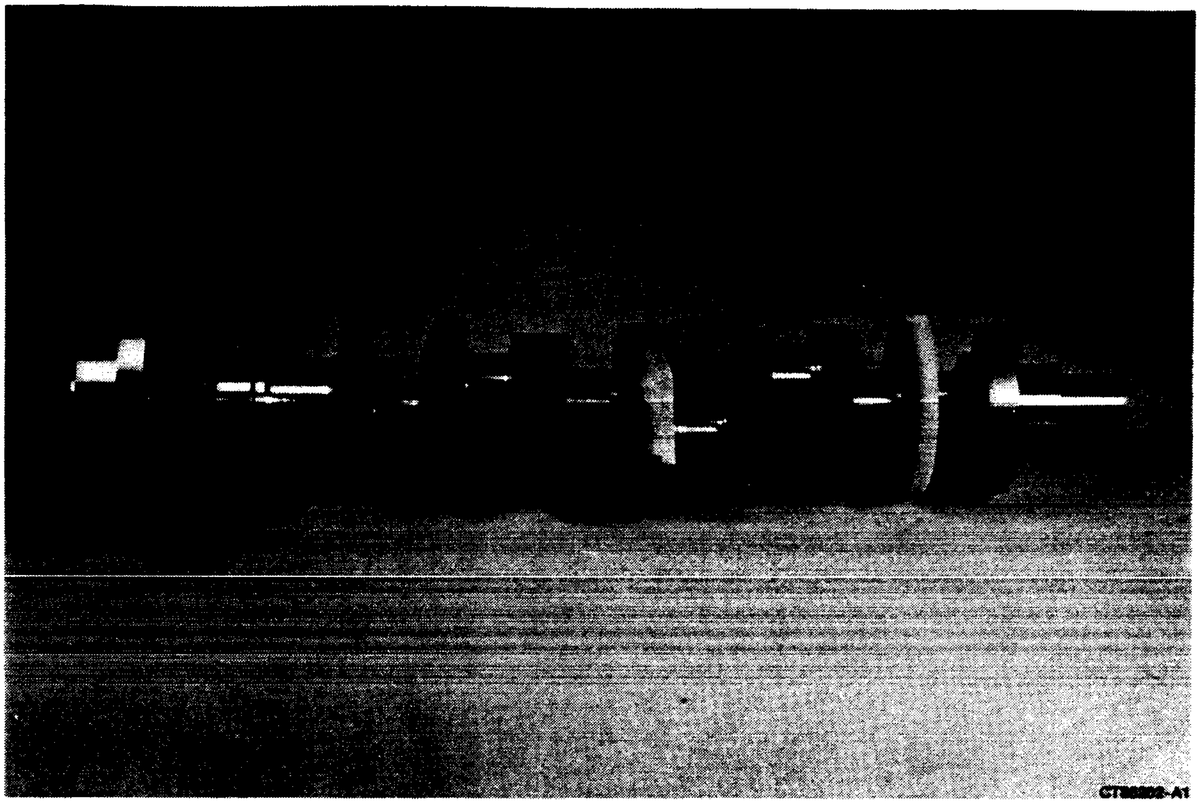


Figure 3-48 Completed Crankshaft for Mod II Durability Rig

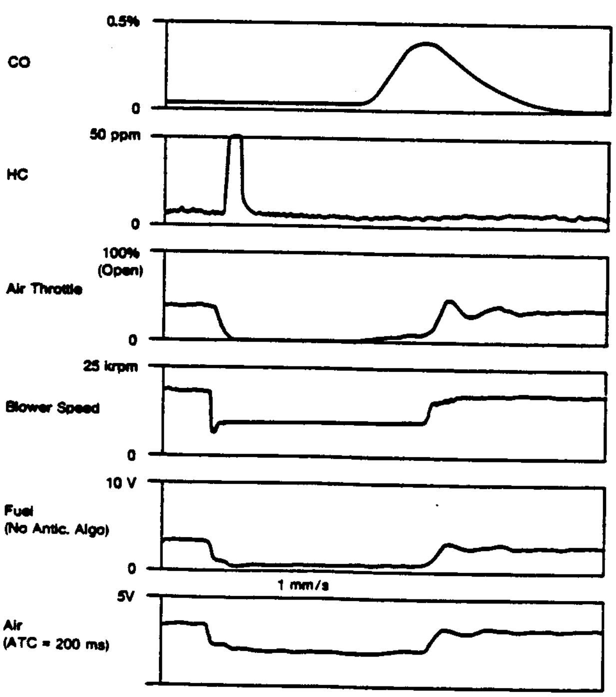


Figure 3-49 Transient Response of DAFC to Engine Speed Reduction without the Anticipatory Algorithm

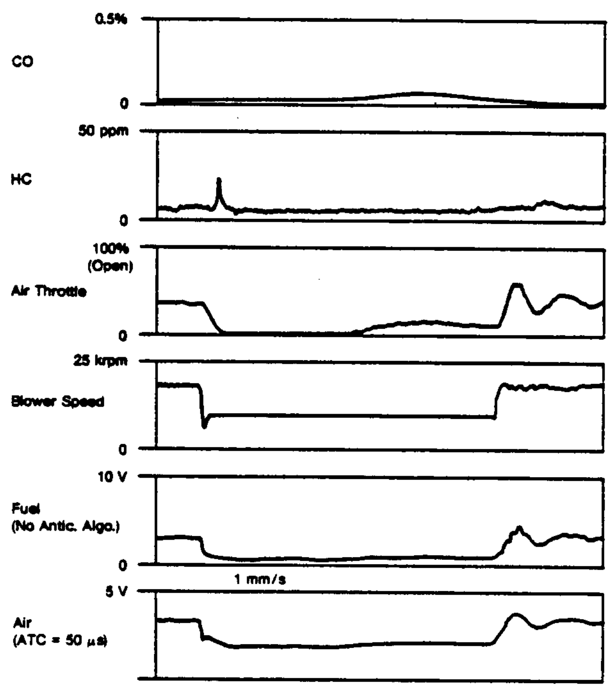


Figure 3-50 Transient Response of DAFC to Engine Speed Reduction with The Anticipatory Algorithm

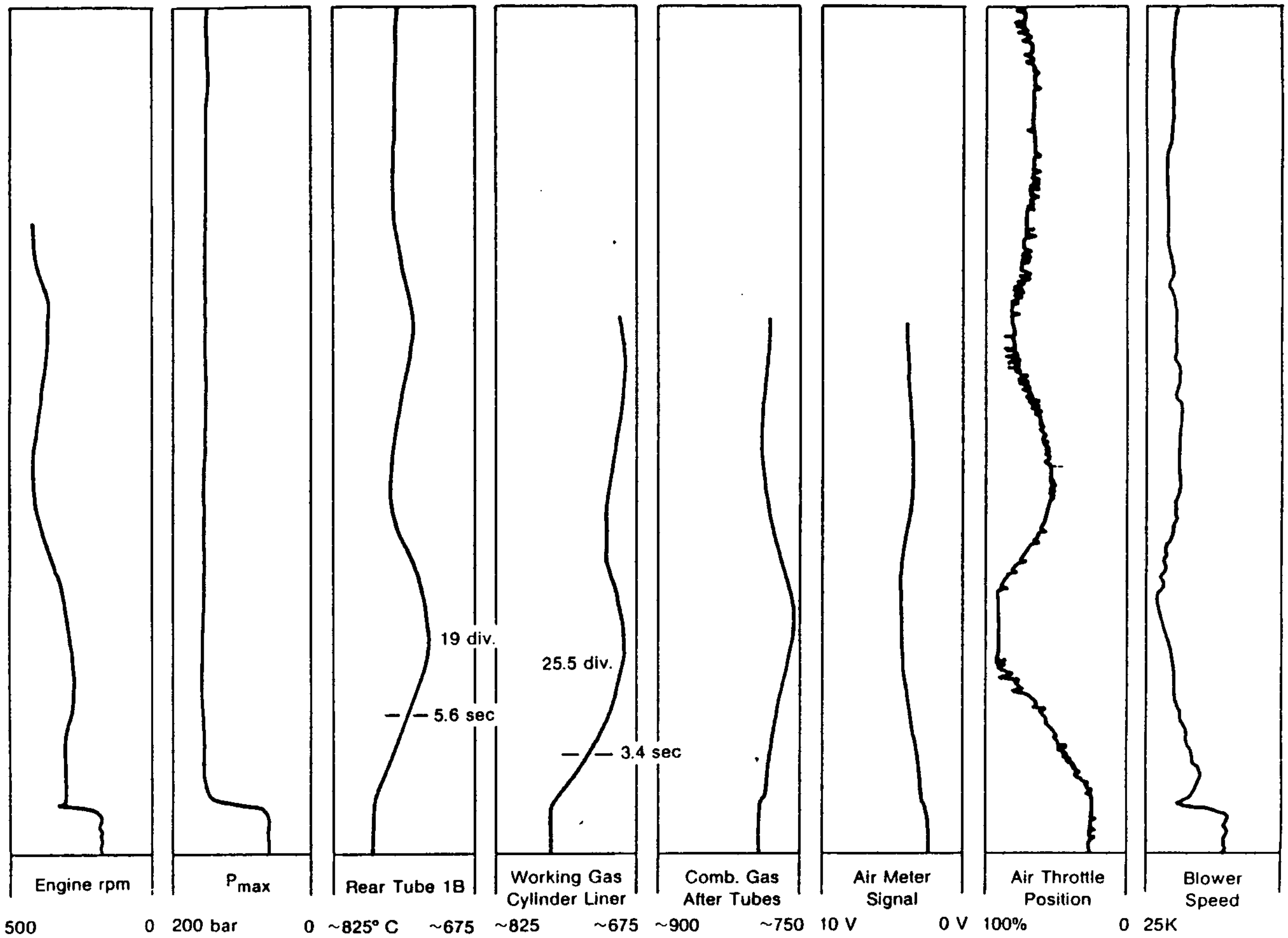


Figure 3-51 Transient Recordings During Hard Acceleration of Engine No. 5

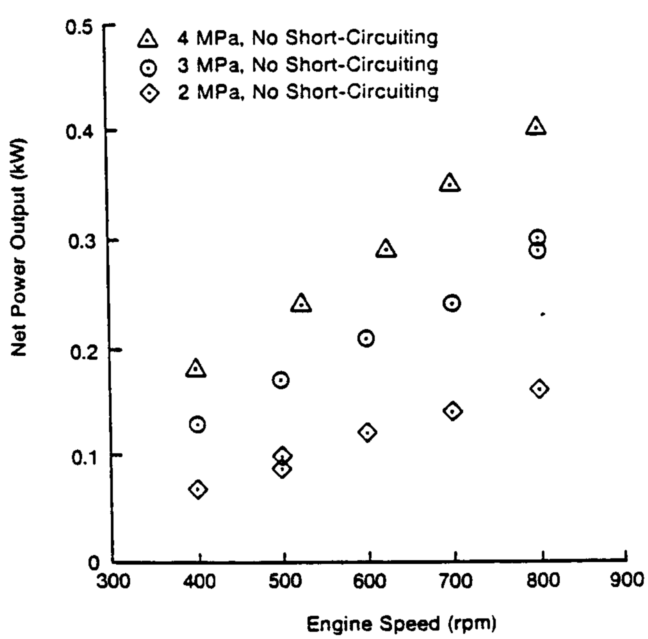


Figure 3-52 Idle Fuel Flow versus Idle Speed Engine No. 8, 820°C, EGR

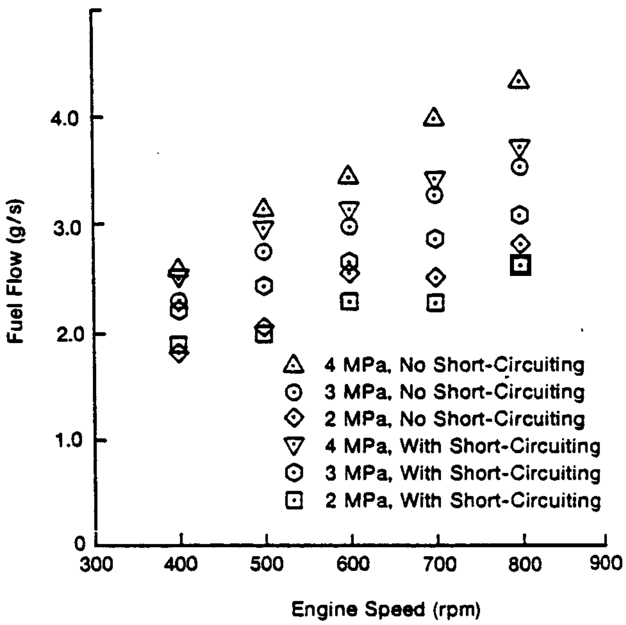


Figure 3-53 Idle Power Output versus Idle Speed, Engine No. 8, 820°C, EGR

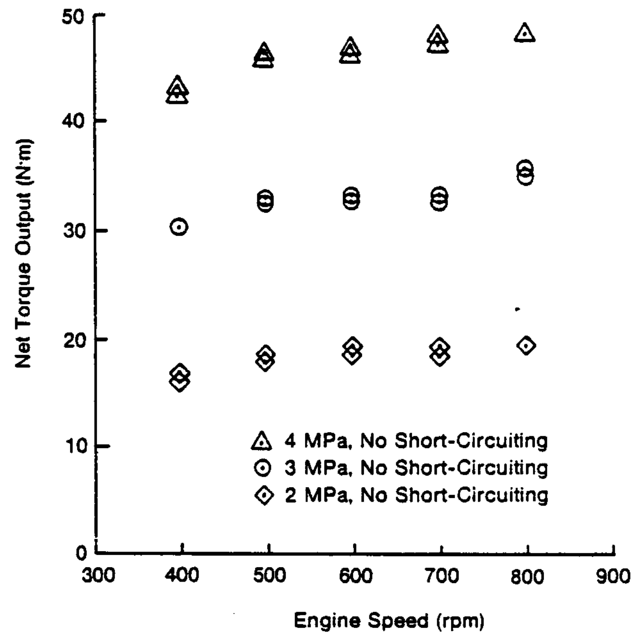


Figure 3-54 Idle Torque versus Idle Speed Engine No. 8, 820°C, EGR

IV. INDUSTRY TEST AND EVALUATION PROGRAM

The major objectives of the 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.
- Provide a larger engine test base to assess engine performance and durability.

