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DOE/NASA/0032-30
NASA CR-180839
MTI 87ASE555SA11

# AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

**SEMIANNUAL TECHNICAL PROGRESS REPORT**
**FOR PERIOD: JULY 1 – DECEMBER 31, 1986**

Mechanical Technology Incorporated

February 1988

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

for
**U.S. DEPARTMENT OF ENERGY**
**Conservation and Renewable Energy**
**Office of Vehicle and Engine R&D**

(NASA-CR-180839) AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM Semiannual Technical Progress Report, 1 Jul. - 31 Dec. 1986 (Mechanical Technology) 59 p CSCL 13I

N90-11654

Unclas
G3/85 0234600

## NOTICE

# AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

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SEMIANNUAL TECHNICAL PROGRESS NARRATIVE REPORT
FOR PERIOD OF JULY 1 - DECEMBER 31, 1986

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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. The program team consisted of MTI as prime contractor, contributing their program management, development, and technology-transfer expertise; United Stirling of Sweden (USAB) as major subcontractor for Stirling-engine development; and AM General (AMG) as major subcontractor for engine and vehicle integration.

Most Stirling-engine technology previously resided outside of the United States and was 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, multifuel capability, low-emissions, and low-noise potential of the Stirling engine made it a prime candidate for an alternative automotive-propulsion system.

ASE program logic called for the design of a Reference Engine 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 the end of the program 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 to 1984 due to Government funding cutbacks.

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 an 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) that 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 upgraded Mod I engine was 100-lb lighter than the Mod I). The RESD has been revised periodically throughout the course of the program to



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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%.

The 1983 RESD configuration was changed substantially from previous designs to achieve these goals. The new design used 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, the Mod II design effort has been focused on translating the new RESD concepts into preliminary Mod II design drawings. Casting drawings of the annular heater head and single-piece V-block were implemented 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, 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.

The preliminary design phase of the Mod II was concluded in September 1984 with a Technology Assessment which selected specific technologies and configurations from competing contenders for each component of the Mod II engine. These component configurations were then moved into the initial detail design phase where the design was completed in preparation for manufacturing. Component development was intensified for certain components that needed further improvements; combustion gas recirculation (CGR) combustor, as well as controls and auxiliaries. The Spirit vehicle with upgraded Mod I engine No. 8, after the successful completion of its testing during the General Motors (GM) portion of the Industry Test and Evaluation Program (ITEP), was utilized to evaluate new controls and auxiliaries concepts to be incorporated in the Mod II. Analysis efforts were concentrated on finalizing loss models and algorithms for all aspects of the Mod II engine, which were then integrated into computer codes to be used in optimizing the engine.

In January 1985, the CGR combustor external heat system (EHS) was selected as the prime design, and the first optimization of the Mod II was completed. This optimization identified key engine parameters such as power and efficiency levels, bore and stroke, as well as key component design specifications, such as preheater plate aspect ratios, regenerator and cooler dimensions. This initial optimization was then honed and refined through many successively smaller iterations, including a preliminary final version presented at the Basic Stirling Engine (BSE) Design Review, until the design was finalized at the Stirling Engine System (SES) Design Review in August 1985. Improvements in the projected Mod II engine design and performance resulted from vendor feedback on the prototype Mod II V-block and heater heads, from component development tests of low idle fuel consumption and from extended Mod I engine testing of seals, piston rings and appendix gap geometry. During this period as well, a 1985 Celebrity with a 68.9 kW (92 hp) I4 engine was selected for the Mod II vehicle installation. This vehicle is representative of the vehicle class (3000 to 3500-lb front-wheel drive) that is extremely popular in the U.S. automotive market. It also has the best fuel economy in its class, thereby establishing a high-level internal combustion (IC) reference mileage for the Mod II evaluation.

The BSE and SES Mod II designs were both completed on schedule, and the design was approved by NASA for manufacture. The final performance predictions indicate the Mod II engine design will meet or exceed the original program goals of 30% improvement in fuel economy over a conventional IC powered vehicle, while providing acceptable emissions. This was accomplished while simultaneously reducing Mod II engine weight to a level comparable with IC engine power density, and packaging the Mod II in a 1985 Celebrity with no external sheet metal changes. The projected mileage of the Mod II Celebrity for the combined urban and highway CVS cycle is 40.9 mpg which is a 32% improvement over the IC Celebrity. If additional potential improvements are verified and incorporated in the Mod II, the mileage could increase to 42.7 mpg.

The Mod II engine start date of 31 January 1986 was met, using a machined analog V-block and manifold-type (configuration 1) heater heads in place of the cast block and no-manifold (configuration 4) heater heads, which were not complete.

During this report period an additional 203 hours were accumulated on the Mod II analog block engine with configuration 1 heater heads, bringing the total to 223. Once hardware and mechanical development problems were resolved, characterization of the Mod II BSE was begun. These tests were completed in January 1987.

The fatigue test of the cast V-block was successfully completed, while work continued on the first cast V-block engine for the motoring rig. By the end of the period, installation of the rig in the cell was progressing. Work on the configuration 4 heater head commenced.

In support of the Mod II effort, existing Mod I and upgraded Mod I engines were used to develop Mod II components and SES hardware. Cold room testing of the single solid and split piston rings, and various combinations of both low-temperature fluorocarbon and ethelene propylene O-rings were completed. Dynamometer testing of the entire Mod II mean pressure control and blower/alternator systems was conducted on the Spirit vehicle.

The Mod I and upgraded Mod I engines reached 16,588 hours in support of Mod II development and on the NASA technology utilization (TU) van. Installation in the van was completed, and the van was shipped to Langley Air Force Base, Virginia, where it logged 319 hours and 1200 miles as a flight line service vehicle.

During the next reporting period, performance characterization of the Mod II in BSE configuration will be completed and SES characterization will commence. The configuration 4 heater head will be completed, installed, and tested. The motoring rig engine will complete 10 hours at maximum pressure and speed to validate the engine drive system design.

# I. SUMMARY

## Mod II Engine Development

During this reporting period, progress was made toward the performance characterization of the Mod II engine, and by the end of the period, the engine hours had increased by 203 hours to a total of 223. Performance evaluation of the Mod II Basic Stirling Engine (BSE) began. During the semi-annual period, the water pump was characterized and flow patterns inside the two-manifold heater head assessed. Orderly progress toward completion of the characterization was initially hampered by problems in the cold engine and drive system (CEDS) and lack of satisfactory replacement hardware.

At first, engine operation was with low-pressure heater heads. These were limited to a maximum pressure of 10 MPa because of substandard castings resulting in concern for possible fatigue failure problems at higher pressures. The high-pressure heads were installed in the engine in October 1986. Mechanical problems associated with coolers, high external heat system temperature gradients, and crosshead/liners were resolved and BSE performance characterization begun.

Early in the report period, engine operation was observed to deteriorate due to premature wear of the cylinder liners and piston rings. This was caused by a basic incompatibility between the rulon-LD piston rings and the chromium-plated coolers. Substitution of ion-nitrided coolers corrected this deficiency.

Problems with high-temperature gradients in the external heat system were resolved after a series of investigative tests. A combination of hardware was identified that gave an acceptable differential.

A failure of the crosshead liner, similar to those failures experienced previously in this component, was corrected by changing liner material to a better grade of cast-iron and increasing clearances.

Fatigue testing of the cast V-block was completed following epoxy resin impregnation to eliminate porosity. Buildup of the first cast block engine for the motoring rig was initiated.

A major effort was started to complete design and manufacture of the configuration 4, no-manifold heater heads. A finite element stress analysis of the design was completed.

## Mod I Engine Development

Program emphasis continued to shift away from Mod I and into Mod II development. In the NASA Technology Utilization (TU) Program, the van was completed and delivered to the Air Force, where 319 hours and over 1200 miles were accumulated. The Ft. Belvoir Army Generator Set Program was terminated due to funding limitations after an incident in which leaking hydrogen was ignited, resulting in some damage to the generator set. Operation of the Spirit vehicle continued, and the entire Mod II Mean Pressure Control (MPC) System, including 3-volume compressor, 2-tank hydrogen storage system, and electrically driven power control valve, was successfully tested. Engine No. 6 completed a total of 4060 hours endurance testing, whereupon it was torn down, inspected, and found to be in satisfactory condition. The cold room seals testing at United Stirling was completed.

## Component and Technology Development

Work continued on Mod II component development and design substantiation. The conical fuel nozzle was incorporated into the Mod II engine, demonstrating reduced heater head temperature variation and nonplugging operation. Evaluation of engine sealing/cranking capability at low ambient temperatures was performed. Significant progress was made in the validation of mean pressure control system hardware and software. The entire mean pressure control system was successfully tested in the Spirit. The system was able to demonstrate engine idle at 2 MPa and 400 rpm, a major milestone toward Mod II development. Development of the electrically driven blower and battery charge systems continued, with emphasis on improving electronic hardware durability of the charge system.

Fatigue tests of all major high-pressure engine hardware, including all the engine and power control system components, were completed.

## External Heat System Development

The primary goal of the external heat system (EHS) is low emissions while maintaining high efficiency for a 30:1 fuel turndown ratio in a minimum volume.

The design must consider durability, heater head temperature profile, and expected use of alternative fuels while recognizing the significant cost impact of system size and design.

Emphasis during the second half of 1986 was directed toward developing Mod II combustor durability and reducing heater head $\Delta T$. Extensive rig testing of the Mod II center-igniter conical fuel nozzle was also conducted during this report period. Development of a 3-hole conical nozzle for the upgraded Mod I was completed.

Ceramic preheater development continued with the successful fabrication of one test block and completion of over 300 thermal cycles in a test rig.

### Mod II Combustion System Performance

The objective of Mod II combustion system performance is to meet the program emissions goals:

| | g/mi |
|---|---|
| $NO_x$: | 0.4 |
| CO: | 3.4 |
| HC: | 0.41 |
| Particulates: | 0.2 |

In addition, soot-free combustion (Bacharach $\leq$ 2) and low heater head temperature variation ($\Delta T \leq 100°C$) over a 30:1 fuel turndown ratio are required.

In order to meet the $NO_x$ requirement, combustion gas recirculation (CGR) is used.

Emissions, CGR, soot, combustor surface temperatures, and EHS pressure drops were measured using the Mod II engine.

Initial testing, at 720°C rear row set temperature (Table 2-1), revealed soot levels and heater head $\Delta T$s which exceeded program goals at fuel flows of $\sim$0.5 g/s. Soot levels decreased when an instrumented combustor (used to measure CGR) was replaced with a new combustor. High $\Delta T$s were unchanged. Realignment of the CGR ejectors appeared to lower the $\Delta T$s somewhat, but they were still unacceptably high at low fuel flows.

TABLE 2-1
MOD II ENGINE EMISSIONS DATA

| Engine (rpm) | $P_{mean}$ (mpa) | $\dot{m}_{fuel}$ (g/s) | $\dot{M}_{AA}$ (g/s) | $\lambda$ | CGR (%) | EICO (g/kg) | $EINO_x$ (g/kg) | EIHC (g/kg) | Soot Number | $\Delta T_T$ (°C) |
|---|---|---|---|---|---|---|---|---|---|---|
| 1000 | 4.7 | 0.50 | 1.04 | 1.24 | 21.0 | 55.0 | 1.9 | 1.90 | 65 | 135 |
| 1000 | 6.0 | 0.58 | 0.84 | 1.41 | 24.5 | 9.9 | 4.6 | 0.11 | 27/78 | 112 |
| 1004 | 6.0 | 0.58 | 0.83 | 1.41 | 25.3 | 16.1 | 4.6 | 0.16 | 40/67 | 100 |
| 1000 | 6.0 | 0.54 | 0.86 | 1.48 | -- | 0.85 | 4.3 | 0.08 | 2 | 71 |
| 1000 | 6.0 | 0.56 | 0.85 | 1.28 | -- | 0.13 | 3.6 | 0.13 | 16/7 | 92 |
| 1500 | 6.9 | 0.87 | 0.82 | 1.26 | -- | 3.6 | 3.9 | 0.10 | 2 | 91 |
| 1000 | 6.17 | 0.54 | 0.82 | 1.34 | -- | 1.8 | 4.5 | 0.11 | 3 | 161 |
| 1000 | 6.0 | 0.59 | 0.50 | 1.31 | -- | 44.0 | 5.9 | 0.65 | 100 | 93 |
| 1000 | 6.0 | 0.57 | 1.07 | 1.30 | -- | 5.0 | 2.0 | 0.10 | 24/16 | 109 |
| 2500 | 9.0 | 1.71 | 0.84 | 1.21 | -- | 8.9 | 2.0 | 0.09 | 2 | 65 |
| 704 | 5.0 | 0.41 | 0.84 | 1.30 | -- | 29.0 | 2.2 | 0.38 | 35/27 | 126 |

Emissions and soot data are shown in Figures 2-1* through 2-3. Figure 2-4 indicates the measured CGR levels. Rear row temperature spreads are illustrated in Figure 2-5. EHS and heater head pressure drops are given in Figures 2-6 and 2-7. $NO_x$ emissions, % CGR, and pressure loads were consistent with predictions.

The increases in CO, soot levels and temperature spread, at fuel flows below 1.0 g/s, were contrary to expectations.

*Figures are at the end of this section, beginning on page 2-14.

A series of tests was run to determine the cause of the high $\Delta$Ts:

- Rotate EHS 180°
- Increase $\lambda$
- Replace BOM nozzle with 3-hole conical nozzle.

The first test was designed to indicate whether hardware or assembly-induced flow asymmetries could be responsible. While some reduction in $\Delta$T at high fuel flows occurred, little change on the peak $\Delta$T at lower fuel flows occurred. Rotating only the combustor and nozzle also produced little improvement in the temperature spreads.

The purpose of the second test was to evaluate the influence of free convection on EHS flow patterns due to the 20° tilt of the engine. If buoyancy had an effect, there would be a pronounced change in flow patterns (reduced $\Delta$T) at low flows as $\lambda$ (mass flow) increased, since the ratio of free to forced convection would decrease. This was convincingly demonstrated (Figure 2-8) as $\lambda$ was varied from 1.3 to 1.9. The improvement in $\Delta$T was also influenced by improved air/fuel (A/F) mixing at lower flows due to the increased combustor pressure drop.

The substitution of the 3-hole conical nozzle for the BOM nozzle was intended to demonstrate the effect of tilt on internal nozzle fuel distribution. With the BOM nozzle it is possible for fuel to pool at the lower holes at idle conditions. The conical nozzle has only 3 versus 12 holes, located closer to the nozzle centerline, and a higher fuel pressure drop. This nozzle proved to be much less sensitive to tilt and low flow $\Delta$Ts were dramatically reduced, as shown in Figure 2-9.