The ITEP was completed on schedule during July-December, with the successful completion of Upgraded Mod I engine No. 9's test program at John Deere's Product Engineering Center in Waterloo, Iowa.

Engine No. 9 was delivered to John Deere's facility at the end of August and installed in their test cell in early September. John Deere's Test and Evaluation Program was completed in October.

The engine was subjected to the same test program that any other competitive engine would undergo, and completed the entire evaluation without any major problems.

The test and evaluation program included noise measurements, operation on gasoline, JP-4, and diesel fuels. Additionally, the engine was operated at heater head temperatures of 720°, 770°, and 820°C.

John Deere is currently preparing a test report which will summarize their engine evaluation.

As discussed in the previous Semiannual, the Spirit vehicle with Upgraded Mod I engine No. 8 installed, completed the test and evaluation program at GM Research Laboratories.

Upon return to MTI, engine No. 8 was removed from the Spirit and installed in the engine test cell to determine the engine's power and efficiency. Test cell results indicated the health of the engine to be as good as that during its acceptance tests prior to GM tests.

The engine was then completely disassembled for a detailed part inspection by engineering personnel. Personnel from GM also participated in this inspection. In general, the condition of the engine was very good. The major problem found was that oil had migrated into all four cycles. All parts were cleaned and the engine was reassembled with new piston rings, PL seals, new design steel crosshead guides, and a high temperature (820°C) heater head. The engine was then utilized for various component development activities which are ultimately scheduled for incorporation in the Mod II design, such as DAFC, proportional blower speed control, low idle speed and pressure operation, and DEC modifications to accomodate a manual transmission instead of the automatic.

Engine No. 8 was then reinstalled in the Spirit vehicle coupled with a four-speed manual transmission. Efforts have since been concentrated on transient testing of the component development items that were tested earlier in the test cell.

At the end of this reporting period the Spirit was being readied for a series of

CVS tests in January 1985 to be conducted at Mercedes Benz. These tests will characterize the development items mentioned earlier. Additionally, the optimum rear axle will be selected and the shift schedule will be optimized.

V. MOD I ENGINE DEVELOPMENT

Mod I Hardware Development

Several items were examined as key areas for development of the Mod I engines. However, these items were kept to a minimum so that priorities could be devoted to development of Mod II related items. Items in the Mod I category are as follows:

- High-temperature head manufacture
- Crankcase/bedplate cracking
- Heater tube failures
- Cylinder-liner top O-ring overheating
- Hydrogen compressor connecting rod small-end bearing wear.

The following sections present the details of the Mod I development items.

MOD I HEATER HEAD MANUFACTURE - FIN CORROSION

The problem of heater head fin corrosion was discussed in the previous Semiannual. The development of the slurry-dip process for brazing heater heads continued in late 1984. During this time, a brazing procedure was established for the "high temperature Mod I" heater heads. (The process was developed by USAB as part of a non-ASE Program and is considered proprietary to USAB.) The "high temperature heater heads" are currently on all Upgraded Mod I engines. They are also being evaluated on the vehicle and the endurance engines.

CRANKCASE/BEDPLATE CRACKING

As of the end of 1984, several engine hours had been accumulated on the reinforced crankcase/bedplate design. This particular design was developed to prevent cracking of the crankcase at weak areas of the aluminum casting. All of the current program engines (excluding engine No. 5) have the reinforced crankcase/bedplate design incorporated in them. To date, over 2500 hr have been accumulated with no cracks occurring. Monitoring of this area will continue.

HEATER TUBE FAILURES

During July-December only one heater-head tube failure occurred. This failure occurred on Upgraded Mod I engine No. 5 and was the the result of a mistake in engine control thermocouple analysis. The control of engine temperature levels was removed from the rear tube row and placed on the working-gas thermocouples found in the manifold of the cylinder and regenerator manifolds. This was originally done while the engine was running and this time was mistakenly done while the engine was shutdown. Upon restart, gas does not initially circulate until the starter engages. This engagement is to occur when tube temperature hits a certain level. Since the control was now on working gas (which never saw the temperature heat-up effects because of no circulation) the tubes were overheated and failed upon pressure increase of a manual engagement of the starter.

Several braze joint leaks have occurred. These are manufacturing related problems with the CG-27 tubes. The titanium and aluminum content of the CG-27 alloy adversely effects the brazing characteristics.

CYLINDER LINER TOP O-RING DETERIORATION

The higher operating temperature of the Upgraded Mod I over the Mod I (820° versus 720°C) has resulted in aggravation of a previously infrequent deterioration of the cylinder-liner top O-ring. The deterioration of this O-ring eventually leads to the escape of gas into the water jacket. Operation of the engine at the higher temperature has led to the cracking/aging of the O-ring very quickly. To help keep the O-ring area slightly cooler, the O-ring immediately below the mentioned failing ring was removed. The purpose of the lower ring was only to keep water away from the gas sealing area of the top O-ring in an effort to prevent corrosion. It is hoped the change will provide a cooler O-ring and give it more life. Engine No. 6 has incorporated this and is presently running 820°C endurance (see Figure 5-1).

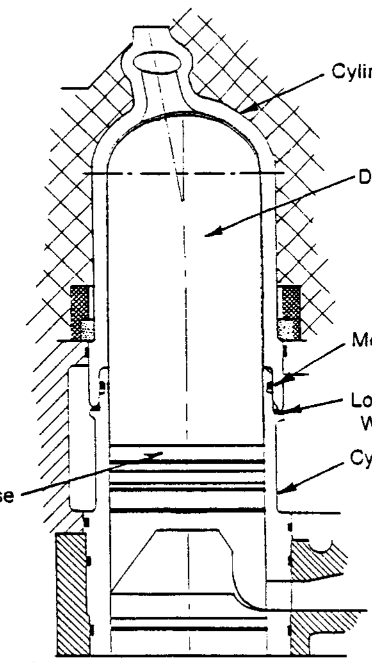


Figure 5-1 Mod I Cylinder Configuration

WRIST PIN BEARING, H_2 COMPRESSOR - CONNECTING ROD

Problems in the hydrogen compressor connecting rod small end have been related to wear of the bearings. This wear is attributed to the fact that as the compressor rotates unloaded (nonpumping) there is a constant downward load on the bearings (no load reversal). This constant unidirectional load has resulted in wear on the bearings in anywhere from 200 to 600 hr. To date, larger load bearing surfaces and forced lubrication have not eliminated the problem. We are continuing to examine potential solutions.

Mod I Engine Test Program

At the end of 1984 a total of seven Mod I engines were operational with the ASE program. Two of these engines were the original Mod I and were located at USAB in Sweden. The remaining engines were Upgraded Mod I engines of which three were located at MTI and two at USAB. The specific purpose of each of the engines is listed in Table 5-1.

TABLE 5-1
ENGINE DEVELOPMENT TEST ASSIGNMENTS

Mod I Engine No. 3	(USAB)	- EHS Development
Mod I Engine No. 7	(USAB)	- Seals/Ring Development
Upgraded Mod I Engine No. 5	(MTI)	- General Development
Upgraded Mod I Engine No. 6	(USAB)	- General Development
Upgraded Mod I Engine No. 8	(MTI)	- Transient Vehicle Performance
Upgraded Mod I Engine No. 9	(MTI)	- Performance, NASA Testing
Upgraded Mod I Engine No. 10	(USAB)	- Performance, NASA Testing

MOD I ENGINE NO. 3

Mod I engine No. 3 accumulated a total of 180 hr during the last half of 1984, bringing its total engine hours to 1816. During July-December it continued on its primary assignment of EHS development. As part of this program the main efforts were in reducing soot levels and the development of both EGR and CGR combustors. The selection of the Mod II combustor design, EGR or CGR, will be made in early 1985.

The specific areas evaluated included: combustor swirlers, fuel nozzle spray angles, atomizing air pressure, the conical fuel nozzle, and various CGR combustors. Testing on this engine will continue until mid-1985 when it will be retired. Activity until this time will be on the chosen basic Mod II combustor design.

UPGRADED MOD I ENGINE NO. 5

Upgraded Mod I engine No. 5 is located at MTI and is used for general development work. During the last half of 1984 the engine accumulated 199 hr, bringing its total operating time to 883 hr. Testing on the engine was limited by available test cell time. Engines No. 8 and 9 were tested during July-December in the MTI test cell.

Testing during July-December involved several areas including the following items:

- Extended-neck heater heads
- Kalrez O-rings
- Single-solid piston rings
- Hot shutdowns.

Brief descriptions of the results of each test follow. The extended-neck cylinder housing heater heads were a further attempt to improve flow into the cylinder housing from the cylinder manifold. The original Upgraded Mod I cylinder housing created a poor distribution of flow into the housing proper, causing O-rings at the lower portion of the housing to overheat in a circumferentially preferential manner. This was corrected by welding in flow diverters in the cylinder housing neck areas, or by using Mod I-design cylinder housings. It was determined that the use of a straighter inlet neck would further improve this distribution. The use of this extended or straight neck reduced the circumferential gradient in the O-ring area from 180° to $102^\circ C$. The maximum temperature measured was also reduced by $34^\circ C$, from 686° to $652^\circ C$. Further testing will be done on these cylinder housings to thoroughly evaluate the design.

Kalrez O-rings were evaluated for a short time at the top cylinder liner O-ring position. These O-rings were run for ~10 hr. While no leakage was observed, the O-rings did appear to have taken a permanent set in the shape of the O-ring groove, and several circumferential cracks were found. The O-rings were not reinstalled and no further testing is planned.

A series of hot starts were performed on the engine in the test cell. Instead of allowing the engine to coast down normally, the engine was deliberately stalled, and cooling water flow was reduced to the level provided by a vehicle after-cooling pump. The test was completed 10 times and no leakage was observed into the water jacket. It was concluded that the deterioration of the upper cylinder-liner O-ring is time-dependent and not the result of melting. In any event, the hot shutdown appears to be acceptable for use in the vehicle.

Evaluations were also made of the component development area's single-solid piston rings. Engine starts with the new ring design were found to be identical to those with the BOM split-solid rings. A gain of 1.8 kW at maximum power was measured while a slight power loss was noted at lower speeds. The pressure balance from cycle-to-cycle was acceptable and the rings were kept in the engine for further reliability testing. At the end of the year, 78 hr had been accumulated on the rings and testing was continuing.

During July-December several short duration tests were performed to characterize electronic control functions and various combustor/fuel nozzle modifications. Upcoming in early 1985 a special heater head configured with the Mod II tube/fin arrangement will be evaluated. Testing on this design will include the measurement of front-row tube temperatures on both the special and BOM heater heads.

MOD I ENGINE NO. 7

Mod I engine No. 7 at USAB was devoted to seals development during July-December. The engine accumulated 960 additional hours bringing its total hours to 2339. The main seal test, started in early 1984, was terminated at 2158 hr on two main seals. The remaining two cylinders had hours of 980 and 568. The reason for termination of testing was a failure of the dynamometer cooling loop which caused a secondary failure of the engine regenerators due to overspeed. The regenerator failure distributed debris throughout the engine and damaged the seals.

Testing was not resumed on this unit during the remainder of the reporting period. When resumed, emphasis will be on testing piston rings.

UPGRADED MOD I ENGINE NO. 8

Engine No. 8 accumulated 287 hr of transient vehicular hours during the last half of 1984, bringing total engine hours to 684.

The car was tested early in 1984 at GM and the results of the testing were reported by GM at the October 1984 CCM in Detroit.

The analysis of the testing at GM pointed out two possible areas to be worked on to improve performance, namely; the CSP and the idle fuel consumption. To address these and other items related directly to the Mod II engine's eventual performance, a program of development was started.

The major developmental items included in this program were:

- High temperature heater head
- Roller bearing drive
- Four-speed manual transmission
- Optimized rear axle ratio
- Hot shutdowns
- Low idle speed/low idle pressure
- Proportional blower speed control
- DAFC
- Revised fan schedule.

To start on this program the engine was installed in the test cell and a number of these items were checked out on a preliminary basis. The roller bearing drive was eliminated from consideration because of crankshaft material problems. Following the test cell work, the engine was reinstalled in the vehicle with the manual transmission. At 820°C the maximum power output was 64 KW and the maximum efficiency was measured to be 38.3%.

Immediately upon completion of installation and at the start of testing the proportional blower speed control was eliminated because of insufficient hydraulic supply to the system.

The tendency of the engine to overspeed up to 4750 rpm when the transmission was shifted at or above 3000 rpm was noted, as well as severe instability at idle. Both of these problems were the result of having a "free running" engine when the engine is in neutral or the clutch is depressed. Both of these problems are being addressed by a DEC programming approach, and this effort is continuing. A January 1985 retest at the Mercedes Benz Emissions Laboratory is planned.

UPGRADED MOD I ENGINE NO. 9

Engine No. 9 became operational and accumulated a total of 218 hr during the last half of 1984. The engine was performance tested in MTI's test cell and delivered to John Deere as part of the ITEP program, and was retested upon return.

The engine was evaluated with three different working fluids and three different rear-tube temperatures. Working fluids tested included hydrogen, helium, and nitrogen. The tube temperatures evaluated were 720°, 770°, and 820°C. The results of the testing are plotted in Figures 5-2 through 5-10. It should be remembered when reviewing these data that the engine was designed for hydrogen, not for helium or nitrogen. It is apparent from the data that the power and operating speed range are adversely affected by use of helium or nitrogen.

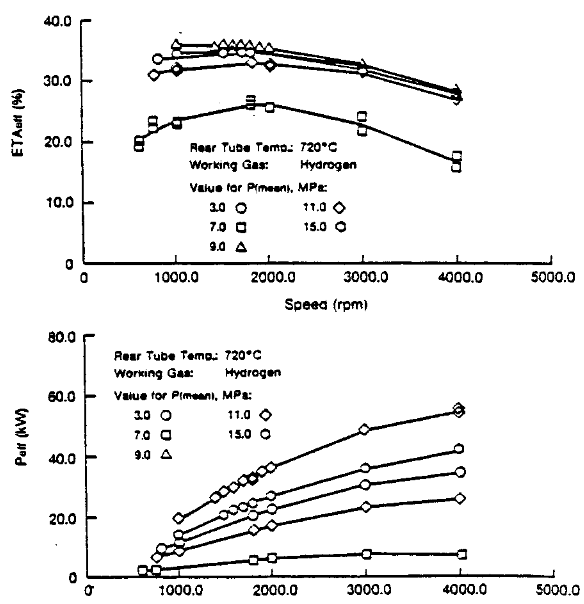


Figure 5-2 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 720°C Rear Tube Temperature with Hydrogen Working Gas

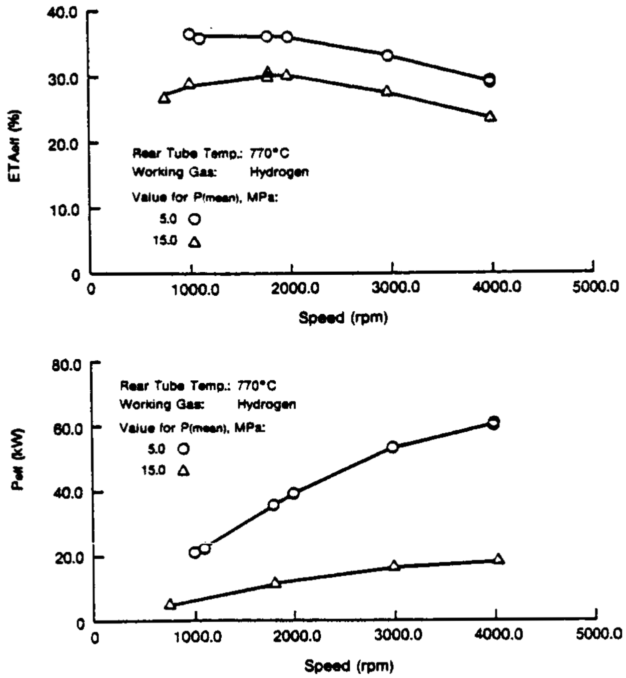


Figure 5-3 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 770°C Rear Tube Temperature with Hydrogen Working Gas

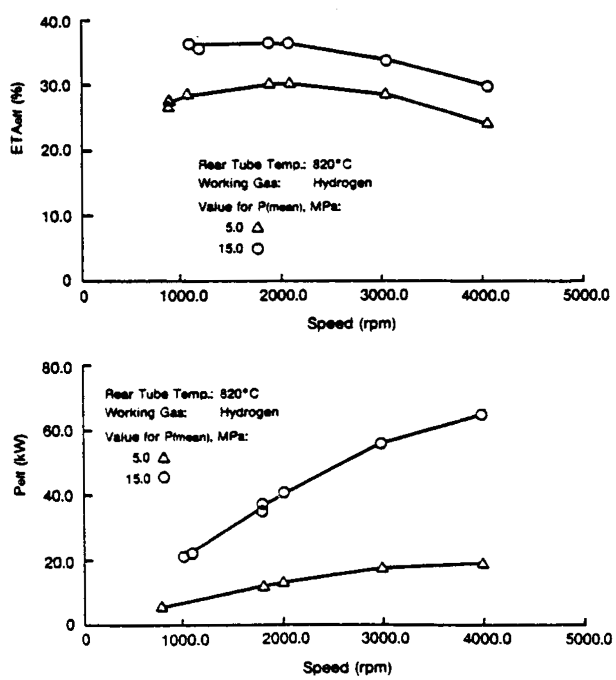


Figure 5-4 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 820°C Rear Tube Temperature with Hydrogen Working Gas

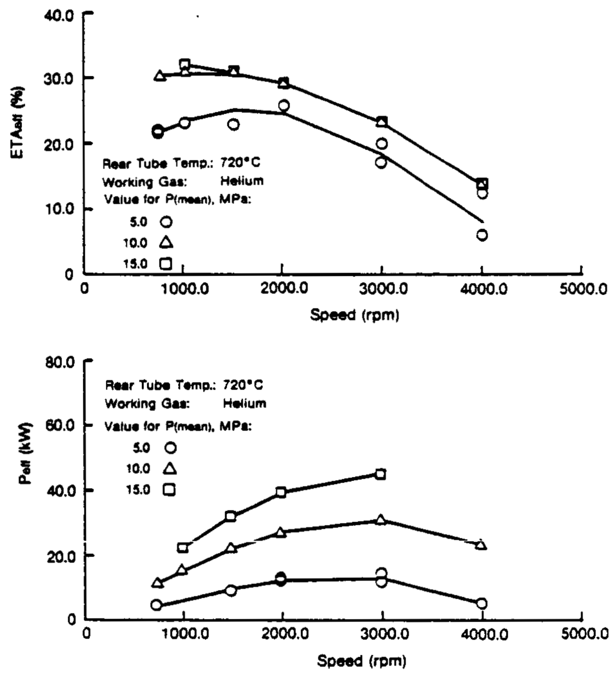


Figure 5-5 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 720°C Rear Tube Temperature with Helium Working Gas

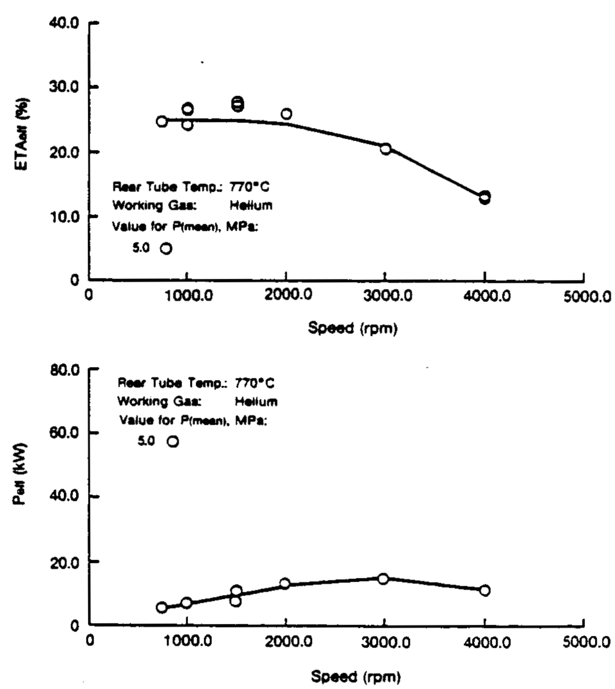


Figure 5-6 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 770°C Rear Tube Temperature with Helium Working Gas

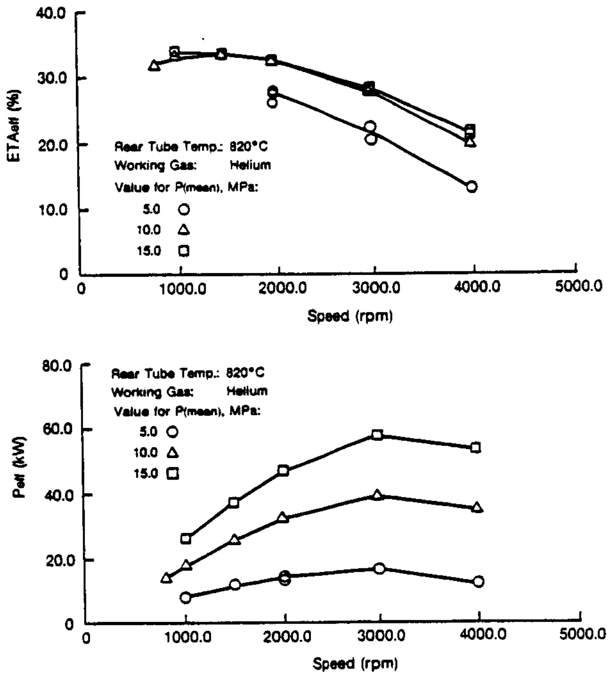


Figure 5-7 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 820°C Rear Tube Temperature with Helium Working Gas

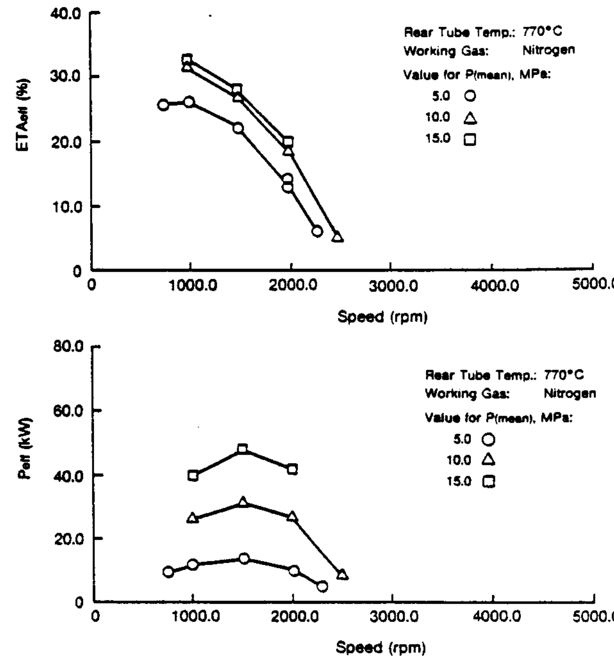


Figure 5-9 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 770°C Rear Tube Temperature with Nitrogen Working Gas

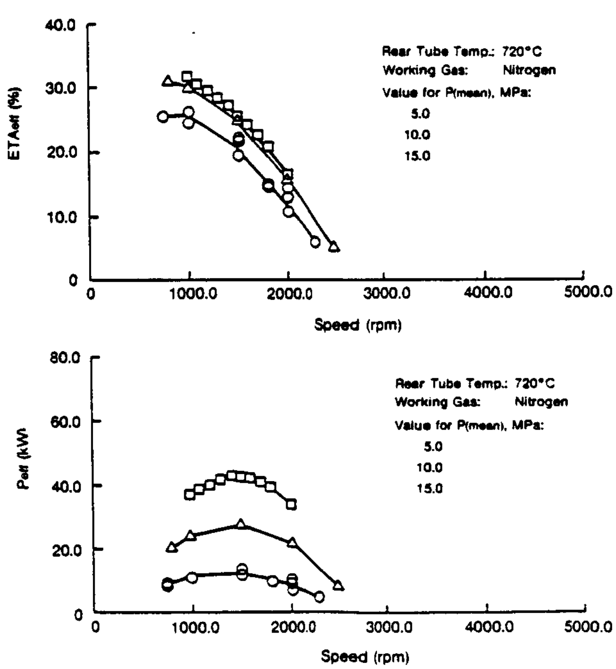


Figure 5-8 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 720°C Rear Tube Temperature with Nitrogen Working Gas

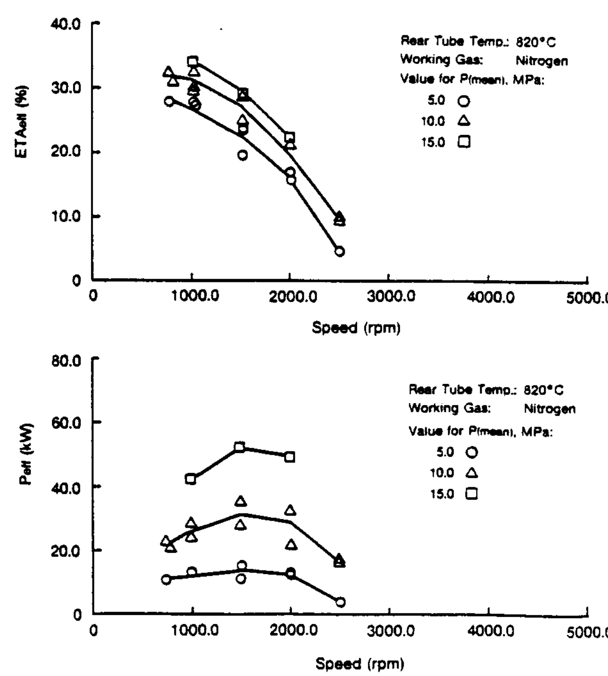


Figure 5-10 Upgraded Mod I Engine No. 9 - Efficiency/Power versus Speed at 820°C Rear Tube Temperature with Nitrogen Working Gas

Following a demonstration for John Deere, the engine was shipped to their facility and evaluated in their test cell. The results of the testing will be presented in reports under the ITEP program written by John Deere. Upon return to MTI the engine was reinstalled in the test cell and its performance measured as returned. At this time, the engine was found to be 5 kW low in power at 3000 rpm. This is the maximum speed evaluated by John Deere due to test cell restrictions. The piston rings were replaced and the engine was retested and found to be at acceptance-test levels. It is believed at this time that initial facility problems may have contributed to the performance deterioration, as both engine overheating and high exhaust restrictions occurred.