In order to provide for ignition, a modification was made to one of the 3-hole conical-nozzle retaining bolts to allow insertion of an igniter (Figure 2-10).*

The 3-hole conical nozzle was left in service in the Mod II engine. While heater head $\Delta$Ts were significantly lower compared to the BOM nozzle, they tended to be unpredictable and to increase with time. Continued monitoring of the Mod II engine indicated that the probable causes of the behavior were due to multiple reasons:

- Improper nozzle insertion
- Random deformation of the combustor
- Tearing and cracking of the combustor shell, particularly near the igniter ground tube
- Severe overheating of the combustor at 820°C, causing warping and misalignment.

Design was completed for an improved CGR combustor. The system, which improves nozzle-to-combustor alignment and incorporates the 3-hole conical nozzle with an integral igniter, will be procured and tested during the next reporting period.

In an effort to improve low fuel flow soot and $\Delta$T, a Mod I reduced-diameter mixing tube (19 mm vs 26 mm) CGR combustor was tested in the performance rig with the BOM and conical nozzles.

Results with the BOM nozzle indicated an improvement in heater head $\Delta$T and soot and similar $NO_x$ emissions, compared to the standard-diameter-tube combustor. Tests with the conical nozzles were inconclusive because of rig and hardware problems.

*The 3-hole conical nozzle was originally designed for use with an exhaust gas recirculation (EGR) combustor that had a separate igniter. The Mod II CGR combustor does not allow for a separate igniter.

A Mod II reduced-diameter-tube combustor will be manufactured and used to complete the evaluation of the conical nozzles in the Mod II performance rig.

Center-Igniter Conical Nozzle Development

This nozzle was developed for the Mod II engine with the CGR combustor. It was intended to be a one-for-one replacement of the BOM nozzle.

Extensive testing of the center-igniter conical nozzle was conducted during the latter half of 1986. Table 2-2* defines the 15 configurations evaluated.

Testing of No. 000 through 006 was hampered by the failure of the lower O-ring, which deformed and extruded into the fuel passage (Figure 2-11). This allowed air and fuel to mix internally, leading to a nonuniform spray and high heater head $\Delta$Ts. Hardware modifications were made to correct this problem.

All the configurations tested, up through No. 008, were designed to reproduce the flow area of the 3-hole conical nozzle. This resulted in a narrow annular passage whose concentricity to the center electrode was sensitive to assembly procedures. As a result, a skewed flow pattern and plugging of the narrow passage could result.

Assembly No. 008 attempted to reproduce the 3-hole conical nozzle in two dimensions. The hydraulic diameter of the annular passage of No. 008 was approximately the same as that of the original 3-hole nozzle.

Increasing this annular area introduced a new problem: a reversing of the atomizing airflow. Modifications beyond No. 008 were directed at correcting this phenomenon.

Soot numbers for Nos. 008 and 009 are shown in Figure 2-12. Configuration No. 014 resulted in acceptable $\Delta$Ts (Figure 2-13) but at the sacrifice of $NO_x$ and soot (Figures 2-14 and 2-15).

Several configurations were also tested in the free burning rig (FBR) to observe the flame pattern. The observations indicated the following:

- A wide spray pattern causes the flame to impinge on the CGR mixing tubes, which blocks flow through the tubes. The result is high soot levels.
- Breakdown of the seals results in a skewed flame, which causes high $\Delta$ts and soot.
- Flashback can occur whereby the flame will periodically move from its normal, stabilized position outside the swirl region to another stabilized position inside the swirler. It is believed this was responsible for a wide scatter in the soot numbers.

The tests also indicated that a narrower, well defined, stable spray pattern may be achieved by reducing the atomizing air swirl angle and decreasing the OD of the annular exit. Designs to include these changes will be built and tested during the next reporting period. An improved sealing mechanism will also be used.

3-Hole Conical Nozzle Development

The 3-hole conical fuel nozzle continued to perform well during the latter half of 1986. Table 2-3 lists the plug-free operation of the nozzle to date.

Nozzles No. 2 and 3 were removed from service due to damage from overheating, which resulted from the failure of the atomizing air compressor motor.

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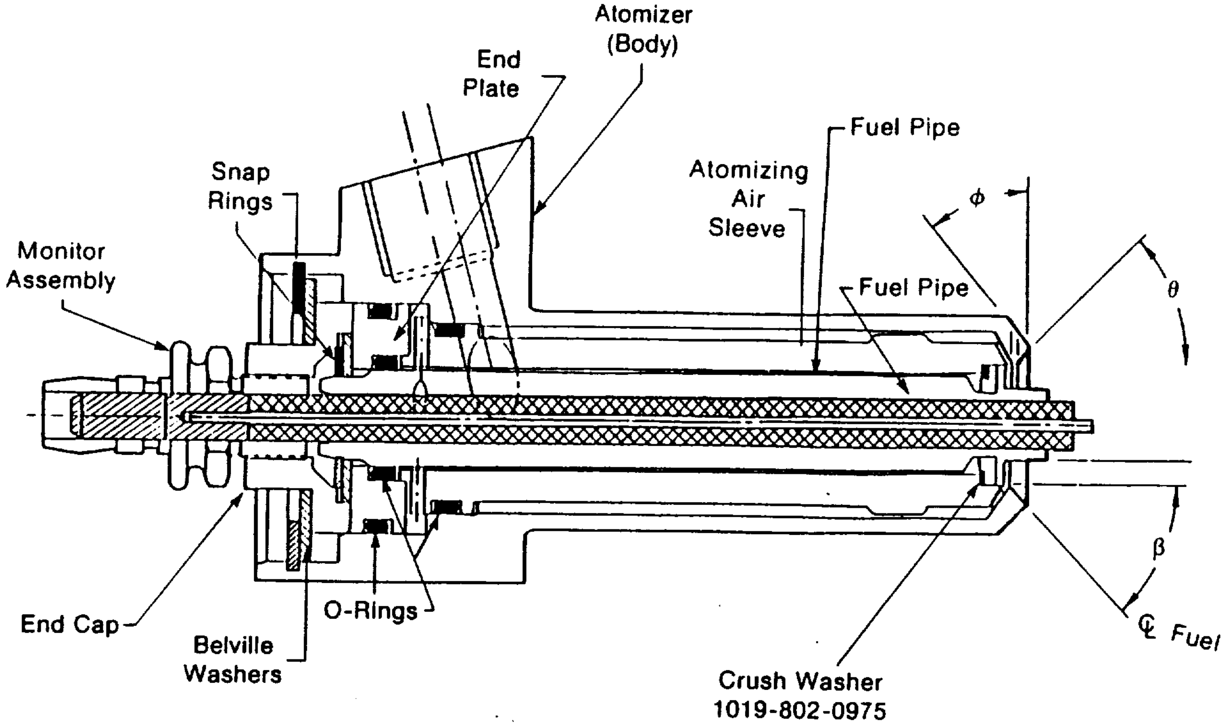
*Table 2-2 is on following page.

TABLE 2-2
MOD II CENTER IGNITER CONICAL NOZZLE DEVELOPMENT SUMMARY

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| | 000 | 001 | 002 | 003 | 004 | 005 | 006 | 007 |
|---|---|---|---|---|---|---|---|---|
| Atomizer body: | | | | | | | | |
| θ(°) | 30 | → | → | → | 15 | 30 | 45 | → |
| φ(°) | 35 | → | → | → | 35 | 35 | Reduced L/D | → |
| Fuel pipe: | | | | | | | | |
| β (°) | 30 | → | → | → | 15 | 30 | → | → |
| No. of holes | 6 | → | → | → | 6 | 6 | → | → |
| Diameter of holes (mm) | 0.3-0.35 | → | → | → | 0.3-0.35 (no cone) | 0.3-0.35 | → | → |
| Atomization sleeve: | | | | | | | | |
| Swirl angle (°) | 55 | → | 55 | 45 | 55 | → | 45 | 45 |
| No. of holes/slots | 12 slots | → | 12 slots | 12 slots | 20 holes | → | 12 slots | 12 slots |
| Dimensions (mm) | 0.575x0.039* | → | 0.575x0.039 | 0.575x0.039 | 0.3 | → | 0.575x0.039 | 0.575x0.039 (moved O-ring) |

| | 008 | 009 | 010 | 011 | 012 | 013 | 014 | 015 |
|---|---|---|---|---|---|---|---|---|
| Atomizer body: | | | | | | | | |
| θ(°) | 45 | 45 | → | 30 | 20 | 0 | 0 | 0 |
| φ(°) | Reduced L/D | 45 | → | 35 | 35 | 35 | 35 | 45 |
| Fuel pipe: | | | | | | | | |
| β (°) | 30 | → | → | 15 | 20 | 30 | 20 | 20 |
| No. of holes | 3 | → | → | 9 | 8 | 6 | 8 | 8 |
| Diameter of holes (mm) | 0.4-0.45 | → | → | 0.4-0.45 | 0.3-0.35 | 0.3-0.35 | 0.3-0.35 | 0.3-0.35 |
| Atomization sleeve: | | | | | | | | |
| Swirl angle (°) | 45 | → | → | 45 | 55 | 45 | 45 | 45 |
| No. of holes/slots | 12 slots | → | → | 12 slots | 12 slots | 12 slots | 12 slots | 12 slots |
| Dimensions (mm) | 0.575x0.039* (moved O-ring) | → | → | 0.575x0.039 | 0.575x0.039 | 0.575x0.039 | 0.575x0.039 | 0.575x0.039 |

*slots undersized - not as listed

As noted before, a redesign of the mounting flange and igniter for Mod II application was made and will be tested in the first half of 1987.

TABLE 2-3
3-HOLE CONICAL NOZZLE PLUG-FREE OPERATION

| Installation | Nozzle No. | Hours |
|---|---|---|
| Mod II Engine No. 1 | 4 | 75.5 |
| Upgraded Mod I, Engine No. 6 | 1 | 275.7* |
| NASA TU Van | 6 | 38.0 |
| | 2 | 167.4* |
| | 3 | 225.4* |
| Performance Rig | 5 | 18.6 |
| | | 800.6 |

*Inactive

Ceramic Preheater Development

The objective of ceramic preheater development is to demonstrate the feasibility of fabricating low-cost, ceramic matrices by producing mixed-oxide preheater blocks that will have less than 5% leakage after 300 thermal cycles.

After it was discovered that most of the 15 binders used in fabricating the previous blocks were well beyond their shelf-life, new binders were used and a new block was manufactured. The block was tested for both heat transfer performance and durability. All tests were successful. The 2% initial leakage remained 2% after 316 thermal cycles. The performance results are given in Figure 2-16. Additional blocks will be procured in the next report period.

## Hot Engine System Development

The objectives of this task during the second half of 1986 were

- Determine the pressure drop characteristics of alternate porosity and used Mod II regenerators

- Evaluate the internal flow loss of the Mod II, manifolded heater head.

Regenerator Pressure Drop

A set of Mod II regenerators was flow checked after 138 hours of engine operation. Only a slight increase in pressure drop was noted compared to the same regenerator when new (Figure 2-17).

Alternate porosity (65 and 70%) regenerators were also evaluated prior to engine test. Compared to the Mod II design porosity of 68%, pressure drop varied with porosity, as expected (Figure 2-18).

Mod II Heater Head Internal Flow Losses

Cold-flow measurements were made on two Mod II manifolded heater head quadrants to determine static pressure variations in both flow directions, i.e., cylinder to regenerator and vice versa (Figure 2-19). Results indicated that the stuffer channel has greater than predicted flow losses. The additional losses are consistent with swirling flow.

## Materials and Process Development

The goal of this task is to use low-cost, low-strategic-element-content materials in the ASE, capable of surviving a 3500-hr automotive duty cycle. Additionally, this task provides materials support to the Mod II design and component development activities.

Accomplishments during the second half of 1986 included proof-fatigue testing of the Mod II piston rod/base/crosshead for single solid piston rings and the Mod II ductile cast iron V-block.

Activities planned for the next report period include continued proof-fatigue testing of the Mod II piston rod/base/crosshead for single solid piston rings and the Mod II Configuration No. 4 heater head casting.

Mod II Piston Rod/Base/Crosshead Fatigue Test

The objective of this subtask is to develop a fatigue-resistant piston rod/base/crosshead assembly for use with Mod II single solid piston rings. A design change to unvented piston rods necessitated the proof-fatigue testing of the new configuration. The piston rods are the same as those used with Mod I type split-solid piston rings, except that the $P_{min}$ hole is removed and a 1.5-mm hole is put along the centerline of the piston rod through the center drill hole in the shank end to the countersink as shown in Figure 2-20.

Two test assemblies were manufactured and tested in this report period. The test assemblies were subjected to the test cycle sequence shown in Table 2-4. Acceptance will be obtained after completing Step No. 4, $10^6$ cycles at the 150% load level, which includes a safety factor.

TABLE 2-4
MOD II PISTON BASE/ROD/CROSSHEAD
TEST CYCLE

| Step No. | No. of Cycles | % of Full Load |
|---|---|---|
| 1 | $10^7$ | 100 |
| 2 | $10^6$ | 120 |
| 3 | $10^6$ | 140 |
| 4 | $10^6$ | 150 |
| 5 | $10^6$ | 160 |
| 6 | $10^6$ | 180 |

The first test assembly completed 1.2 x $10^6$ cycles at the 100% load level. Failure was caused by fatigue, which originated at the root of the top thread in the shank of the piston rod. It appeared that the preload applied to the shank was too low. The preload is applied by torquing the shank into the fixture to 64 N·m, which is the normal preload for the standard Mod II piston rods.

To confirm that the correct preload would be applied to the next piston rod, a calibration curve was produced showing the extension versus the applied torque on the shank of the piston rod. While making this curve, it was noticed that the shank would develop a permanent extension when torqued to 80 N·m. This indicated that the shank was being prestressed beyond the elastic limit of the RC 50 hardness material.

The second assembly was tested to determine if the results of the first test were just an anomaly. The second assembly was torqued using the extension of the shank to determine when the preload reached 64 N·m. This second test assembly also failed at 1.2 x $10^6$ cycles at the 100% load level with failure initiated by fatigue at the root of the top thread of the prestressed shank, identical to the first piston rod.

Shot peening is being applied to the thread roots to provide a compressive stress and to increase the stress required to initiate fatigue cracking. Specimens are being prepared to test this concept during the next report period.