UPGRADED MOD I ENGINE NO. 10

Engine No. 10 is an upgraded version of Mod I engine No. 1. The engine was built and tested at USAB prior to delivery to NASA/Lewis Research Center. The engine will be used for NASA testing programs, has accumulated 81 hr, and is expected to be shipped to NASA in January 1985. Following a check-out test at NASA, the engine's hardware will be upgraded while an assembly training class takes place.

Mod I Engine Performance Analysis

UPGRADED MOD I SPIRIT ANALYSIS

During this report period, analysis efforts were concentrated on comparing actual Spirit performance to predicted levels and on estimating the best Spirit performance that could be obtained with a "best" Upgraded Mod I engine installed.

During tests of the Spirit vehicle at GM under the ITEP program, driving-cycle fuel economy tests and accelerations were conducted on a chassis dynamometer with a road load setting of 6.9 hp (5.2 kW) to simulate a Chevrolet Chevette vehicle. The MTI vehicle simulation code was exercised to predict the fuel economy and acceleration times compared to the actual test data. Results of this comparison are presented in Table 5-2. Excellent agreement is shown between predictions and actual data. The largest discrepancy is in the 30- to 50-mph acceleration time (6.7 predicted versus 7.4 actual), probably due to the occurence of the 2nd to 3rd shift just at 50 mph. Any errors in modeling of the shift dynamics could cause this error (a 0.7 sec difference is not considered to be significant). The capability to accurately model vehicle performance, as demonstrated in the Spirit testing, provides a sound base for performance predictions for other vehicle/engine systems such as the Mod II.

TABLE 5-2

UPGRADED MOD I - SPIRIT VEHICLE PERFORMANCE

AMC Spirit at Chevette Road Load - 6.9 hp (5.2 kW)

		Actual	Predicted
Urban	(mpg)	22.1	22.5
Highway	(mpg)	41.1	42.0
Combined	(mpg)	28.0	28.4
0-60 mph	(sec)	17.3	17.3
30-50 mph	(sec)	7.4	6.7

An estimate was prepared for performance that could be achieved if the "best" possible Upgraded Mod I engine were to be built and installed in the Spirit vehicle. The following changes would be made to the engine/vehicle system:

- 820°C operation
- Rolling element drive
- DAFC
- Four-speed manual transmission
- Modified hot shutdown technique
- Reduced idle fuel flow.

The impact of these changes is shown graphically on Figure 5-11, compared to Spirit GM ITEP test results. As noted thereon, the GM ITEP tests demonstrated a fuel economy ~13% lower than the 1984 fleet average at the tested inertia weight. The impacts of the previous changes are shown separately on Figure 5-11. For each of the changes, vehicle performance (0-60 mph acceleration times) was held constant by allowing the vehicle weight to increase as engine power increased or as drivetrain improvements allow potentially faster acceleration rates. The total effect of all changes would result in a vehicle which achieves a fuel economy ~30% better than the 1984 fleet average at its particular inertia weight.

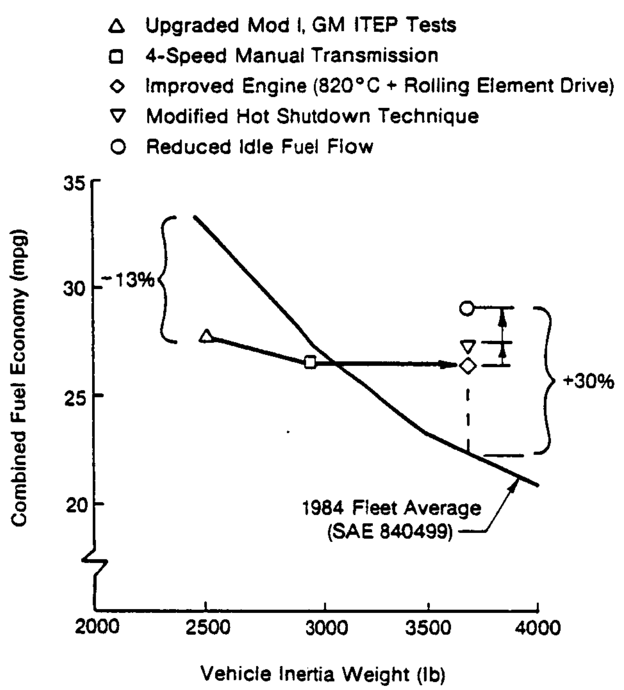


Figure 5-11 Vehicle Performance Potential, "Best" Upgraded Mod I Engine

JOHN DEERE ENGINE DATA ANALYSIS

An Upgraded Mod I engine (engine No. 9) was tested at the John Deere faciltiy to assess its performance potential as a generator set power source.
Data accumulated at John Deere indicated degraded performance relative to the final acceptance testing at MTI, so a retest was conducted after the engine was returned to MTI. The retest confirmed John Deere performance levels and indicated a possible piston ring problem. Replacement of the piston rings restored performance to the final acceptance levels. Problems encountered during testing at the John Deere facility could have contributed to the piston ring degradation. These included cooling system problems which caused a hot engine shutdown, and an unbalanced drive shaft which caused excessive engine vibration.

The performance of the engine for the various tests is shown in Figures 5-12 through 5-17. Power levels, efficiency, and BSFC at final acceptance were the best ever demonstrated for an Upgraded Mod I engine. The degradation at John Deere is apparent from Figures 5-12 and 5-13. As noted on Figures 5-14 and 5-15, the John Deere performance was confirmed upon retest at MTI. The heat balance also supported the power and efficiency levels, indicating higher heat rejection to the cooling water for the poor performance data. This can be symptomatic of poor piston ring or regenerator performance. The engine was disassembled and inspected. The regenerators were found to be in excellent condition, so only the piston rings were replaced. A repeat test conducted after ring replacement did provide a return in performance to final acceptance levels (Figures 5-16 and 5-17).

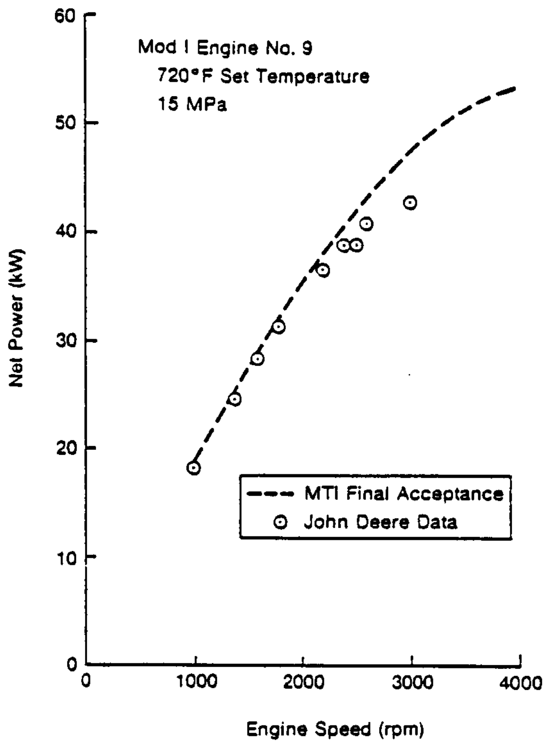


Figure 5-12 Comparison of MTI Final Acceptance Net Power to John Deere Data

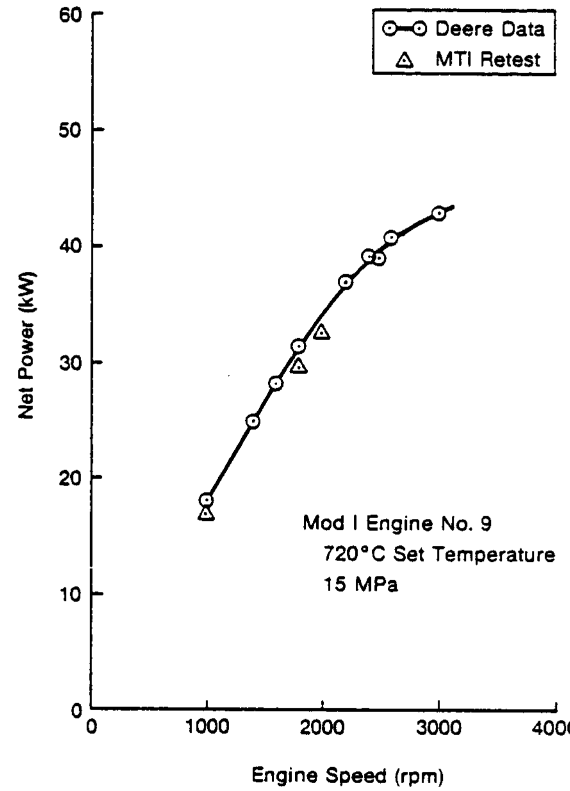


Figure 5-14 Comparison of John Deere Net Power Data to MTI Retest

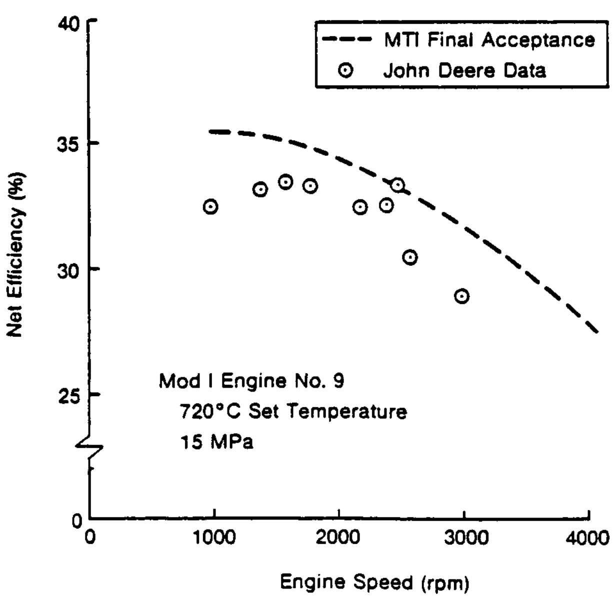


Figure 5-13 Comparison of MTI Final Acceptance Net Efficiency to John Deere Data

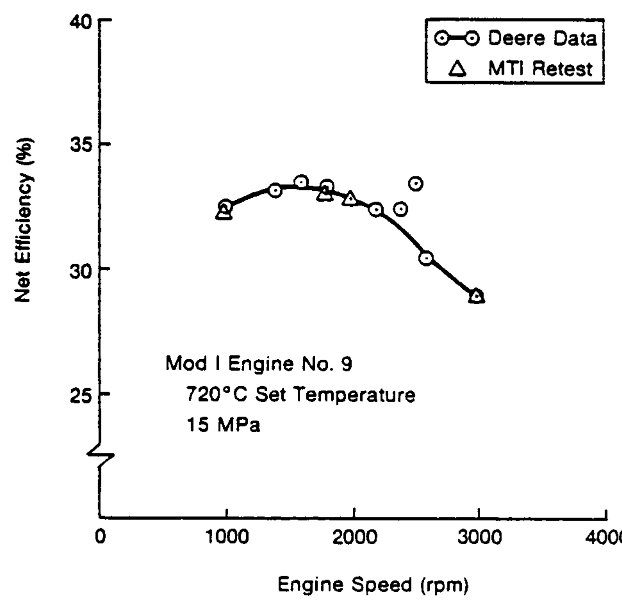


Figure 5-15 Comparison of John Deere Data Net Efficiency to MTI Retest

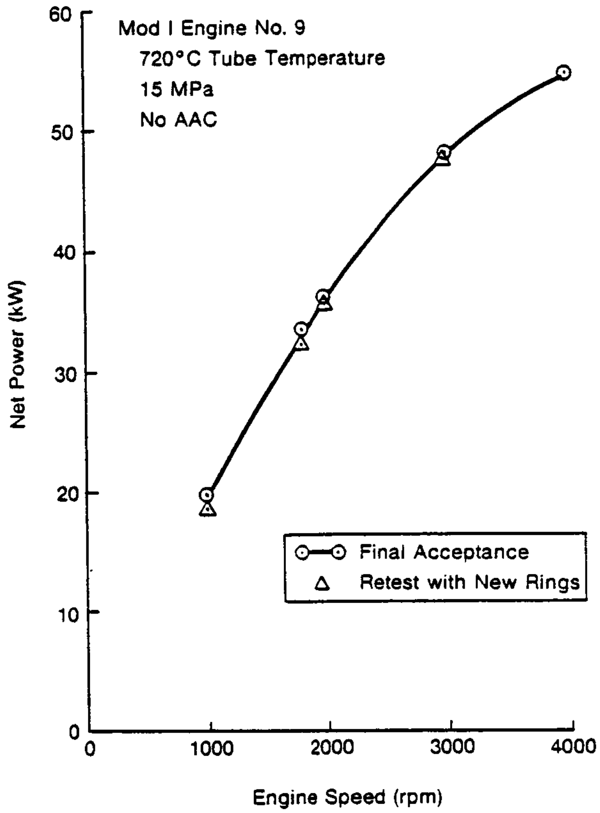


Figure 5-16 Comparison of MTI Final Acceptance Power Data to MTI Test with New Piston Rings

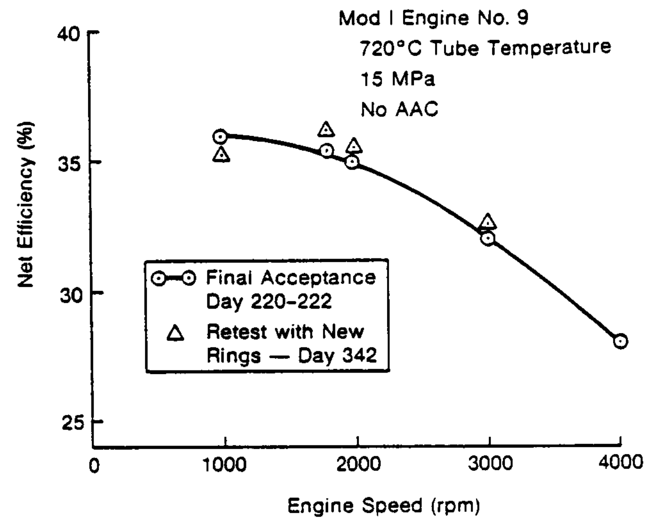


Figure 5-17 Comparison of MTI Final Acceptance Efficiency Data to MTI Test with New Piston Rings

VI. MOD II PRELIMINARY DESIGN

Introduction

The Mod II engine will be used to demonstrate the overall accomplishment of the ASE Program objectives in vehicle dynamometer tests. The development will be accomplished over a four-year period, progressing through design and analysis, formal design reviews, hardware procurement, engine assembly, test and development, followed by vehicle system assembly, test and development. This task is the focal point of all component development/Mod I engine development activities throughout the ASE Program.