Proof Fatigue Test of a Mod II Cast Iron V-Block

A fully machined cast iron Mod II V-block was subjected to proof-fatigue testing to verify the design, material, and manufacturing processes. The material was ductile iron 80-55-06.

The cast iron V-block was completely machined; however it had some porosity located in an external cold connecting duct, which allowed helium to leak from the gas cycle to atmosphere. This was not acceptable since the V-block could not hold the engine working fluid, hydrogen. A repair using epoxy impregnation successfully sealed the leak and allowed the fatigue testing to proceed.

Prior to fatigue testing, the V-block was pressurized to just over 30 MPa (4350 psi) several times during the course of the initial pressure checks and impregnation trials, and no damage was imparted to the casting.

The duty cycle of the V-block required a special fatigue test setup to simulate the state of stresses at a thin (5 mm) section between the pressure-containing, cold-connecting ducts. The V-block was fixtured such that the cyclic pressure was applied to the four cycles, each 90° out of phase to the cycle ahead and behind it. This required a complex setup using the two interconnected hydraulic fatigue test machines shown in Figure 2-21.

The casting was subjected to the test cycle shown in Table 2-5. The test was successfully run through Step No. 4, the 137.5% load level. However, when attempting to run at the 150% load level, the O-rings in the fixturing leaked, allowing only 18,000 cycles to be completed. Several attempts to repair the tooling and restart the test were unsuccessful.

Based on the completion of step 4, and finding no failure in the cast iron V-block after 18,000 cycles at step No. 5, the test was considered successfully complete.

TABLE 2-5
MOD II V-BLOCK TEST CYCLE

| Step No. | No. of Cycles | % of Full Load |
|---|---|---|
| 1 | $10^7$ | 100.0 |
| 2 | $10^6$ | 112.5 |
| 3 | $10^6$ | 125.0 |
| 4 | $10^6$ | 137.5 |
| 5 | $10^6$ | 150.0 |

## Cold Engine System Development

The primary objective of cold engine system (CES) activity is to develop reliable, effective, long-life rod seals, piston rings, and static seals. Activity during the reporting period has been concerned with the evaluation of seals and piston rings in engines. The low-temperature operation of piston rings, rod seals, and static seals has been investigated in cold room tests. Seals and piston ring development will continue through Mod II engine tests.

### Main Seals

The Mod I and Mod II Pumping Leningrader (PL) seals are designed to have a parallel bore in the lower seal section and a tapered bore in the upper pumping ring section. During this reporting period newly manufactured seals were found to have bores that tapered continuously throughout their length. A review of the manufacturing process revealed no deviations from the prescribed procedure. However, in the same time frame, a "stress-relieving" heat-treatment cycle had been introduced for the raw material prior to machining with the objective of improving the dimensional consistency of the finished seals. A small batch of seals was produced from raw material that had not been stress relieved, and these seals had the required final bore geometry. This is a strong indication that the stress-relieving process had an adverse effect on the final forming process of the seal manufacture, resulting in the continuously tapered bores.

The Mod II PL seals are shorter than the Mod I seals. To evaluate the effect of the reduced length on performance, equivalent short seals were made to fit on Mod I-size piston rods and were installed in Mod I engines No. 3 and No. 6. As reported in the last semiannual report, the results showed that the life of the short seals was inferior to that of the BOM Mod I seals. A modified short seal design was developed to try to improve the life. Four seals of this modified design failed, due to oil leakage, after 103 hours of engine operation. The reasons for the reduced life have not yet been resolved; however, there is a strong suspicion that stress-relieving the raw seal material, referred to above, could have been a major contributor to the relatively poor performance of the short seals. Seal life is currently being evaluated in Mod II engine No. 1.

Piston Rings

A set of BOM split-solid piston rings completed 3575 hours of operation in Mod I engine No. 6. During this time there was no deterioration in engine performance, which indicated that the piston rings were sealing effectively and consistently.

Three different piston ring systems (Figure 2-22) were evaluated in back-to-back performance tests in Mod I engine No. 3.

- BOM split-solid piston rings
- Single solid piston rings
- Split-single (single ring with a lap joint).

The differences in power and efficiency with the three piston ring systems were generally small, but overall there was an indication that the single ring systems did improve performance (Figures 2-23 and 2-24). Also, there was an indication that the split-single ring gave better efficiency than the single solid ring. The improvement in performance with the split-single ring was most pronounced at low pressures and speeds, where it would be most advantageous for an automotive application.

Cold Temperature Tests

To achieve an acceptable recharge cycle, the rate of leakage of hydrogen from an engine system must be maintained at a low level under both static and operating conditions. In an automotive application, an engine typically runs for less than 10% of the total time, and therefore the amount of hydrogen that leaks out during static periods will probably be the predominant part of the total leakage. For a six-month recharge cycle the static leakage objective for the Mod II engine has been set at 3.9 mg/hr (0.047 nl/hr) for all ambient conditions, including temperatures to -29°C (-20°F).

The range of operating conditions, particularly temperatures, is very demanding on the static O-ring seals. This severely limits the choice of O-ring materials that can be used to seal the hydrogen. The current BOM O-rings in the Mod II engine are all Viton, a fluorocarbon material. Viton O-rings have performed well in Mod I engine testing but published data on the material's low-temperature properties suggest that it will be marginal at maintaining an effective seal at the low temperatures encountered in an automotive application.

In 1983, several O-ring materials were screened through cold room tests at USAB with temperatures down to -40°C. The main conclusions from these tests were

- EPDM (ethylene-propylene) O-rings gave the lowest rate of hydrogen leakage at low temperatures, but the published data for EPDM indicate that its high-temperature properties are inferior to Viton.

- Viton GLT O-rings had better low temperature properties than standard Viton and maintained an efficient seal down to -40°C.

- Silicone O-rings maintain their elastic properties at low temperatures, but they have a high diffusion rate with hydrogen that makes them impractical for sealing external leak paths. However, this does not preclude the use of silcone for the expander rings used in conjunction with the piston rings and capseals, where the main concern is gross leakage, not diffusion.

To establish the overall leak rates that could be expected with the different O-ring materials at low temperatures, the tests were carried out on a motored engine with the cooling system drained. The engine was pressurized with hydrogen to 2 MPa (normal shutdown condition) and stabilized at the required test temperature. The engine was then isolated from the gas supply, and the engine pressure monitored with respect to time. The total hydrogen leakage was determined from the change in engine pressure. Direct leakages into the crankcase and water jacket were measured by a water displacement method. The first measurements were made at the prevailing ambient temperature and then at successively lower temperatures until either the leak rate became excessive or the low temperature limit of the cold room (-40°C) was reached.

For the first test, the engine was assembled as a motored unit, including the power control system. In this build all the O-rings were EPDM, and the piston rings were BOM split-solid. The engine was motored to break in the piston rings and then moved into the cold room. The results of the cold leakage tests are shown in Figure 2-25. With the EPDM O-rings, it was possible to go down to -35°C with no excessive leakage, but at -40°C the total leakage was starting to increase rapidly and exceeded the goal. This suggests that -40°C probably represents the limit for good sealing with EPDM O-rings. Leakage into the crankcase was very small and not measurable below 0°C. At all temperatures there was no measurable leakage into the water jacket.

When the test was completed, the power control system was removed and the leakage measurements repeated. The total leakage is shown in Figure 2-26 together with the data from the previous test, including the control system. There is no distinct divergence between the two sets of data, and one can only conclude that the control system was not a dominant part of the total leakage.

The engine was torn down, and the O-rings at the top of the cylinder liners and top of the coolers were replaced with Viton GLT O-rings; all other O-rings were kept the same (EPDM). The split-solid piston ring system was changed to a single solid ring system. The quad-ring expanders for the piston rings and the O-rings for the capseals were also changed to silicone components. After reassembly with the power control system, the engine was motored to break in the piston rings and then transferred to the cold room. The cold leakage data are shown in Figure 2-27. At -30°C total leakage remained low but at -40°C there was a significant increase in leakage.

Overall the combination of EPDM and Viton GLT O-rings gave the lowest leakage, but even then the leakage was beginning to increase rapidly at -40°C. However, considering the range of operating conditions, this represents the best choice of available material combinations.

Using the same motored engine, cranking tests were carried out to compare the sealing properties of BOM split-solid piston rings, single solid piston rings, and split-single rings.

Prior to each cranking test, the new piston rings were broken in by motoring the engine at 5 MPa (1500 rpm) for 100 hours. The cranking tests were carried out using the engine's own starter motor with the battery located outside the cold room and with the engine pressurized to 2 MPa. For each temperature during the tests, the four cycle pressures, minimum and maximum pressures, and the cranking speed were recorded.

The first cranking test was carried out with the motored engine assembled with split-solid piston rings. The cranking speed dropped off rapidly with temperature, as shown in Figure 2-28. At -40°C, it was impossible to crank the engine, mainly due to the increased drag in the crankshaft/drive system. Figure 2-29 is a plot of the maximum ($P_{max}$) and minimum ($P_{min}$) pressures. In this context, $P_{max}$ is the largest value of the four maximum cycle pressures and $P_{min}$ is the smallest of the four minimum cycle pressures. The difference between $P_{max}$ and $P_{min}$ decreases with temperature, but since the cranking speed also decreases with temperature, these data represent the combined effects of temperature and cranking speed.

The next cranking test was carried out with a single solid piston ring system using the testing procedures previously described. The decrease in cranking speed with temperature was similar to that with split-solid rings. $P_{max}$ and $P_{min}$ are illustrated in Figure 2-30.

For the final test, split-single piston rings were installed. The measured $P_{max}$ and $P_{min}$ are given in Figure 2-31. Cranking speed again falls off with temperature.

The cycle pressures generated during cranking are a function of piston ring leakage and cranking speed. At a constant speed, any changes in the cycle pressures generated can be attributed to the operation of the piston ring system. Unfortunately, in the cold-cranking tests, the starter motor could not maintain a constant speed at all temperatures due to drag in the drive system, which made it more difficult to draw conclusions.

Comparing the differences between $P_{max}$ and $P_{min}$ at 20°C (Figures 2-29 and 2-30), it is clear that the difference is much larger with the single solid piston rings. Since the opposite was suggested, this implies that the cycle pressures were not uniform with the single solid rings, and there was evidence of this from the individual pressure traces. On the other hand, at 20°C, the pressure differences for the split-single rings (Figure 2-31) are similar to the split-solid rings. Another method of comparison is shown in Figure 2-32 where the average cycle pressure ratios for the three different ring systems are plotted as functions of temperature. This comparison, which ignores differences in individual cycle pressures, indicates that the split-solid piston rings generated a higher cycle pressure ratio at all temperatures and suggests that they provide a better seal as expected.

At the lower temperatures, the cranking speeds achieved would probably have been too low to allow an engine to start. To evaluate the quality of the seal provided by the different piston ring systems, it is necessary to carry out tests in which the cranking speed at any temperature is constant. Such tests will be conducted during the next report period.

### Drive System Development

The major test objective of drive system development was to demonstrate 10 hours of endurance at maximum speed and pressure to verify the integrity of the Mod II engine bottom end, specifically the Koyo rolling element bearings. A secondary objective was to test Mod II single solid piston rings for their tendency to contribute to cycle-to-cycle pressure imbalances.

All hardware was received, but the machined cast-iron V-blocks were found to have porosity present that could result in overboard leakage of gas used in the working cycle. This would mean repeated addition of gas if the leak were moderate, or possibly no operation for large leakage. Sealing of the porosity with a polyester resin was attempted since it is common practice in industry for such problems.

High-pressure resin impregnation of the cast V-block was successful in eliminating helium leakage up to 2000 psi. Buildup of the engine was started. Some problems with interference of the Koyo bearing crankshaft with the crankcase and connected rods were discovered and corrected. A problem with alignment of the outer races of the roller bearings was corrected by installing a snap ring retainer on the races. The single solid piston rods have been received. Lack of coolers has prevented operation to date, however, testing will be completed in the next report period.

## Control System/Auxiliaries Development

The major goals of this task are the development of the engine control and auxiliaries systems. Specific systems goals include the development of a highly flexible digital air/fuel control (DAFC) with low combustion air pressure drop and low idle fuel consumption, a simplified mean pressure control (MPC) that does not require a servo-oil actuator, a high-efficiency combustion air blower and electric drive system, and logic to operate the various control systems.

Throughout the semiannual report period, primary emphasis was placed on acquiring, assembling, testing, and debugging the various pieces of Mod II control systems hardware.

Digital Engine Control (DEC) software modification efforts required for incorporation of Mod II controls continued during this period. Results to date have been highly successful. The Spirit vehicle DEC was incorporated and operated with full Mod II SES capabilities; the Mod II test cell full SES software was completed and awaits installation. Final software modifications for the 3-volume compressor logic and MPC are anticipated, pending further testing in the Spirit and test cell environments. Testing to date indicates that, while operation of the 3-volume compressor switching logic has been adequate for the conditions tested, additional changes will be required for full realization of anticipated benefits, with acceptable vehicle drivability. Prototypes of the Mod II blower drive and battery charge systems began bench evaluation.

### Digital Engine Control

All remaining DEC printed circuit boards were populated for use as spares.

Software was written that will increase the fuel metering resolution by a factor of 10 at low fuel flows. An associated electronic modification was designed and installed in the DEC. The system was bench tested and is currently operating in the ASE test cell and the Spirit.

Significant progress was made in upgraded DEC design during this report period. Hardware specifications were published and hardware selection based on these specifications completed. The Computer Processing Unit (CPU) will be based on the Intel 8086 (CMOS logic) family, and an industry-accepted standard bus structure (STD bus) will be used. Hardware selection and design criteria are focused largely on meeting Mod II environmental and vehicle constraints, including extended temperature operation (-40°C to +85°C), low power operation, single battery operation, and improved noise immunity. Preliminary software specifications were published. The software architecture adopted is based on a multitasking environment aimed at providing highly maintainable and reliable software.

In support of these goals, the industry-accepted programming language "C" was selected for software coding. Among its positive attributes, C-based software is highly readable, and therefore maintainable, and highly portable among other computers (eliminating the coupling of software life to hardware life). Complete software standards for the upgraded DEC were published. The standards rigidly define software documentation practices and software configuration control procedures for both software developers and field personnel. The standards were designed to improve the quality of documentation and reduce the overall manpower requirements for DEC software and hardware support efforts.