MTI has the prime responsibility for all phases of Mod II development. In this capacity MTI will demonstrate that Stirling-engine technology is fully transferred to the U.S. in accordance with the goals of the program (i.e., MTI analytic and design tools/procedures will be developed, applied, and verified). Further, U.S. manufacturers and vendors will be used for most components.

The Mod II engine/vehicle system will be optimized within the constraints of accepted design practice and generic annular V-block geometry to achieve maximum fuel economy over the EPA driving cycle, and to provide acceptable vehicle response characteristics. Optimum Mod II engine geometry will be determined using the MTI Harmonic Stirling Code. However, final engine geometry will reflect the necessary compromises between optimum performance and practical engine design.

Performance predictions for this final engine geometry, along with system component performance characteristics, will be used in conjunction with the MTI Vehicle Simulation Code to evaluate engine/vehicle system performance. Vehicle drive-train characteristics will be optimized within the constraints of available drive-train components to achieve maximum fuel economy and specified vehicle performance goals.

Detailed layouts of the Mod II engine concept, as well as a preliminary parts' list, will be prepared in order to define a prime BOM configuration. Based on this definition, performance estimates and vehicle mileage projections will be prepared. After performance considerations have been satisfied, the mechanical systems will be analyzed to ensure that the proposed configurations are sufficient to accomplish the program goals.

Engine/vehicle performance characteristics will evolve as the design of the Mod II engine/vehicle system progresses. The impact of specific design changes will be evaluated on a continuing basis to ensure that program objectives are being achieved. Formal performance projections will be updated on a periodic basis to reflect the current state of the design process.

Analysis

The objective of this activity is to develop and refine the capability to predict engine performance and then to use this capability to select an optimized design for the Mod II and support the ongoing engine design activity.

During this report period, work was directed toward making a preliminary definition of the Mod II engine, finalizing the Mod II vehicle definition, developing computer models for all engine components and loss mechanisms, and using these models to optimize the engine design. The previously defined goals for the Mod II engine were updated to reflect the results of recent design and development work. A summary of these goals is presented in Table 6-1. Performance and weight were identified for all subsystems of the Mod II engine. For example, the EHS (preheater, combustor, atomizing air, and manifolding) was quantified in weight for each part, amount of atomizing air, pressure drop for the combustion gas flow, and excess air ratio (λ). In addition, full system goals were established; these include: engine weight, power and efficiency; vehicle fuel economy; and, acceleration rates. These goals will be used to monitor the design and development of the Mod II engine as the program progresses.

The Mod II vehicle definition was finalized. Earlier Mod II and reference engine studies had been based on the Pontiac Phoenix. In reviewing the current automotive picture, it was noted that the X-body cars, the Phoenix being one, are planned to be phased out of production in the near future. It was felt that the Mod II engine should be installed in a vehicle which would remain in production for some time in the future. It was noted that the GM A-body series of vehicles, which includes the Celebrity, were responsible for $\sim$20% of GM sales, were planned to be in production for several years, and were the right size for the Mod II engine. The Celebrity is also the best vehicle in its class in terms of fuel economy (31 mpg combined mileage relative to $\sim$26 mpg for fleet average of all cars of the same inertia weight). For the Mod II installation, the desired transmission to best match the characteristics of the Stirling engine would be an automatically-shifted, manual transmission with no torque converter. Since this type of transmission is currently not available on the automotive market, a four-speed, manual transmission will be used to demonstrate the concept.

TABLE 6-1
OVERALL MOD II GOALS

	Mod II Goal	
Weight		
Engine System (kg)	200.8	
Vehicle (Celebrity) Weight (lb)	3187	
Engine System Specific Weight (lb/hp/kg/kW)	5.5/3.35	
Vehicle Performance (Celebrity)		
Acceleration (seconds)		
0-60 mph	15.0	
50-70 mph	10.5	
0-100 feet	4.5	
Gradeability - Start and maintain 5 mph on 30% grade		
Fuel Economy - Combined (mpg)	40.4	
	SES	BSE
Engine Performance		
Maximum Power at 15 MPa 4000 rpm, kW (hp)	60 (80.5)	63.4 (85.1)
Maximum Efficiency (%)	42.2	43.2
Efficiency at 5 MPa, 2000 rpm (%)	37.3	40.1
Efficiency at 5 MPa, 1000 rpm (%)	33.9	36.1
Cold Start Penalty (g)	141	--
Emissions after 50,000 miles		
NO_x (g/mi)	< .4	
HC (g/mi)	< .41	
CO (g/mi)	<3.4	
Particulates (g/mi)	< .2	
Start-up		
Time to Drive Away (seconds)	30	

Packaging
Fit in a 1984 Chevrolet Celebrity
Manual four-speed transmission
No changes to front-suspension geometry or axle pick-up points
Vehicle compatability
- Maintain ground clearance
- No sheet metal changes

Weights

	Mod II Goal	8/84 Audit
Controls and Auxiliaries	82	106.3
Controls Total		65.5
Mean Pressure Control		31.0
Combustion Air Control		22.7
Digital Engine Control		11.8
Auxiliaries Total		40.8
Auxiliary Drive and Tensioner		
Starter		
Alternator and Pulley		
Battery, Case, and Cable		
Brackets		
Water Pump		
Basic Stirling Engine System	118.8	133.8
External Heat System	24.7	27.0
Flamestone and Insulation		1.0
Preheater Assembly		13.6
Combustor		4.0
Cover		7.3
Support/Transition Plate		0.5
Nozzle		1.0
Hot Engine System	24.2	25.5
Heater Head Assembly		13.9
Stuffer and Partition Wall		3.1
Regenerator		1.7
Flanges		6.8
Cold Engine and Drive Systems	70.0	81.3
Block		45.0
Crankshaft and Gears		9.6
Bearings and Caps		--
Connecting Rods (Including Crosshead)		4.5
Piston/Rod Assembly		4.9
Seal Housing Assembly		5.2
Oil Pump Assembly/Filter		2.8
H_2O Inlet/Outlet Stubs/Sump		2.9
Cooler		6.4

In defining the Mod II configuration, key components were analyzed, including the EHS, piston rings, and loss mechanisms for all components. An analytical model of the EHS was built to compare the types of EHS that could be utilized in the Mod II, namely; metallic versus ceramic preheater, and EGR versus CGR combustor for emissions control. The code models the complete EHS, including conduction losses, pressure drop, and both rows of the heater tubes. A comparison of the EHS performance for three of these systems is shown in Figure 6-1. The CGR system shows better performance than EGR, and the ceramic preheater is better than the metallic. The ceramic preheater, however, has a higher CSP which offsets the efficiency improvement in terms of overall vehicle system fuel economy and is a higher risk system. Therefore the metallic preheater with CGR was selected for the Mod II.

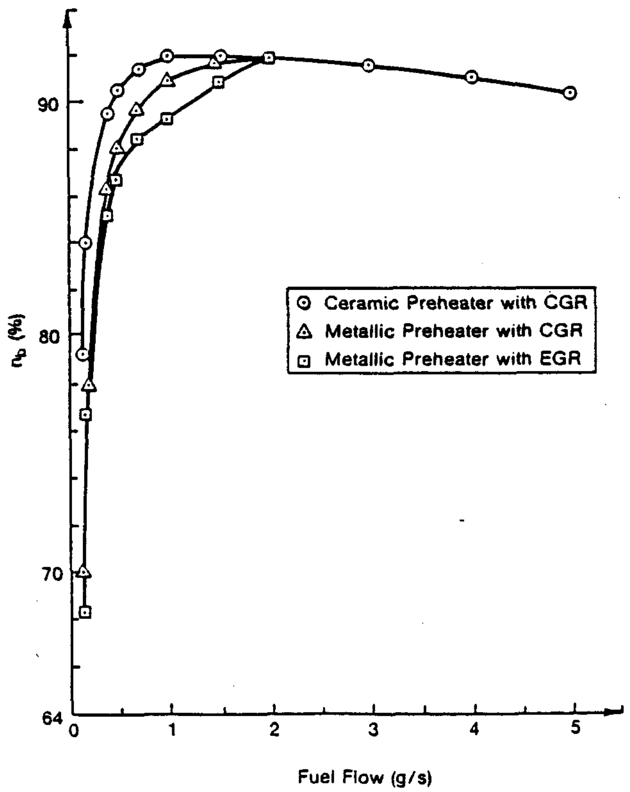


Figure 6-1 Upgraded Mod I Preheater CGR-EGR Performance Comparison

Engine tests of split-solid versus single-solid piston rings were conducted to assess the performance difference of the two systems. The power difference between the systems, normalized to maximum pressure levels is shown in Figure 6-2 and indicated lower power for the single-solid ring at low engine speed and higher power at high engine speed. This relationship implies higher leakage and lower friction with the solid ring. Since performance at low engine speeds is key to Mod II fuel economy, split-solid piston rings are retained in the Mod II design.

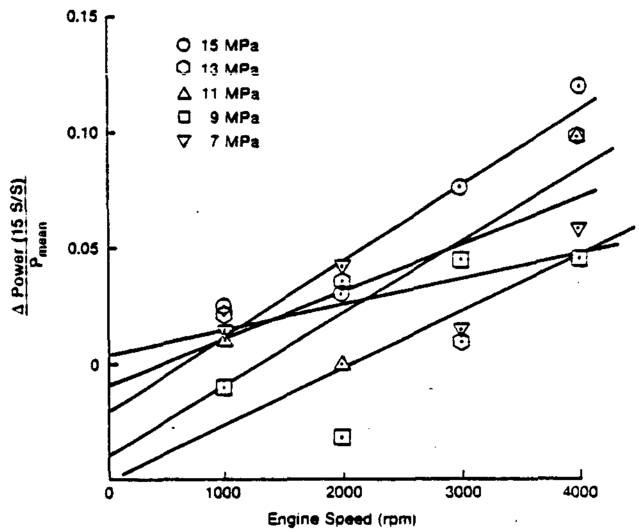


Figure 6-2 Power Difference/P_{mean} (Single-Solid, Split-Solid) (ASE Engine No. 5)

As previously noted, loss mechanisms were identified for all Mod II components. These losses include identification of all auxiliary power requirements, including alternator, combustion air blower, water pump, atomizing air compressor, radiator, oil pump, hydrogen compressor, and combustion air system pressure drops. Drive unit friction modeling, appendix gap loss representation, heater head heat transfer calculations, and EHS thermodynamic modeling were also documented. These data are not presented in this report for the sake of brevity; however, they can be found in MTI 85ASE428MT80, entitled, "Monthly Technical Progress Narrative Report for November 1984."

The first preliminary optimization of the Mod II engine was conducted in December. Cooperative and parallel efforts were conducted by USAB and MTI analysis teams. Some discrepancies were noted between the two sets of optimized engine geometry, which will be resolved by continued analyses during the next report period. This first optimization analysis incorporated the input modeling data previously referenced (given in MTI 85ASE428MT-80). These data will be updated and revised as analytical techniques are improved, and as definitions of auxiliary equipment and losses are modified and updated.

Preliminary Design

The objective of this task is to establish design procedures and specifications, then to utilize them to complete a preliminary design of the Mod II engine based on a rough estimate of the engine sizing. A cross section of the preliminary design of the Mod II is shown in Figure 6-3.

The preliminary design of the Mod II was initiated in January and has extended to a technology assessment conducted September 25-26, 1984. Preliminary design has been based primarily on the configuration of the RESD as updated in May 1983. The technology assessment constituted a review and selection of specific configurations and technologies to be incorporated into the final Mod II engine. Final engine detail dimensions will be determined by optimization analyses.

During this report period, preliminary design continued to focus on support of the durability test rig and prototype design drawings of the annular, two-manifold heater head and the annular cooler design. The durability test rig is a preliminary version of the Mod II CEDS, including the cast-iron V-block, roller-bearing single crankshaft, and new lightweight piston/rod/crosshead/connecting-rod systems. All drawings and follow-up ECNs required for fabrication of the durability rig were completed. It is anticipated that many of these CEDS design drawings will be incorporated into the final Mod II engine design with only minor changes.

Castings and machining drawings for the XF-818 annular heater head housing were completed. Drawings for the heater tubes, fins, and brazed assemblies were also completed. These drawings were required in support of manufacturability studies and trials for this new heater head design, which was viewed as having some development/manufacturing risk associated with it. Complete detail component drawings, brazing assembly, and final machining drawings were also completed for a prototype Mod II annular cooler. This was considered to have development/manufacturing risk, due to the brazed assembly of the cooler body and the closer spacing of cooler tubes. Considerable design support and study was also devoted to the prototype, integrated cast-iron V-block during the procurement and casting-development process. This has been a continuing effort to improve castability and correct design errors.

Technology Assessment

The Mod II technology assessment was conducted September 25-26, 1984. Its purpose was to decide what technologies would be designed into the initial Mod II detail design, identify risks associated with these technologies, and to decide what development activities were required to reduce these risks. The technologies selected for initial design of the Mod II engine are identified in the paragraphs that follow.

EHS

The CGR combustor (ribbon configuration) was selected in preference to the EGR, due to higher η_{EHS}, lower weight and better packageability. The conical fuel nozzle was selected over the BOM nozzle, as it allows a lower atomizing airflow with consequent higher η_{EHS} at low fuel flows. The metallic preheater was selected over the ceramic preheater, as the

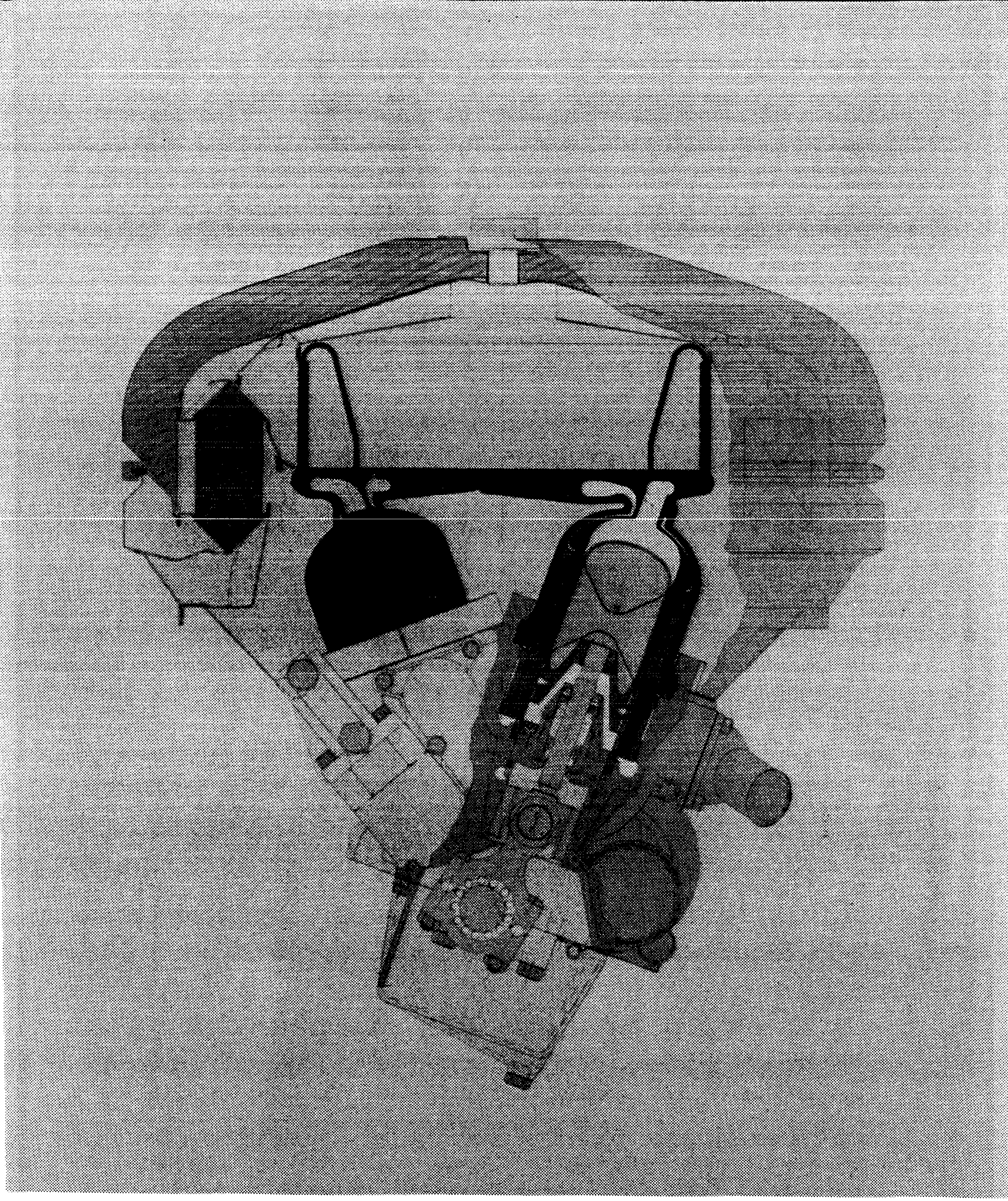


Figure 6-3 Cross Section of Mod II Engine

metallic offered reduced CSP and a lower risk for manufacturability, leakage, and durability. A separate Auburn igniter and Kanthal flamestone were also selected.

HES

Two parallel configurations were identified to be designed and developed for the Mod II heater head configuration. Configuration No. 1 will incorporate two manifolds and will employ XF-818 material. This configuration was selected due to its perceived low risk for manufacturing lead time. Configuration No. 2 will employ the higher strength HS-31 material for the housing material, and will have only one or perhaps no manifolds, to provide reduced engine dead volume. Both heater head configurations are to be fully interchangeable at all interfaces with other engine systems. Configuration No. 2 was selected for parallel development because it offers the potential performance benefit of an additional 1.5 points on efficiency and 0.7 mpg on vehicle mileage, together with a higher risk on manufacturing lead time. As a consequence of this parallel selection, two engine maps and two vehicle mileage projections must be developed.

Heater tubes were selected to be 2.5-mm I.D., 0.75-mm wall thickness, CG-27 material with Inconel 625 as an acceptable back-up material for development. The new single-tube, separate fins in densepack configuration and Inconel 601 material were chosen. Manifold/tube connections will be optimized to achieve uniform tube wall temperature and gas flow distribution.

An annular mesh-type regenerator was selected, which will have no outer shell or can, but will be press-fitted directly into the heater-head housing. The I.D. will be brazed to the partition wall, which was selected to be optimized, thin metallic wall. The annular gas cooler will be configured with a thin, high-strength inner wall to act as cylinder liner with 316L stainless steel as a tentative material selection, to be ion-nitrided for good wear on the cylinder bore. Stainless steel tubes will be brazed into the cooler shell.

CEDS

Roller bearings were selected for mains and connecting rods, due to their lower losses and low-speed load capacity. The crankshaft material was selected to be AISI 4320 H per ASTM A534, with inclusions rated better than ASTM A45. This material was selected to ensure strenth and durability under surface fatigue imposed by the roller bearings. A hardened steel-on-steel, squeeze-film-lubricated journal configuration was selected for wrist-pin bearings.

The integrated cast-iron V-block was returned to development and an interim multi-piece (brazed or welded) analog block design was selected for first build of the Mod II. This decision was an outcome of the difficulties associated with the first cast blocks from Motor Castings. The lightweight, single piston ring design was selected, with the piston rod attached to the crosshead by means of a thread, and to the piston base via a shrink fit.

The seal system selected consisted of the single-solid piston ring, the PL seal made of Rulon J, and the cap seal. There is some evidence that Rulon J main seals give longer life than does HABIA material. The cap seal was retained because it allows the seal housing to be vented to the P_{min} manifold so that the main seal is not subjected to cyclic gas pressures.

CONTROLS AND AUXILIARIES

The idle condition was selected to be 400 rpm at 2 MPa with an interim value of 0.15 g/s for idle fuel flow. The H_2 PCV will be actuated by an electric drive motor with speed-reducing gearbox and rack and pinion final drive element. A 4 ℓ H_2 gas storage bottle and two-stage H_2 gas compressor were selected, since a 10:1

pressure ratio will be required with the lower idle engine pressure.

Optimized lambda control was selected for the DAFC, together with a Micropump fuel-metering pump and the Hitachi air measuring unit, with no air throttle valve. The TI 9995 microprocessor with five boards was selected for the DEC. The combustion air blower will be driven by a high-speed, high-efficiency electric motor with no variator. The alternator will be a permanent-magnet, three-phase, 120 V, high-efficiency unit to operate up to 30,000 rpm. Dual radiator-cooling fans were selected, to operate in staged sequence. Vehicle selection was confirmed to be the Chevrolet Celebrity, in the 3125 lb inertia-weight class, with four-speed manual transmission.

ENGINE INITIAL DETAIL DESIGN

The technology assessment marked the end of the preliminary design phase for the Mod II engine. Initial detail design of the engine commenced immediately thereafter on October 1, 1985. During October-December 1984, substantial changes were made to the engine block design. The design was first formulated for a cast-iron configuration, and then translated to a machined analog of the casting, in accordance with the technology assessment decision. The water pump suction housing, provisions for mounting the water pump and H_2 compressor and the front main bearing were all deleted from the block casting and incorporated into a separate, aluminum engine front cover. Modified geometry was developed for the annular cold gas ducts under the coolers. These new changes will simplify and improve castability of the engine block, as well as reduce overall engine weight. Equally important, there will be greater assurance of obtaining sound metal in the thin wall separating adjacent cold gas ducts at the apex of the engine V. The casting design was reviewed with the technical staff of John Deere's nodular iron foundry, who offered further recommendations for improving castability.

A design was developed for a machined analog block, based on use of a single-piece forged billet. This will avoid the risks associated with brazed or welded joints of a multi-piece block design. The design allows near-exact simulation of the cast internal passages for cooling water and cold-connecting ducts by drilling from the outside surfaces of the block billet, with welded plugs to seal penetrations of the exterior walls.

An analysis of bending deflection of the crankshaft under connecting-rod loads was performed for maximum load conditions (4000 rpm, 15 MPa). The crankshaft, as analyzed, incorporated 35-mm journal diameters at all main and connecting rod bearings, to accommodate the existing Torrington and INA roller bearings. The results of this analysis are shown in Table 6-2. Bending deflection of the crankshaft constitutes angular misalignment for the roller bearings. The magnitudes of misalignment shown in Table 6-2, in particular for main bearings No. 2 and 3, exceed the maximum allowable misalignment recommended by roller bearing manufacturers by a significant margin. It was concluded from this analysis that crankshaft journal diameters should be increased to 38-40 mm to reduce deflection/misalignment to acceptable values.

TABLE 6-2

ANGULAR DEFLECTION MOD II CRANKSHAFT, PRELIMINARY DESIGN

Axial Location on Shaft	Crank Angle During Shaft Rotation	Max. Angular Displacement Minutes of Arc
Main Bearing M1	230	4.9
H Compressor Crank	230	5.4
Main Bearing M2	240	6.5
No. 1 Crank	300	4.1
No. 2 Crank	240	6.7
Main Bearing M3	250	8.6
No. 4 Crank	60	5.0
No. 3 Crank	140	3.6
Main Bearing M4	100	5.0

An analysis of roller bearing life was conducted by bearing specialists in MTI's R&D Division, which was later reviewed with technical staff of SKF Industries, Inc., one of the premier organizations in the world for design and manufacture of rolling-element bearings. This review indicated that life-multiplying factors selected during the MTI R&D analysis were overly optimistic. New SKF data were selected for the life-multiplying factors, and a revised analysis was performed for the new three-main-bearing crankshaft configuration. Results of the previous R&D analysis and the revised analysis based on SKF data are given in Table 6-3, in terms of L_{10} bearing life under maximum-power operating conditions. The Mod II engine goal is to ensure an L_{10} life of at least 100 hr for the entire seven-bearing system (three main and four connecting-rod bearings). With a predicted L_{10} system life of only 60 hr, it became apparent that the existing 35 mm Torrington and INA roller bearings could not ensure meeting the goal. Therefore, larger bearings for journal diameters between 38 and 40 mm, with higher dynamic load capacity, will be selected during the next report period.

TABLE 6-3

COMPARISON OF L_{10} LIFE PREDICTIONS FOR 4000 RPM MAXIMUM POWER CONDITIONS (35-MM JOURNAL DIAMETERS)

Location	Bearing Type	Predicted L_{10} Life, Hours Previous	Based on New SKF Data
Crankpin	INA	463	237
Main M2	Torrington	1308	670
Main M3	Torrington	1296	664
Main M4	FAG	5806	2974
7-Bearing System		117.3	60.

An analysis was performed of the effects of skew misalignment of the crosshead on crosshead-to-guide minimum clearance and crosshead lateral motion, for both straight cylindrical and barreled crosshead configurations. The analysis was done to guide the selection of design tolerances for misalignment of the separate crosshead with respect to the piston rod. One method of assembly being considered might allow a skew misalignment angle of up to 0.05 degree. The two geometries considered are defined in Figure 6-4, and results of the analysis are shown in Figures 6-5 and 6-6. The barreled crosshead was seen to offer an advantage in maintaining a greater clearance, however, at the expense of greater machining complexity and greater lateral motion at skew angles above 0.025 degree.