### Mean Pressure Control

The Mod II power control valve (PCV) and electric actuator completed 1000 hours of endurance testing. All components of the valve and actuator except the electric motor were in excellent condition. One motor failure occurred after 450 hours due to worn-out brushes; the second motor survived more than 550 hours. The motors should survive 3000 to 4000 hours in an automobile as the endurance cycle test is more severe. The printed circuit board version of the electric actuator drive electronics was completed and bench tested for 100 hours. It was then installed in the Spirit along with the Mod II PCV and actuators. Performance of the PCV in the Spirit has been excellent.

The Mod II PCV and the upgraded Mod I PCV were flow tested and characterized (see Figures 2-33 and 2-34). The opening patterns of the two valves are significantly different by design. After installation in the Spirit, the Mod II PCV was found to have undesirable nonlinearities. The design was modified and hardware for engine testing was ordered.

The Mod II engine check valves successfully completed 930 hours of endurance testing on engine No. 6 at USAB. Two check valves leaked at 350 hours due to contamination.

The Mod II 3-volume hydrogen-compressor piston rods were repaired and testing of the compressor resumed. One unit is on rig test at USAB. It will be characterized for round-pumping and pumping powers. It will then be endurance tested for 500 hours. A second compressor was installed in the Spirit for further engine and vehicle testing.

The entire Mod II MPC system was installed in the Spirit. This included the Mod II PCV and electric actuator, the two-tank system, and the 3-volume compressor. Development of the control logic for the system, in particular the 3-volume compressor, is now in progress. The system is fully functional though not optimized. Response of the system under all operating conditions is very good; drivability is good, but increases with reduced idle pressures. The Mod II idle conditions of 2 MPa mean pressure and 400 rpm idle speed have been demonstrated. Extensive testing of the system will continue into 1987 in the Spirit and in the Mod II engine and vehicle.

Mod II Electronics Development

Three separate electronic control packages to control the electric PCV, the pulse width modulated (PWM) fuel valves, and the 3-volume compressor were completed. Ten copies of all three packages, which included printed circuit boards, connectors, packaging, and a wiring harness, were assembled.

### Auxiliary Development

Second prototypes of both the blower drive and the battery charge regulator were built. These were installed in a dedicated test cell (see Figure 2-35) to permit independent development of the electronics. Shaft-to-shaft performance and endurance testing will take place using this setup. Initial performance testing of both the battery charge and the blower drive systems caused failures of components, so it was decided to improve the circuits until reliable operation over the required conditions could be achieved before attempting shaft-to-shaft testing.

Spirit testing with the first prototype sets exhibited repeated failures of the battery charge system but not the blower drive circuit. The first blower drive prototype has accumulated approximately 50 hours without failure.

Several modifications were made to both sets of prototype battery charge regulator hardware. These included improved component cooling, more powerful switching transistor drive, and larger, higher-rated components. These modifications appeared to improve the survivability of the system, but failures still occurred during testing. Development will continue in the next report period.

An analysis of the Mod II Celebrity battery charge system characteristics was conducted. This analysis indicated that the Mod II battery charge system would exhibit a net battery discharge at 400-rpm idle conditions and over the constant volume sample (CVS) urban driving cycle, with a net battery charge over the CVS highway driving cycle. The net discharge conditions are of minor concern due to the small magnitude of the net discharge (analysis indicates 96 consecutive CVS urban cycles can be run prior to battery discharge to "inoperable" levels). The study concluded that a second, smaller battery should be used to power the DEC, due to the DEC's failure to operate at battery voltages of less than 10.5 V. During high-load current conditions, i.e., with the engine starter operating, battery voltage levels well below the DEC operating limit are possible.

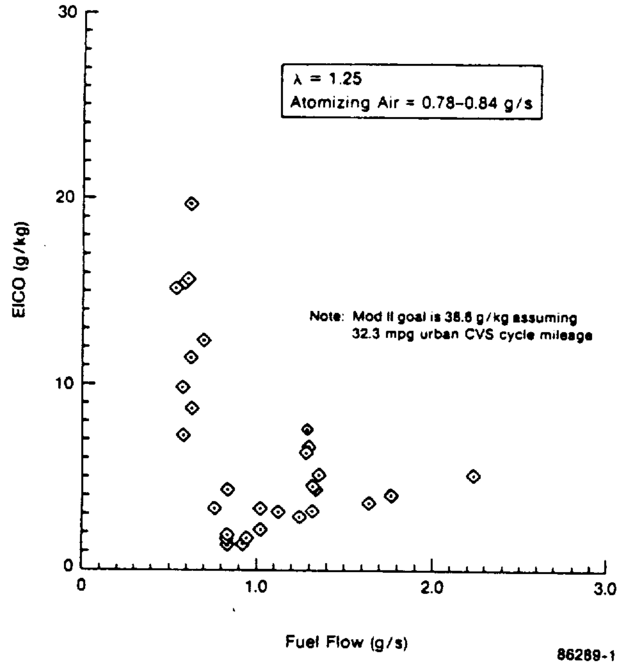


**Fig. 2-1 Mod II Engine No. 1 CO Emissions at 720°C Set Temperature**

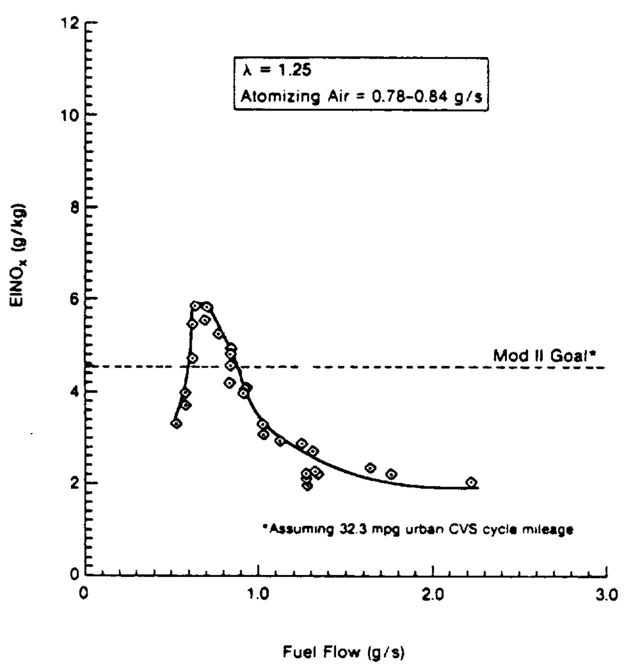


**Fig. 2-2 Mod II Engine No. 1 $NO_x$ Emissions at 720°C Set Temperature**

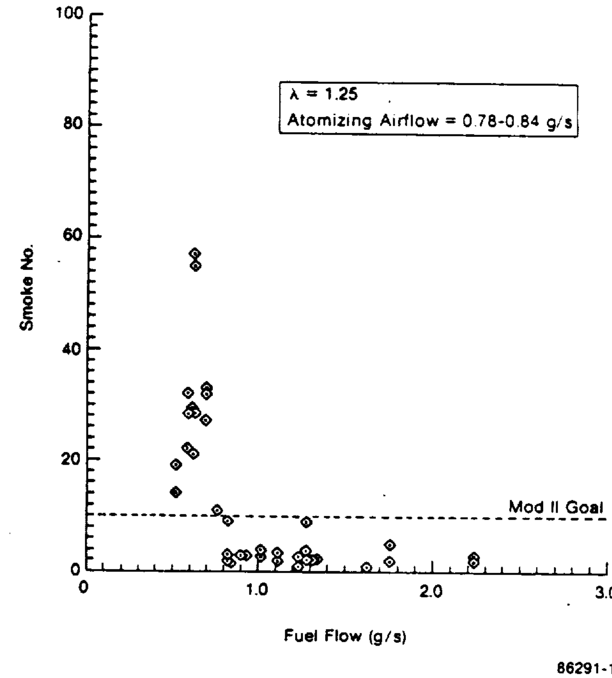


**Fig. 2-3 Mod II Engine No. 1 Soot Emissions at 720°C Set Temperature**

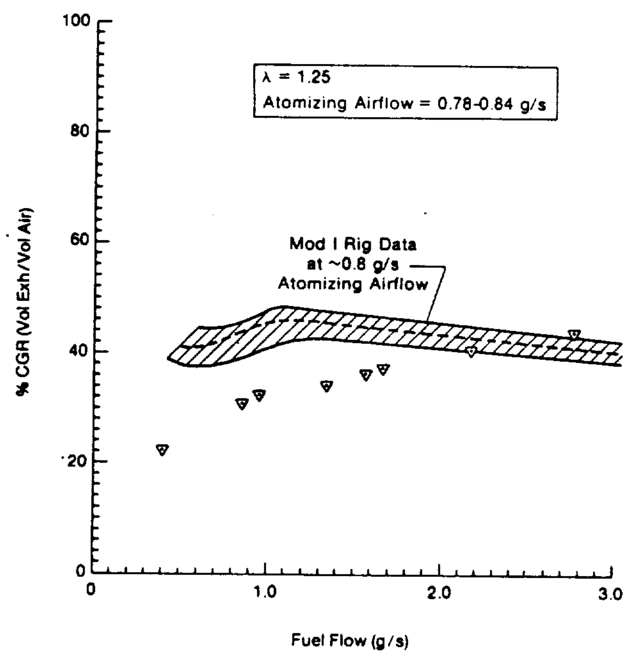


**Fig. 2-4 Mod II Engine No. 1 Combustor CGR Level at 720°C Set Temperature**

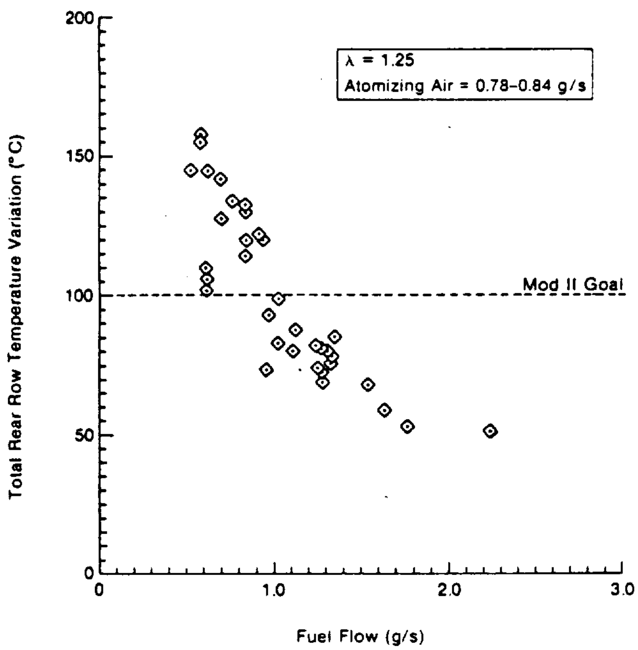


**Fig. 2-5 Mod II Engine No. 1 Heater Head Rear Row Temperature Variation at 720°C Set Temperature**

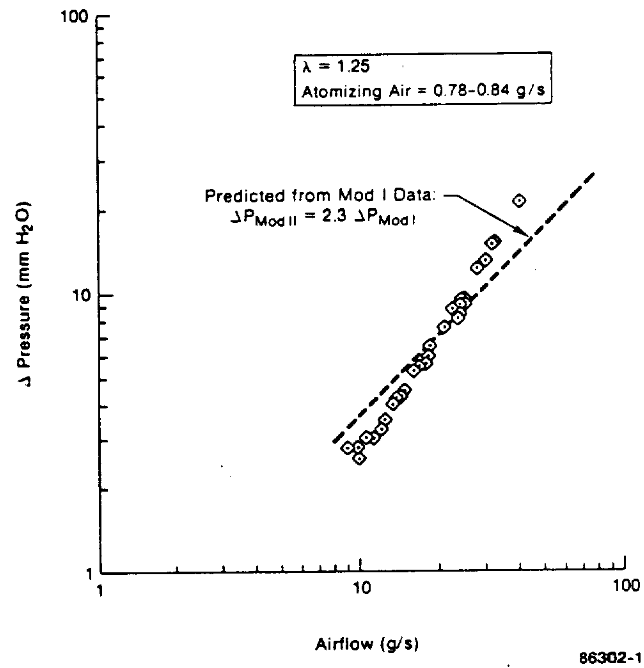


**Fig. 2-7 Mod II Engine No. 1 Heater Head Pressure Drop (Combustion Gas Side) at 720°C Set Temperature**

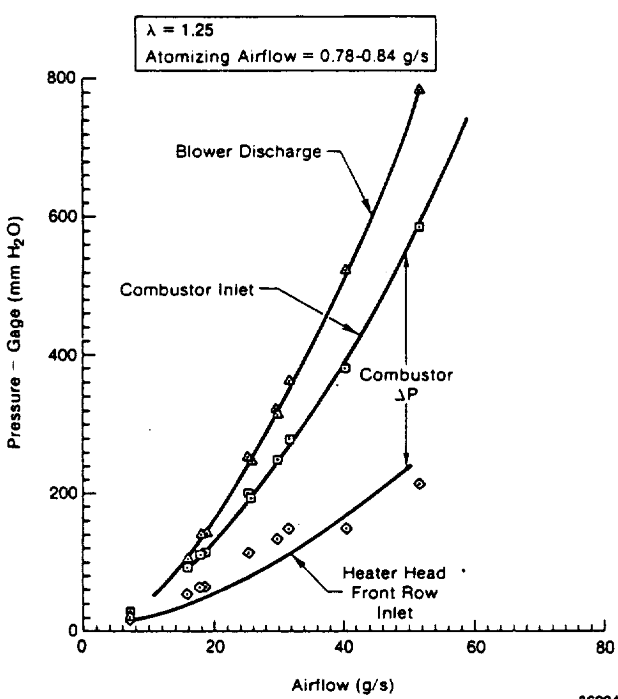


**Fig. 2-6 Mod II Engine No. 1 EHS Pressure Drops at 720°C Set Temperature**

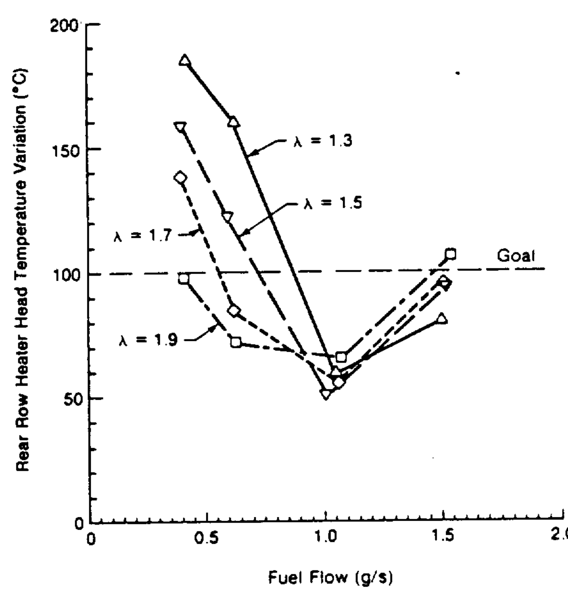


**Fig. 2-8 Effect of Lambda on Mod II Engine Temperature Variation**

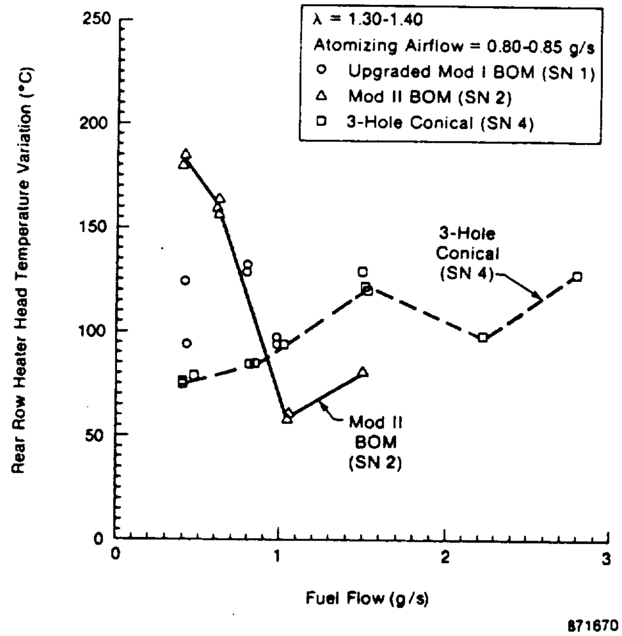


Fig. 2-9 Effect of Fuel Nozzle Type on Mod II Engine Heater Head Temperature Variation

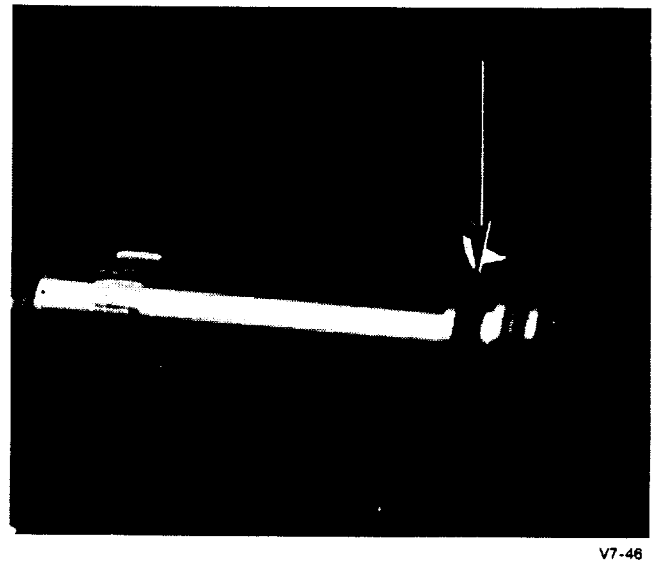


Fig. 2-11 Conical Nozzle Showing Extruded O-Ring after Cold Flow Test

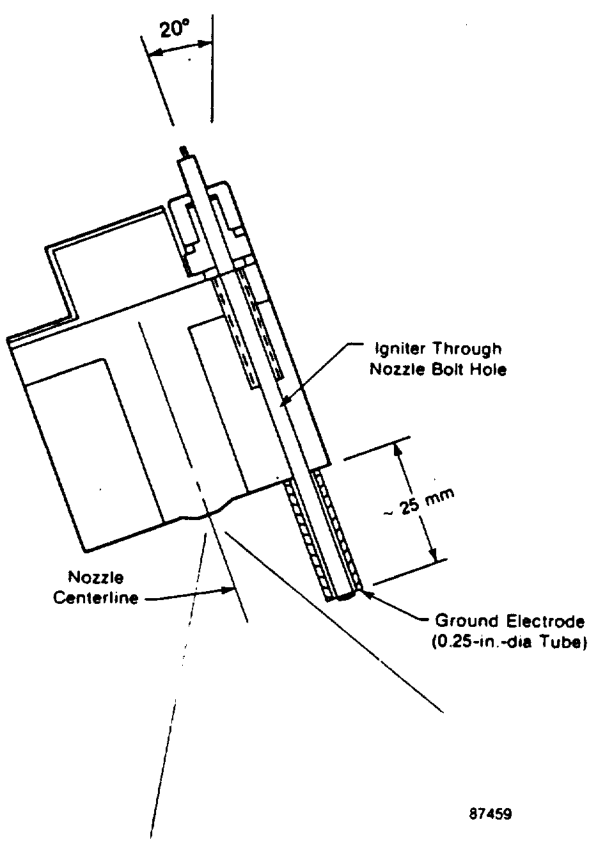


Fig. 2-10 Igniter Location for 3-Hole Conical Nozzle In Mod II CGR Combustor

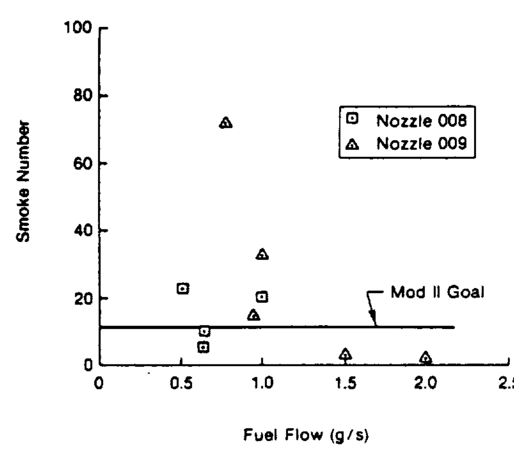


Fig. 2-12 Center Igniter Conical Nozzle Rig Soot Emissions



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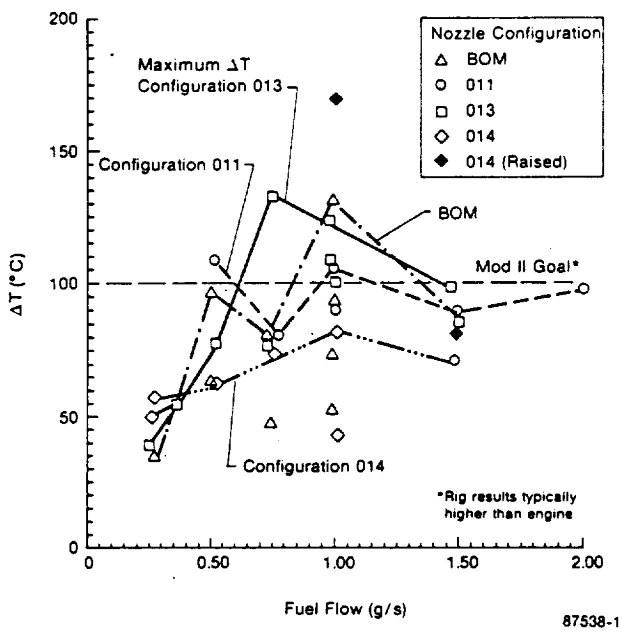


**Fig. 2-13 Center Igniter Conical Nozzle Performance Rig Rear Row Heater Head Temperature Variation**

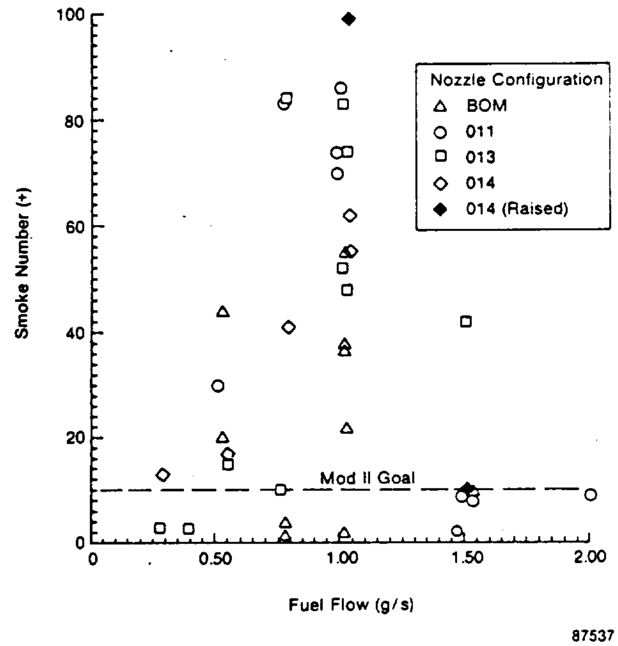


**Fig. 2-15 Center Igniter Conical Nozzle Performance Rig Soot Emissions**

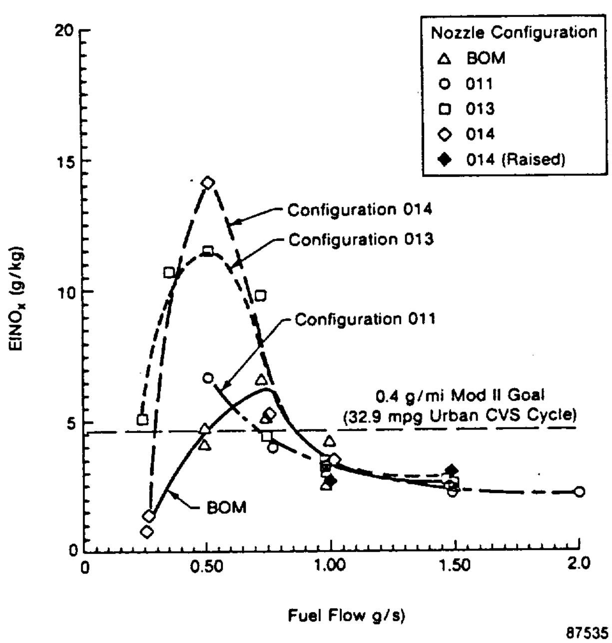


**Fig. 2-14 Center Igniter Conical Nozzle Performance Rig $NO_x$ Emissions**

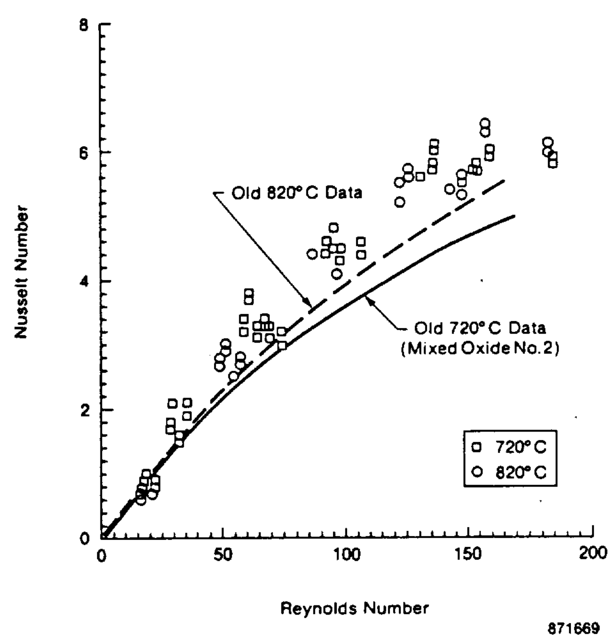


**Fig. 2-16 Ceramic Preheater Heat Transfer Test Results**

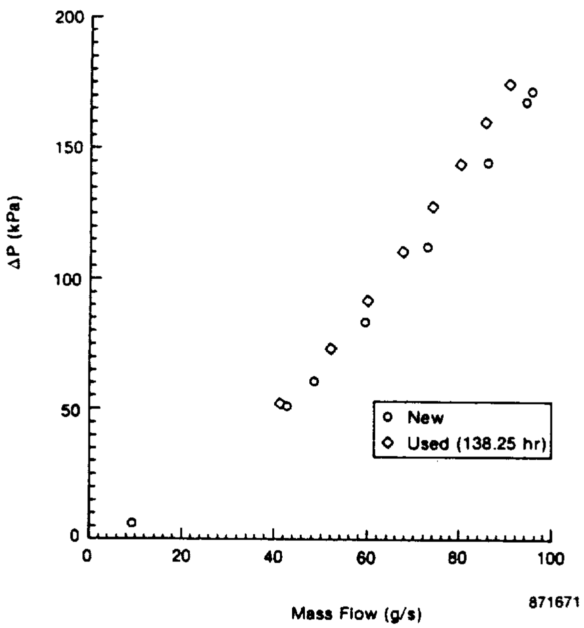


Fig. 2-17 Mod II Regenerator Pressure Drop (Serial Number 3)