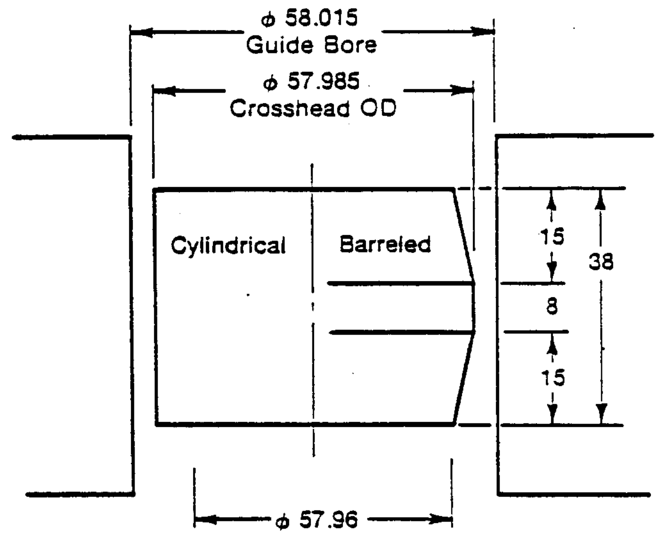


Figure 6-4 Significant Dimensions of Cylindrical and Barreled Crosshead Configurations

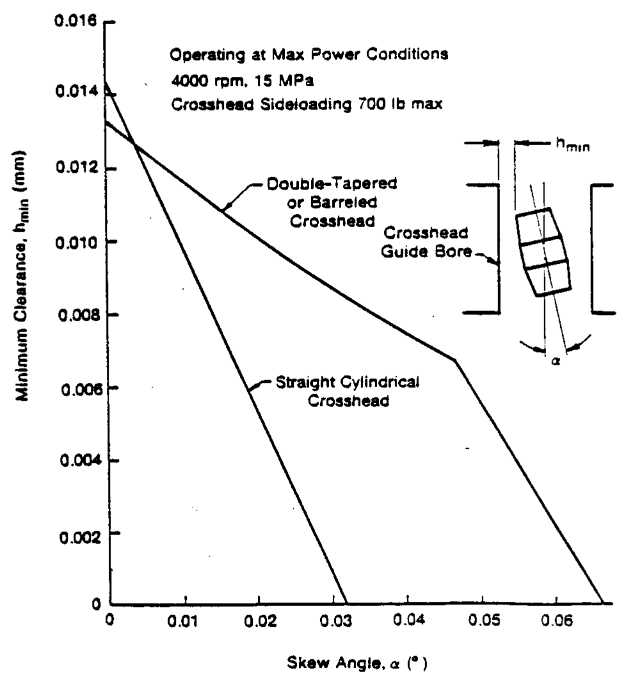


Figure 6-5 Minimum Crosshead Clearance versus Misalignment Skew Angle

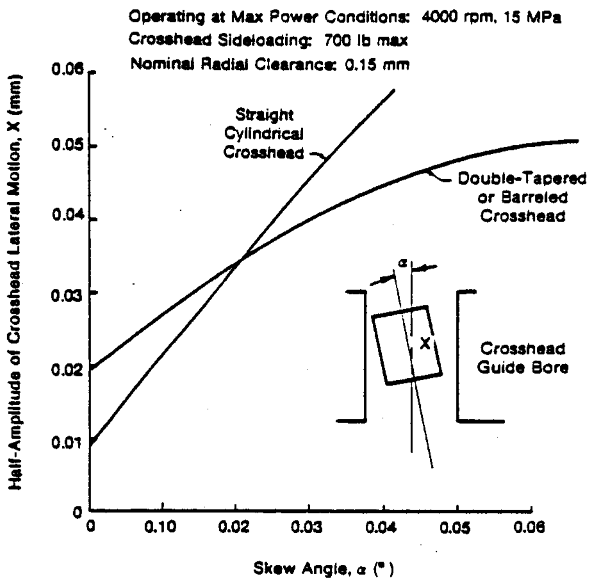


Figure 6-6 Lateral Amplitude of Crosshead Motion at Maximum Power Conditions versus Misalignment Skew Angle

Preliminary analyses of the Mod II cooling water circuit had indicated that system pressure drop would be higher than that of the Mod I. The annular cooler design presents to the cooling water a longer, higher flow-resistance path than does the Mod I cooler. The Mod II flow path splits at the inlet into two parallel paths, each only 4 tubes wide by √43 tubes long in the flow direction. A model of the water-side of the Mod II cooler was made and flow tested to verify the pressure-drop algorithm used in predicting the cooler performance.

Due to the higher pressure drop (and to some observed flow instability of the Mod I centrifugal water pump at high-head conditions), positive-displacement water pumps were considered for the Mod II engine. A Gerator gear type was selected, which will experience no flow instability at the higher head condition, and offers significantly higher efficiency than the centrifugal type. The water pump drive scheme is being designed for one-to-one gear drive off of the crankshaft, with an over-running clutch between the driven gear and the water pump shaft. This allows the pump to be driven by a small electric motor after engine shutdown, to circulate enough cooling water to prevent damage to O-rings by heat soak-back. The driven gear also carries a balance weight which is one component of the engine dependent-balance system.

The remainder of the detail design task for the Mod II will be completed during the next Semiannual report period. The BSE will be completed for presentation at NASA-LeRC BSE Design Review scheduled for March 1985. The controls and auxiliaries components of the SES will be completed for presentation at a NASA-LeRC SES Design Review in late July 1985.

Mod II Hardware Procurement

A schedule of hardware procurement has been laid out for the Mod II engine. The goal of the procurement schedule was to achieve a January 1986 start-up of the first Mod II engine in a BSE configuration. The highlights of the procurement are summarized in Table 6-4.

TABLE 6-4
HIGHLIGHTS OF MOD II HARDWARE PROCUREMENT

Item	Start Procurement	Delivery Date
Preheater	January 1985	Decmeber 1985
Combustor	March 1985	October 1985
Fuel Nozzles	March 1985	October 1985
Heater Quadrants	December 1984	October 1985
Regenerators	January 1985	October 1985
Piston/Dome/Rod Assembly	December 1984	October 1985
Gas Coolers	December 1984	October 1985
Seals	January 1985	October 1985
Analog Block	November 1984	September 1985
Crankshafts	November 1984	September 1985
Gears	February 1985	September 1985

Positive Seals

During this report period, a rig designed to evaluate metal bellows seals was fabricated and installed in a test cell with the required ancillary equipment. The test-head was designed to test two bellows in a back-to-back configuration as shown in Figure 7-1. Connections are provided to allow both the inner and outer surfaces of the bellows to be pressurized. This feature allows the pressure difference across the bellows to be limited and controlled as required. Two inductive transducers are also provided to monitor the motion of the bellows. Reciprocating motion is provided by a crankshaft/crosshead unit connected to the rod passing through the test-head. The rod is guided by two hydrostatic bearings, one above and the other below the test-head, which minimize any lateral motion. The rig is driven by a variable speed DC motor which gives a maximum speed of 4000 rpm.

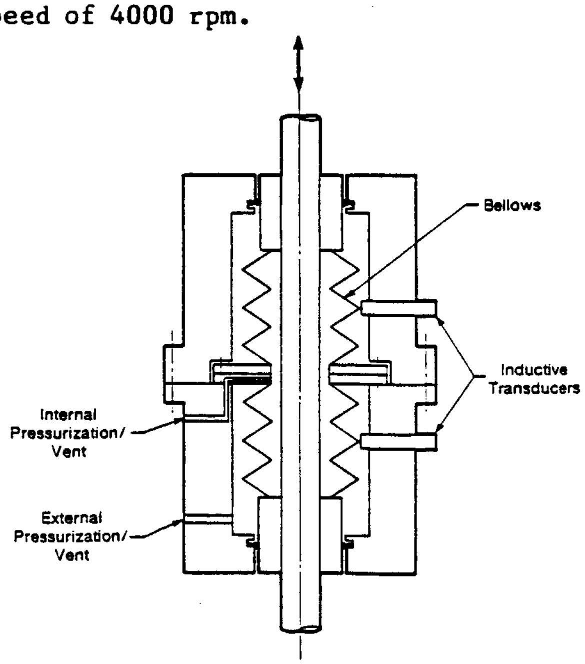


Figure 7-1 Bellows Seal Test Head

Initial tests were carried out on vendor designed and manufactured bellows. The bellows were designed for 30-mm stroke with a welded rippled construction. The material was AM350, 0.003 in. thick and each bellows had 90 convolutions. With minimal pressure difference, less than 50 psi, it was predicted that the life would exceed 3 x 10^8 cycles. The first natural frequency of the bellows was initially predicted to be 51 Hz.

The first exploratory tests were carried out with the inner volume of the bellows vented to atmosphere and the outer surfaces pressurized with nitrogen. The pressure was initially set at 10 psi and the drive speed increased slowly. At ~1000 rpm the bellows started to develop some form of resonant vibration as indicated by the inductive transducers. Increasing the nitrogen pressure slightly subdued the vibration and allowed the speed to be increased before the vibration started to appear again. The pressure was progressively increased up to 25 psi allowing speeds in excess of 2000 rpm. At this point one of the bellows failed due to fracture near one of the welded joints. The total test time prior to failure was 30 min.

The failed bellows was returned to the manufacturer who reported that it was typical of a torsional failure which could be induced by squirm. The manufacturer recommended that we repeat the test with the bellows pressurized externally at 25-30 psi throughout. Under these conditions another bellows failed after ~1 hr of total test time. Following this the manufacturers made measurements on bellows which had not failed and as a result revised their predictions for the first natural frequency to 42 Hz which was close to the frequency at which the rig had been operating when the failures occurred.

To investigate the natural frequency of the bellows, one was assembled in a special fixture and mounted on a shaker table. When vibrated in the axial direction the first natural frequency occurred at 42-43 Hz which confirmed the revised predictions. The bellows was also vibrated in the lateral direction and it was found that some modes of vibration were present in the 16-26 Hz range. At this time funding to continue the development effort was withdrawn and no further activity is planned for 1985.

Hot Piston Ring

In the current engine designs it is predicted that the losses associated with the appendix gap could be substantial and that a significant performance gain could be achieved by reducing these losses. One method of achieving this would be to introduce a piston ring in the upper section of the dome to essentially isolate the gap from the working-gas cycle. This requires that the piston ring be capable of operating unlubricated in the hottest part of the engine, and provide an effective gas seal without incurring excessive friction.

In order to establish whether the potential gains are real and attainable it was decided that short term exploratory tests should be carried out in a Mod I engine. A number of hot piston ring design concepts have been generated and potential materials/coatings for the piston rings and heater heads have been identified.

The concepts will be rated prior to choosing one for the final design. Pin-on-disk type material tests will be carried out at high temperatures in an inert/hydrogen atmosphere before selecting the final materials/coatings.

VIII. PRODUCT ASSURANCE

Quality Assurance Overview

The status of the ASE program Quality Assurance Reports (QARs) as of December 31, 1984 is presented below:

Open QARs (pending further analysis and/or NASA approval)	312
Closed QARs (total to date)	855
P-40 QARs	250
Mod I QARs	512
Upgraded Mod I QARs	371
Preliminary Mod II QARs	29
Total QARs in system	1167

Program QAR activity for the second half of 1984 is as follows:

New QARs (for six-month period)	219
P-40 QARs	55
Mod I QARs	42
Upgraded Mod I QARs	170
Preliminary Mod II QARs	29

Mod I QAR Experience

A summary of trend-setting problems documented via the QAR system is presented in Table 8-1 (shown on page 8-2), and Figures 8-1 through 8-4. Problems are defined as items that: 1) cause an engine to stop running; 2) prevent an engine from being started; or, 3) cause degradation in engine performance. Problems that fall into these categories must be minimized to provide acceptable engine performance, and maximize the mean time between failures.

Major trend-setting problems identified for individual units/assemblies that were established prior to June 30, 1983 are shown in comparison with the results of this reporting period and that of the previous Semiannual report period.

Table 8-2 is a summary of the operating times versus failures for all active ASE program Mod I/Upgraded Mod I engines.