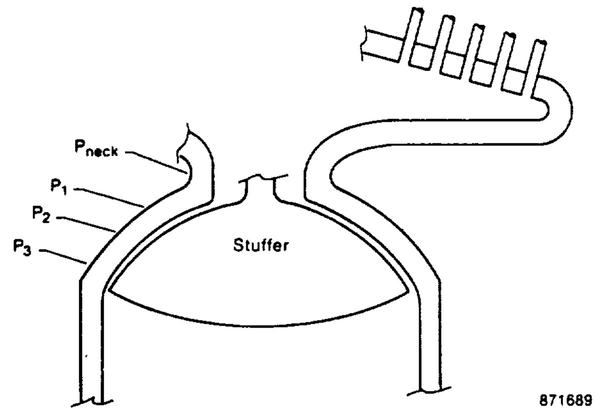


Fig. 2-19 Manifolded Heater Head Pressure Measurements

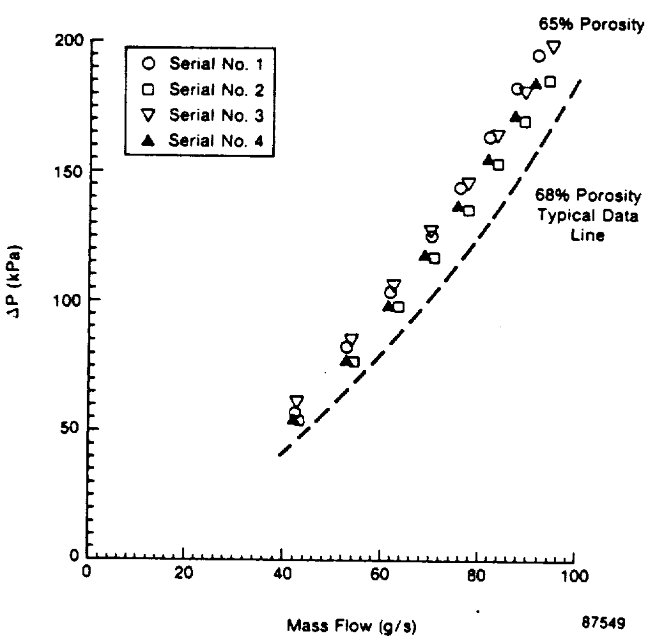


Fig. 2-18 Mod II Alternate Porosity Regenerator Pressure Drop

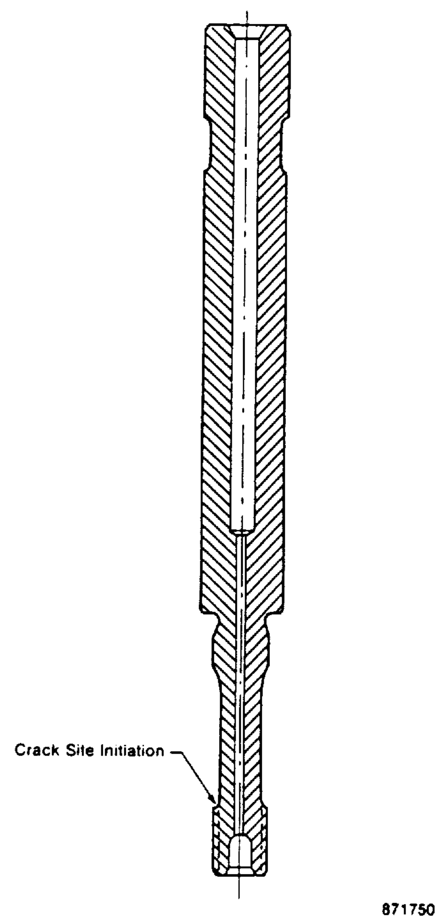


Fig. 2-20 Single Solid Piston Ring Rod Cross Section

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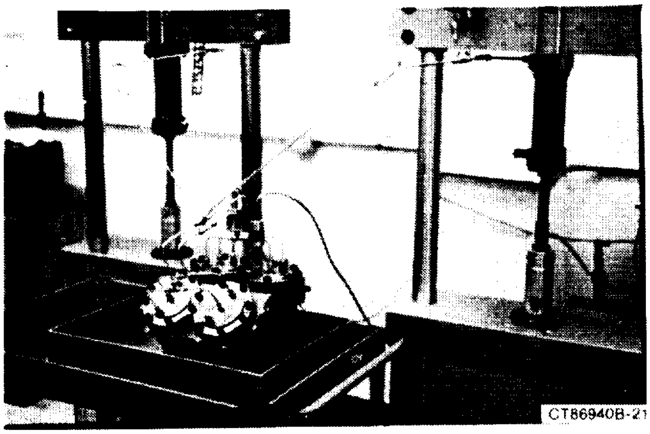


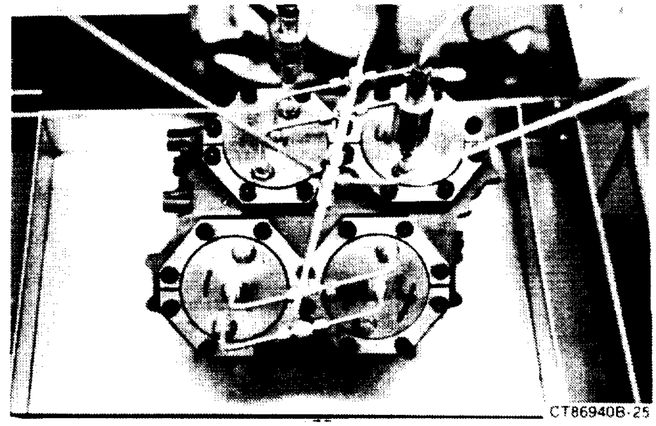
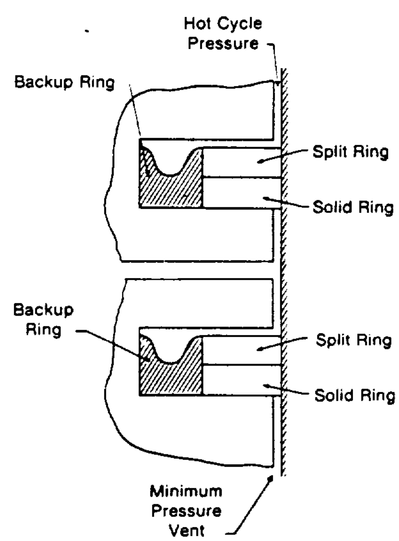
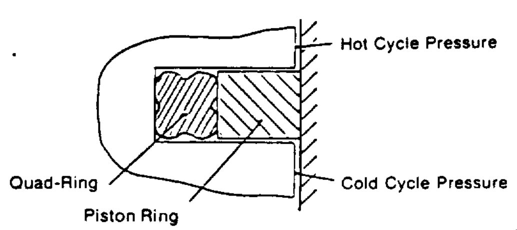




**Fig. 2-21 Interconnected Hydraulic Fatigue Test Machines**




**a) BOM Split Solid Piston Rings**




**b) Single Solid and Single with Lap Joint Piston Rings**

**Fig. 2-22 Piston Ring Systems**

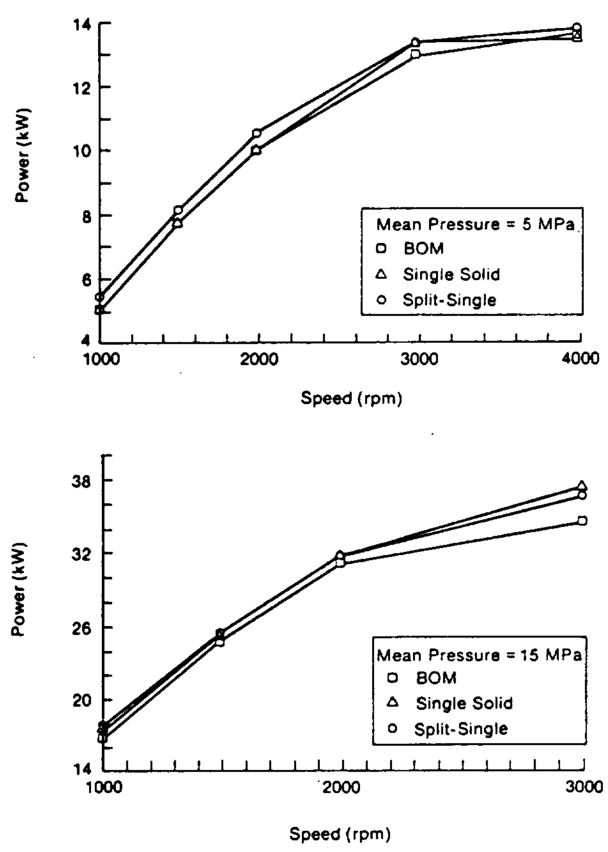



**Fig. 2-23 Mod I No. 3 Piston Ring Performance Tests**

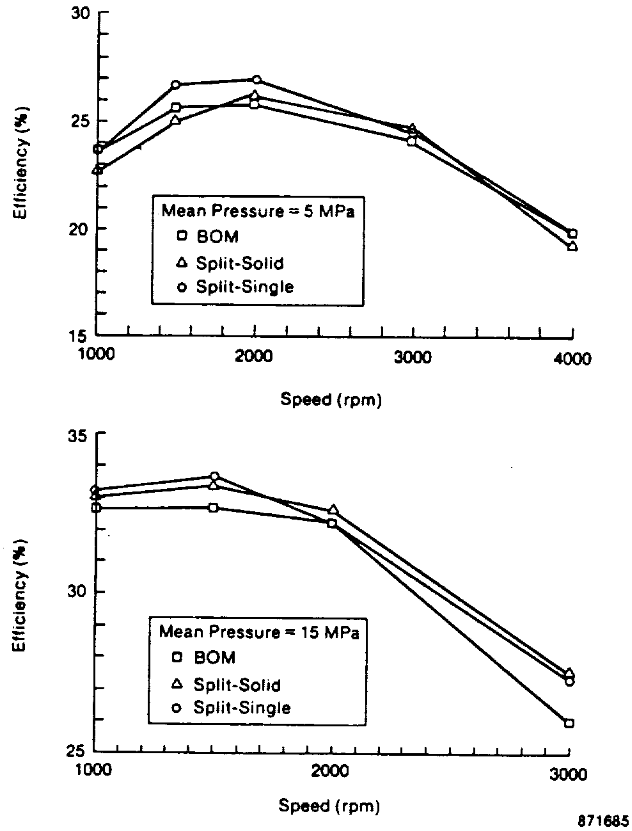



**Fig. 2-24 Mod I No. 3 Piston Ring Performance Tests (continued)**

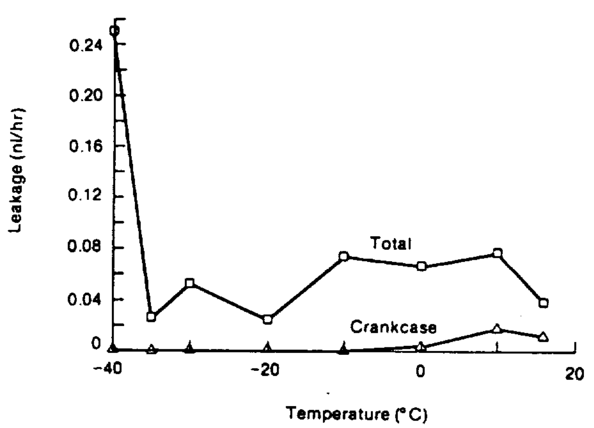



**Fig. 2-25 Cold Leakage Rates with EPDM O-Rings and Control System**

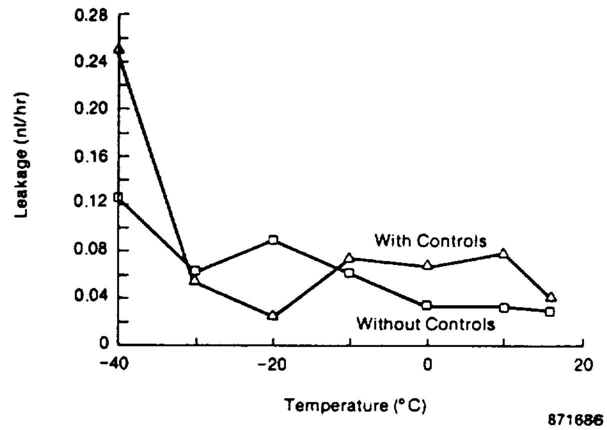



**Fig. 2-26 Cold Leakage Rates with EPDM O-Rings**

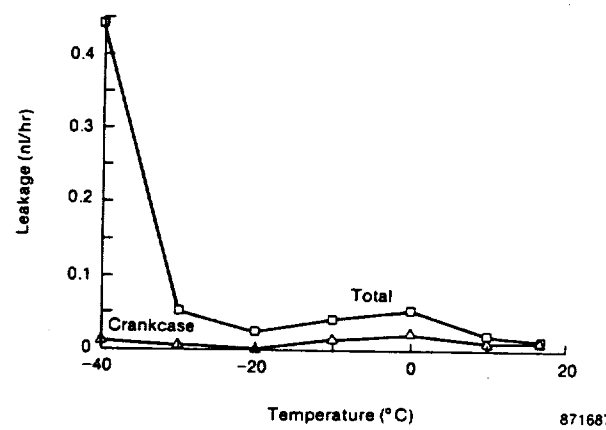



**Fig. 2-27 Cold Leakage Rates with Viton GLT/EPDM O-Rings**

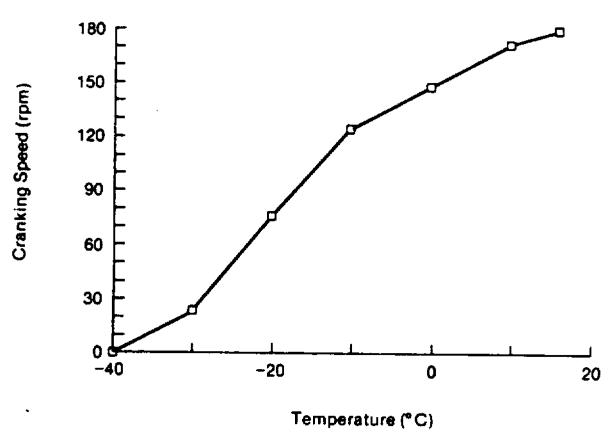



**Fig. 2-28 Cranking Speed with BOM Split-Solid Piston Rings**

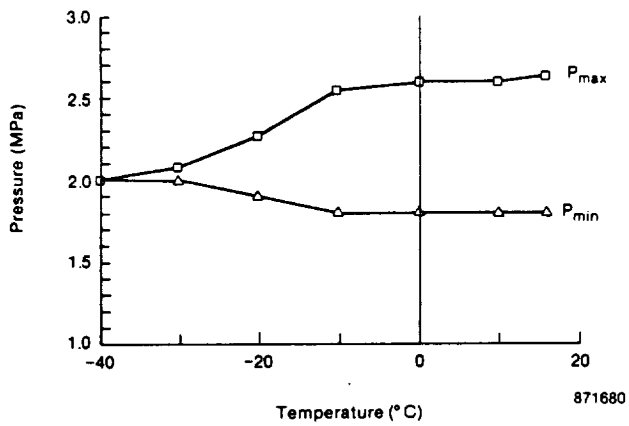



**Fig. 2-29 Maximum/Minimum Cycle Pressure with BOM, Split-Solid Piston Rings**

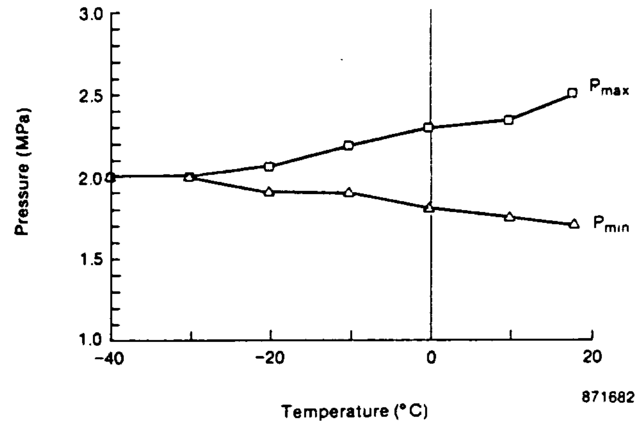



**Fig. 2-31 Maximum/Minimum Cycle Pressure with Split-Single Piston Rings**

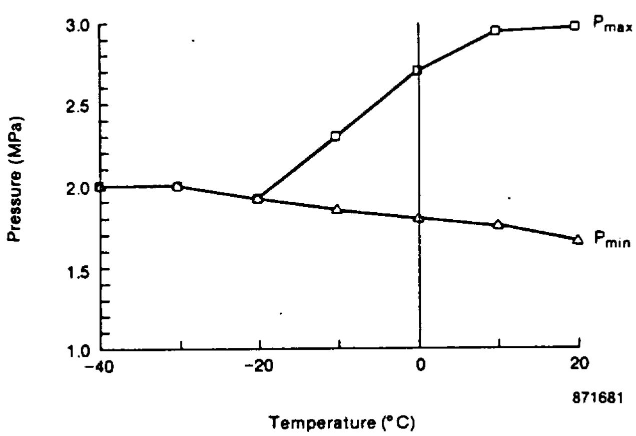



**Fig. 2-30 Maximum/Minimum Cycle Pressure with Single Solid Piston Rings**

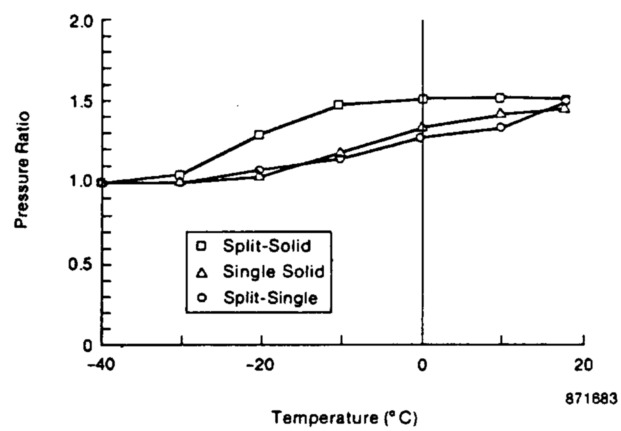



**Fig. 2-32 Pressure Ratios with Various Piston Rings**

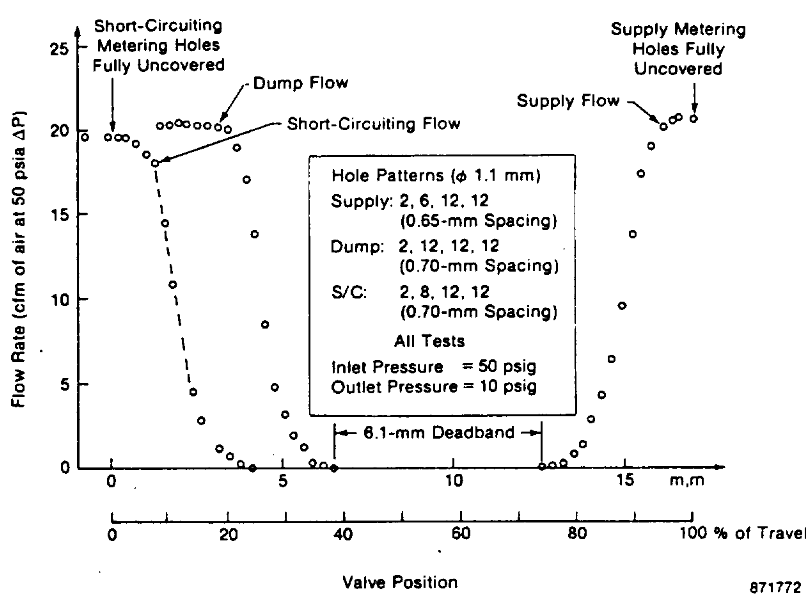



**Fig. 2-33 Mod II PCV Flow Characteristics**

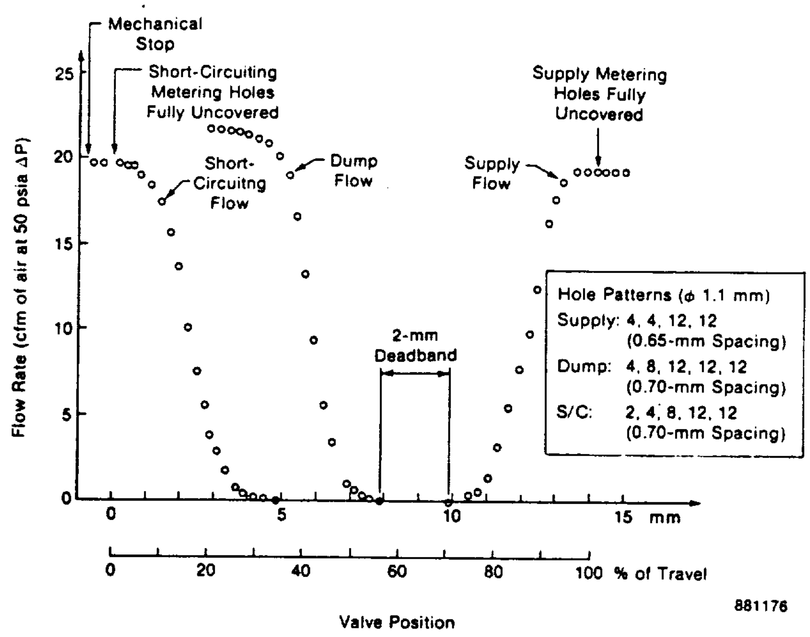



Fig. 2-34 Upgraded Mod I PCV Flow Characteristics

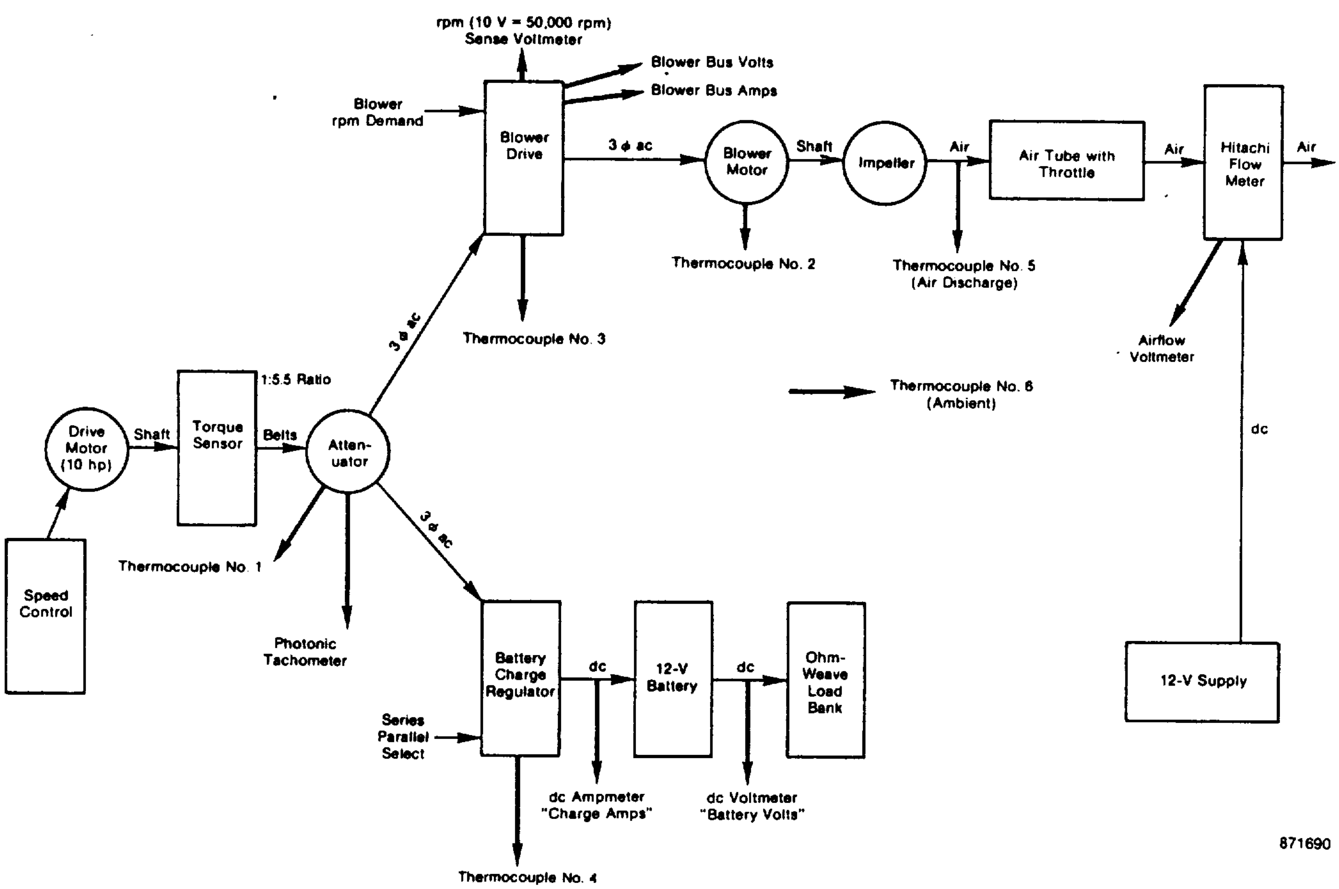



Fig. 2-35 Mod II Blower Drive/Battery Charge System Test Setup

## Mod I Engine Test Program

During the second half of 1986 a total of six Mod I engines were operational at one time or another. Two of the engines are in their original Mod I configuration and are located at USAB in Sweden. The remaining engines are upgraded Mod I engines of which three are located in the United States at MTI and one at USAB. One of the upgraded Mod I engines at MTI was loaned to the Army Generator Set program. The specific purpose of each engine is listed below:

- Mod I Engine No. 3 (USAB): Component Development/Storage
- Mod I Engine No. 7 (USAB): Gas Leakage Evaluations - Cold Ambients
- Upgraded Mod I Engine No. 5 (Langley AFB): NASA TU Program
- Upgraded Mod I Engine No. 6 (USAB): 820°C Endurance Test/Alternative Programs
- Upgraded Mod I Engine No. 8 (MTI): Spirit Vehicle
- Upgraded Mod I Engine No. 9 (MTI): Generator Set/Storage.

Ffollowing is an itemized account of the activities of the program engines. During the period an additional 847 hours were accumulated, bringing the total engine hours to 16,588 (Figure 3-1).*

### Mod I Engine No. 3

Mod I engine No. 3 accumulated a total of 29 hours during the second half of 1986, bringing its total hours to 2376. During the period its prime assignment was seals development. It was also used for development of the CGR combustor, fuel nozzles, and atomizing air systems. The specific results of this engine's test programs are included under "Component Development" in Section II of this report. In August 1986, following the completion of seals performance testing at USAB, the engine was removed from test and torn down. The drive system components were all in good condition and were reused with the exception of the crankshaft bearings, which were replaced. The piston rods had excessive wear in the capseal area and were replaced. The piston pins and guide rings were also worn out and were replaced. Future plans include conversion to an upgraded Mod I for alternative program use.

### Upgraded Mod I Engine No. 5

Engine No. 5 accumulated a total of 370 hours of operation during the second half of 1986, bringing its total operational hours to 1995. During this period the engine was operational in an Air Force step van at Langley Air Force Base, Virginia. Details of its operation are reported separately in this report (Section V).

### Upgraded Mod I Engine No. 6

Engine No. 6 accumulated a total of 246 hours during the second half of 1986, bringing its total operating time to 4060 hours. During the period, the engine's endurance testing at 820°C rear row tube temperatures was terminated. The engine was subsequently torn down and inspected. The inspection revealed that the crankshafts were in good condition except for the hydrogen compressor crankpin, which was cracked. Because the crankshafts were special with reduced-size journals, it was decided that they would be

*Figures are at the end of this section, beginning on page 3-4 replaced with standard BOM items. This necessitated replacing all the bearings and connecting rods, even though they were all in good condition and could have been reused. The wrist pins were excessively worn and had to be replaced. The piston rods evidenced excessive wear in the capseal area and were also replaced. Mainseal area wear was not excessive.

The water jacket was leaky and was replaced. The piston rings were worn out. The piston bases, domes, crosshead guides, and preheater were all satisfactory and were reused. The cylinder liners were worn out and replaced. The gas coolers, regenerators, flamestone, and external heat system were acceptable but were replaced with new, later-design items. The heater heads evidenced some corrosion, but no warping of the tubes. The thermocouple tubes were corroded. Two of the heads had over 3000 hours. Although not in ideal condition, the heads were usable.

Following inspection and rebuild, the engine was performance evaluated as a BSE and delivered to MTI.

Mod I Engine No. 7

During the second half of 1986, Engine No. 7 was used for cold cell gas tightness investigations. No running hours were accumulated. Test results are included in the Cold Engine System, Section II.

Upgraded Mod I Engine No. 8

Engine No. 8 at MTI is located in the Spirit vehicle. During the second half of 1986, the engine accumulated a total of 18 hours, bringing its total operational hours to 1217. During this period the engine was assigned to component development for transient testing of prototype Mod II hardware. The following were evaluated:

- DAFC
- Electrically actuated PCV
- CGR combustors
- Atomizing air systems
- Potential Mod II conical fuel nozzles
- Electric blower
- High voltage alternator
- Two-tank system.

Testing will continue into 1987, until the Mod II Celebrity vehicle becomes active.

Upgraded Mod I Engine No. 9

Engine No. 9 was on loan from the ASE program for the Army's use. It was installed in a generator set with a 25-kW rating at Ft. Belvoir. The engine was performance evaluated and endurance tested for 500 hours before its return. While at Ft. Belvoir the engine was operated with a leak in a oil cooling loop-line. The leak went undetected until the operator noticed no oil pressure during a routine check; the engine was shutdown and the line repaired. Subsequent runs resulted in low oil pressure. The engine was returned to MTI for rebuild with new bearings and crankshaft. After the rebuild, engine break-in was initiated. During a routine check, with the engine not running, the main working gas supply line failed at the storage bottle. The escaping high-pressure hydrogen ignited, causing some damage to the generator enclosure but none to the engine itself. Due to funding limitations, the damage was not repaired, and the program was terminated. The engine was placed in storage with a total of 393 hours.

Subsequent investigation revealed that the failure occurred in a braze joint at a point where a stainless steel tube is connected to a stainless steel block. The failure occurred in the joint itself, not the tube or block. It was caused by insufficient penetration of the braze material into the joint.