TABLE 8-2

OPERATING TIME VERSUS FAILURES
AS OF DECEMBER 31, 1984

Engine No.	Operation Time (hr)	Mean Operating Time to Failure (hr)
3	1810	113
5	883	110
6	709	355
7	2339	1170
8	684	342
9	218	55
10	81	-

TABLE 8-1

MAJOR PROBLEMS SUMMARY

Established Prior to 6/30/83	% of Total	Reports From 6/30/83 to 12/31/83	% of Total	Reports From 1/1/84 to 6/30/84	% of Total	Reports From 6/31/84 to 12/31/84	% of Total
Moog Valve	86.5	1	4.5	1	4.5	1	4.5
Heater Head	54.0	3	12.5	3	12.5	5	21.0
Check Valves	47.1	3	17.6	2	11.8	4	23.5
Combustion Blower	44.5	6	22.2	6	22.2	3	11.1
Fuel Nozzle	35.1	6	16.2	13	35.2	5	13.5
Igniter	62.5	1	12.5	0	--	2	25.0
Preheater	31.8	7	31.8	1	4.6	7	31.8
Atomizing Air Comp./ Servo-Oil Pump	60.0	1	10.0	2	20.0	1	10.0
Combustor	41.2	4	23.5	2	11.8	4	23.5
Flameshield	31.5	3	65.8	0	--	10	52.7
PL Seal Assembly	30.8	3	11.5	8	30.8	7	26.9
Crankcase/Bedplate	--	2 new	66.7	0	--	1	33.3
Piston Rod	--	3 new	37.5	4	50.0	1	12.5

Mod I hours accumulated prior to 6/30/83 - 2438
Mod I hours accumulated from 6/30/83 to 12/31/83 - 2132
Mod I hours accumulated from 1/1/84 to 6/31/84 - 1848
Mod I hours accumulated from 6/30/84 to 12/31/84 - 2263

28% of hours accumulated prior to 6/30/83
24.6% of hours accumulated from 6/30/83 to 12/31/83
21.3% of hours accumulated from 1/1/84 to 6/31/84
26.1% of hours accumulated from 6/30/84 to 12/31/84

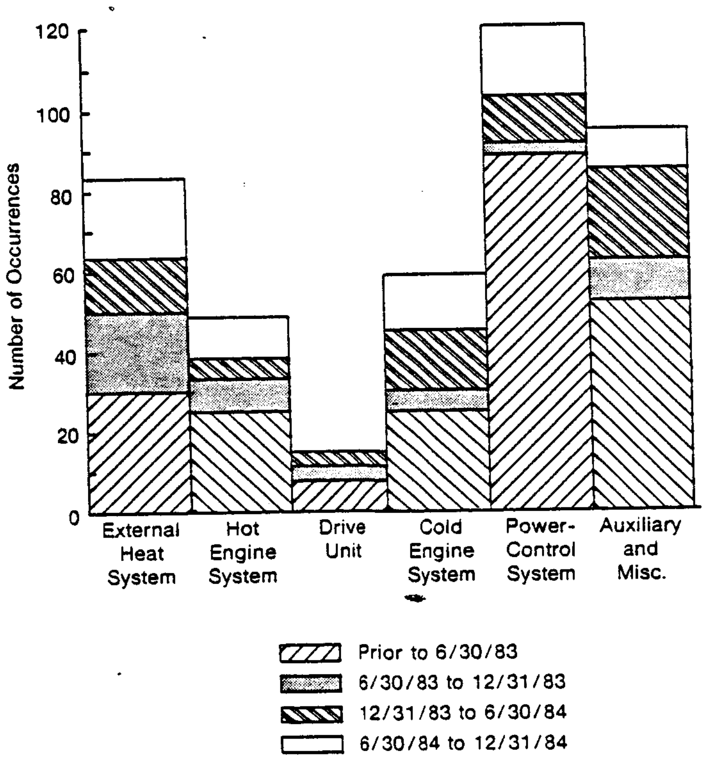


Figure 8-1 Upgraded Mod I Engine Major Failures and Discrepancies Through December 31, 1984

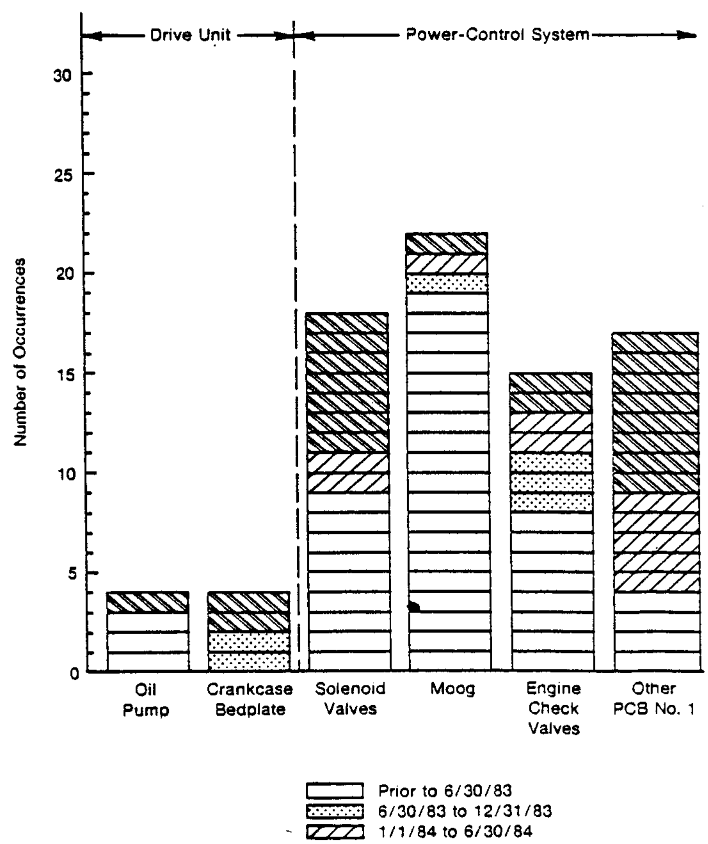


Figure 8-2 Drive Unit and Power-Control System Failures and Discrepancies Through December 31, 1984

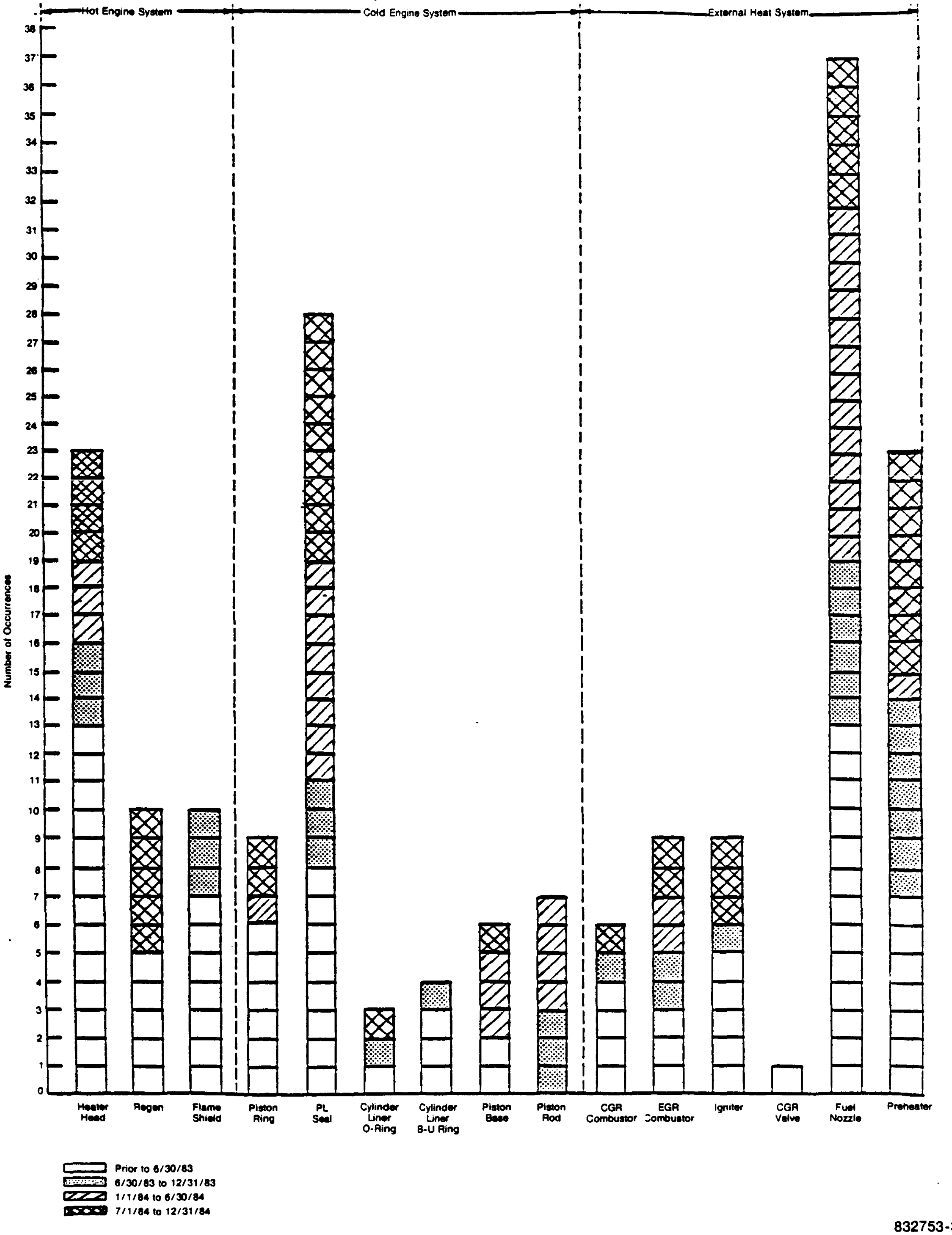


Figure 8-3 Hot Engine, Cold Engine, and EHS Failures and Discrepancies Through December 31, 1984

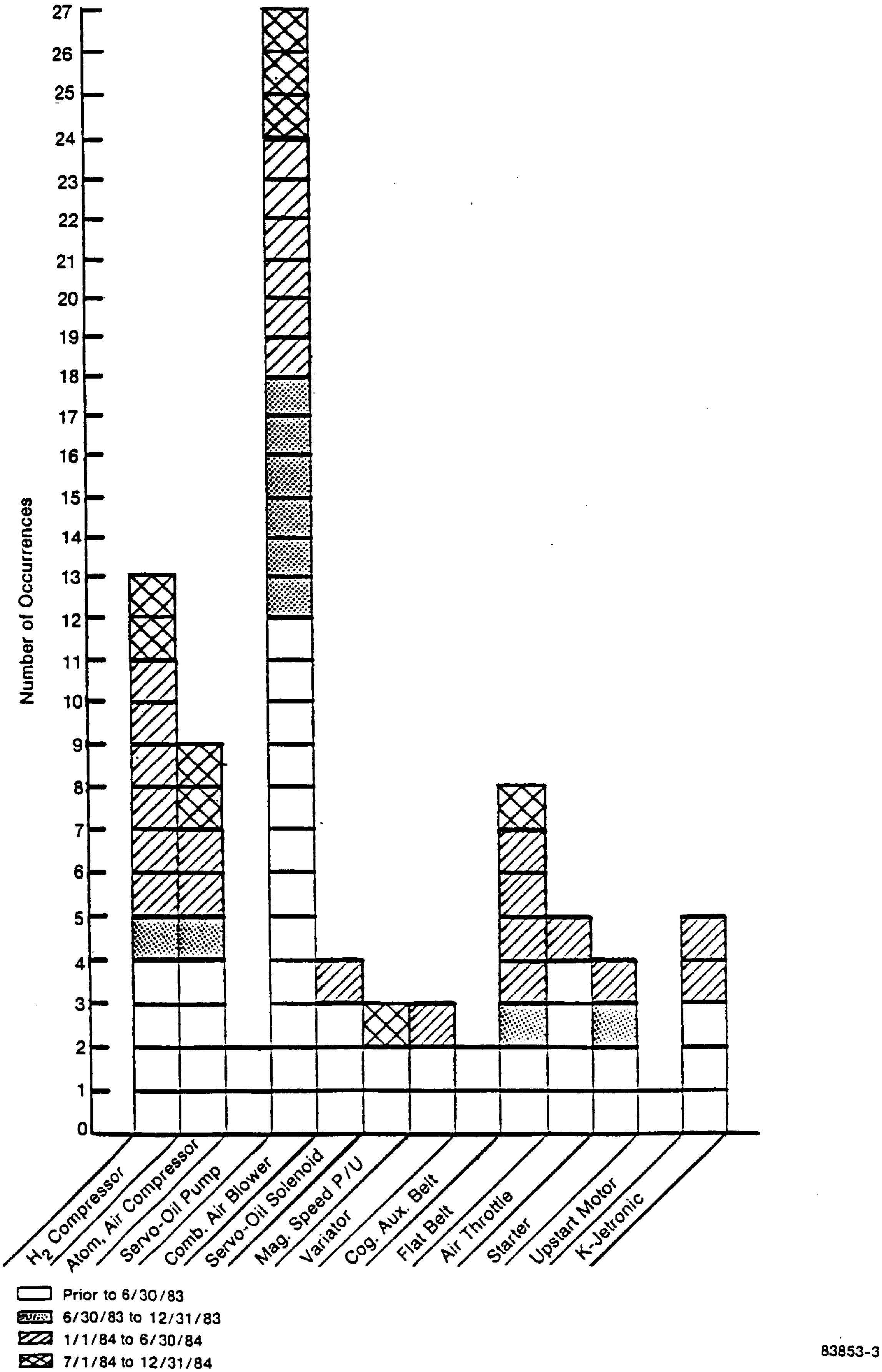


Figure 8-4 Auxiliaries and Miscellaneous Items, Failures and Discrepancies Through December 31, 1984

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15. Supplementary Notes

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16. Abstract

This is the seventh Semiannual Techincal Progress Report prepared under the Automotive Stirling Engine Development Program. It covers the twenty-sixth and twenty-seventh 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 development test activities on Mod I engines directed toward evaluating new technologies for potential inclusion in the Mod II engine. Activities covered were: test of a 12-tube combustion gas recirculation combustor; manufacture and flow-distribution test of a two-manifold annular heater head; a new piston rod/piston base joint; single-solid piston rings, and a Digital Air/Fuel Control. Also summarized are results of a formal assessment of candidate technologies for the Mod II engine, and preliminary design work for the Mod II. The overall program philosophy is outlined, and data and test results are presented.

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