To prevent a recurrence of this type failure, safety requirements were reviewed, and the following procedures were initiated for all high pressure tube fittings (brazed, welded, and swaged):

- Components are 100% inspected for mechanical and physical damage. Braze joints are inspected for wetting of interior and exterior surfaces.

- Each line length, with fittings, is hydrostatically tested to 1-1/2 times maximum operating pressure for five minutes. All hydrogen-containing assemblies are tested to 4500 psi.

- Assemblies that pass are individually, serially tagged. Inspection and test results are recorded on the tag and in a log book.

The test and inspection are conducted by a qualified person other than the one who constructed the fitting.

### Mod I Analysis

Tests were conducted to evaluate the performance differences between the BOM upgraded Mod I split-solid piston rings and a single solid piston ring. The BOM configuration utilizes two sets of piston rings, a split ring and a solid ring, in each of two ring grooves, with the space between the grooves vented to cycle minimum pressure. The single solid ring replaces the two sets with one piston ring and therefore has no venting scheme. Losses caused by piston rings are associated with the combined effects of leakage past the rings and friction of the ring on the cylinder wall.

Analysis of the data indicates a net benefit for the single solid ring (Figures 3-2 and 3-3). Figure 3-4 shows the difference in net rejection to the coolant for the split-solid rings. This can be due to either increased leakage or increased friction.

The differences in these parameters confirm that the single-solid ring provides better performance, with up to 1-kW improvement in power. The combination of improved power and less heat input required would result in a 1-2% efficiency improvement for the single solid ring. Additional information was discussed in Section II.

### Mod I Design

Mod I design installation layout drawings were done to examine the feasibility of installing the upgraded Mod I engine in the Chevrolet S-10 pickup truck and the AMC Jeep Cherokee. It was found that the upgraded Mod I engine height from output-shaft centerline to the top of the engine exceeded the available height from transmission input shaft centerline to the hood on both vehicles, and interference with the front cross member would also occur on the S-10. This work was done in support of the Phase II NASA TU vehicle selection (Section V).

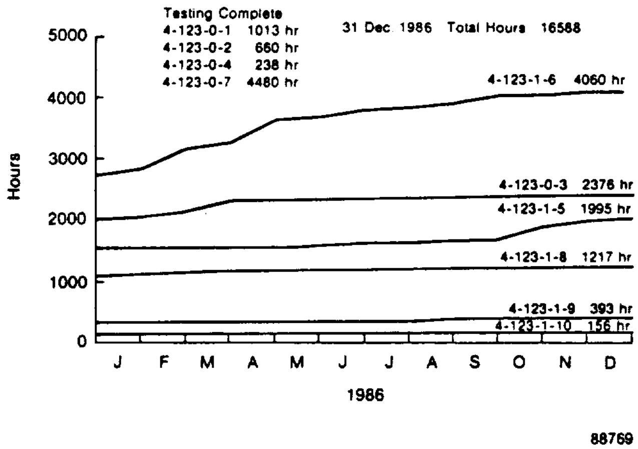



**Fig. 3-1 Mod I Engine Test Hours**

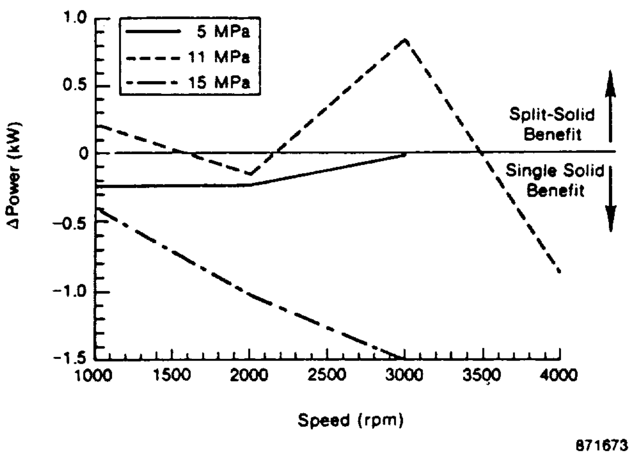



**Fig. 3-2 Piston Ring Net Power Difference (Split-Solid Minus Single Solid)**

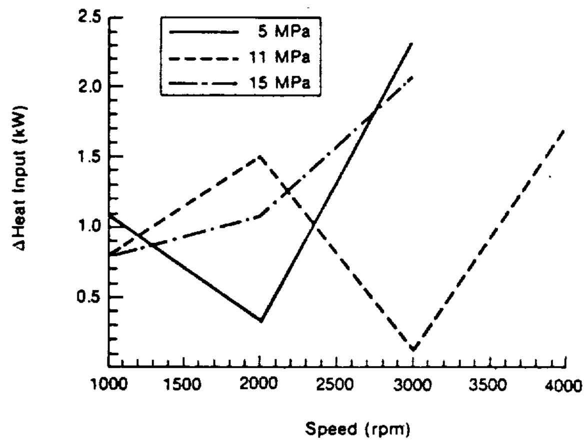



**Fig. 3-3 Piston Ring Heat Input Difference (Split-Solid Minus Single Solid)**

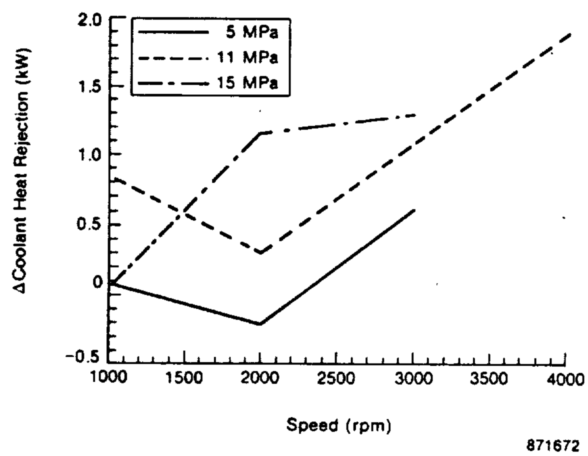



**Fig. 3-4 Piston Ring Heat Rejection Difference (Split-Solid Minus Single Solid)**

# IV. MOD II ENGINE DEVELOPMENT

## Introduction

During the last half of 1986, the engine accumulated a total of 203 hours, bringing its total operating hours to 223.0. During this period, the engine experienced several hardware problems. Each of these was solved and evaluation tests were begun.

## Mod II Engine Test Program

Operation of the engine at pressures above 10 MPa was delayed by lack of proper heater head castings. Testing continued with the so-called "low-pressure" heater heads rather than delay the program. Good quality, high-pressure-capable castings were received and a complete heater head was available in October. All testing was conducted with the interim design, configuration 1, manifolded heads pending completion of the configuration 4, no-manifold-type heads.

### Mechanical Development

During testing, three problems hindered the rapid accumulation of engine operation data: excessive cooler wear, high external heat system temperature gradients, and crosshead liner failures. Each of these problems was solved and performance testing was begun.

Coolers - During engine operation, high cycle-to-cycle pressure imbalances were observed. Inspection revealed excessive piston ring and cooler wear, and a powdery debris that had accumulated in the cycles as a result of the observed wear. The problem was traced to poor wear characteristics of the Rulon LD piston rings on the chrome-plated cooler bores. This was verified by both rig data and vendor consultations. Substitution of ion-nitrided stainless-steel cooler bores for the chrome-plated surfaces was successful in eliminating this problem.

Heater Head Temperature Differentials - High heater tube $\Delta T$ was experienced. This was found to be the result of both internal working gas and external combustion gas flow pattern anomalies.

The temperature of tubes on a single quadrant typically varies 40 to 60°C at a given load point. Generally, the tube at the end of the long-armed manifold was the hottest, while the tube nearest the neck of the manifold was the coolest. This spread compares to 20 to 40°C in an upgraded Mod I heater head. The arrangement of the Mod II is asymmetric (i.e., the short-arm manifold is next to the short-arm manifold on the adjacent head, and the long-arm manifolds are next to long-arm manifolds) due to the staggered V-cylinder configuration. Comparison of temperatures on "similar" tubes of adjacent heads is open to interpretation, since truly similar tubes do not exist except on opposite (not adjacent) heads. It is expected that improved $\Delta T$ will be obtained with the configuration 4, no-manifold heater heads.

It was also noted that the highest differentials occur at low-fuel flows, below 1 g/s, and that the temperatures of quadrants 2 and 3 would fall at these low flows, while those of quadrants 1 and 4 would not. The highest differentials were over 160°C.

Two possible explanations for this differential increasing at low fuel flows were buoyancy and nozzle fuel distribution. At these low fuel flows, combustion gas flow is low enough that natural and forced convection are of the same order of magnitude. Therefore, heater quardants 2 and 4, which are physically higher due to the 20° tilt of the engine, experience natural convection forces that oppose forced convection, resulting in stagnation of flow, reduced heat transfer, and reduced temperatures. In quadrants 1 and 4, which are lower, natural and forced convection forces are additive, causing a rise in temperatures.

In order to eliminate the orientation of the external heat system as a possible contributor to the $\Delta T$, tests were conducted with the entire EHS rotated 180° from its original position. Results indicated that even with lower fuel flows, the EHS does not influence $\Delta T$.

Tests were conducted to investigate possible buoyancy and nozzle distribution effects due to the tilt of the engine. The results of the tests indicated that both factors exert some influence on $\Delta T$. A detailed discussion of these is contained in the "EHS Development" and "Mod II Analysis" sections.

A backup exhaust gas recirculation (EGR) combustor will be procured if heater head $\Delta T$ becomes excessive.

Crosshead Liner Failures - Twice during the report period, a crosshead seized in its liner. Materials analysis indicated that type-A cast iron, rather than type-D, as used, would provide superior lubricity. In addition, the analysis concluded that clearances in the crosshead/liner were reduced due to thermal expansion when the engine was running; therefore, the nominal clearance was increased by 10 microns to bring running clearances to the proper 0.40 to 0.60 mm, as used in the initial upgraded Mod I. The material and clearance changes have resulted in trouble-free operation.

## Mod II Design

Design activity was about equally divided between revisions to existing engine drawings and continued development of the new configuration 4 (C4) heater head design. Revisions or engineering change notices (ECN) were executed to correct dimensional errors revealed during manufacture or assembly, to ease difficulty or cost of manufacture, or to correct design aspects proven deficient by engine or component tests. A total of 82 ECNs were processed, affecting 120 drawings.

### Combustor Designs

An alternative configuration for the fuel nozzle/igniter installation in the external heat system was designed. The initial Mod II design incorporated a fuel nozzle similar to the Mod I BOM nozzle, with a built-in center igniter. The nozzle installation incorporated a stainless-steel bellows seal between the engine insulation cover and the combustor/nozzle subassembly. After initial engine tests, it was determined that the 3-hole conical fuel nozzle yielded better operation, and this nozzle was adopted as the Mod II standard (see Section II). However, this nozzle is not compatible with a center igniter, and other provisions were required for igniter installation. In addition, the convoluted bellows seal member was viewed as an inadequate means of holding the fuel nozzle in proper, vertical alignment in the combustor. Nonvertical alignment, possibly allowed by the bellows, was suspected of causing persistent, nonuniform temperature distribution in heater-head tubes.

The configuration shown in Figure 4-1* was conceived to solve these problems. A new support body was designed for the 3-hole conical nozzle, incorporating an insertion hole and mounting clamp for an igniter installed at a 10° angle to the nozzle axis. The electrical arc occurs

*Figures are at the end of this section, beginning on page 4-8.

from the igniter center electrode to the nozzle body, and provision is made for adjusting the gap width by moving the igniter laterally. A thermally insulated mounting pedestal for the nozzle body is secured to the combustor liner/shell, with provision for small radial movement to accommodate some inaccuracy in location of the mating penetration through the insulation cover. Vertical relative motion between the combustor/nozzle unit and the insulation cover, due to differential expansion, is accommodated by total detachment of these two members, with a sliding seal to prevent gas leakage from within the EHS to outside ambient. The gas seal is a proprietary "Ener Seal" comprised of a filled Teflon member resiliently pressed against the metal sealing surfaces by an internal garter spring. Teflon filled with Ekonol was selected, as this provides the highest available temperature resistance (315°C). The pedestal and sliding seal replace the convoluted bellows and will ensure more positive alignment of the nozzle.

In addition to the combustor design described above, an EGR combustor was designed for the Mod II engine. This was done as a back-up position, in case the CGR primary design proves difficult to develop. The EGR configuration is shown in Figure 4-2 and incorporates the same nozzle/igniter support means described above.

## Configuration 4 Heater Head

The elements of the C4 heater head were defined during the previous report period. These are listed below for reference.

- Heater tube manifolds eliminated from housing casting.
- Front-row tubes enter housing over expansion space.
- Finned rear row tubes enter housing over regenerator.
- Thermal sleeves at all tube entry points to minimize effects on housing of start-up thermal transients.
- Housing is axisymmetric with individual bosses at each tube entry point, sand cast (not investment cast) in Haynes Stellite No. 31.
- 24 tubes per quadrant, Inconel 625, ID/OD = 3.0/4.5 mm.
- Finned height of rear row tubes = 95 mm.

The configuration is illustrated by Figure 4-3. During this report period, the tube configurations from the housing entry points to the bottom of the hairpin sections were finalized. This is a very challenging tube routing problem. In each quadrant, 48 tube entry lengths must be arrayed in the very restricted space between the top of the housing and the bottom plane of the hairpin tubes and combustion chambers. Ideally, the 24-tube sections over the regenerator should all have identical lengths and identical bends, to have equal pressure drop and gas flow within the cycle. The 24-tube sections over the expansion space should likewise be identical in length and configuration. This could not be achieved in practice, but was the guiding objective of the tubing vendor's work. A full-scale model of the four heater-head housings (with tube entry holes) positioned as they will be on the engine block was fabricated and delivered to the tubing vendor. This model also included a sheet metal ring to simulate the inner diameter of the preheater, which is a constraining limit on tube bend configurations.

It was found that, for manufacturability, the tubes should be made in three separate pieces, as indicated in Figure 4-3, and that the braze-joint splices should be made in couplings positioned by a quadrant floor plate, illustrated in Figure 4-4. The floorplate provides accurate positioning of the bottom ends of both front and rear rows of the hairpin, heat-transfer portions of the tubes and acts as a gas seal plate at the bottom of the combustion gas flow channel. Special fins were made to fit on the rear row tube coupling, to provide the requisite heat-transfer area within the allotted vertical height for the tubes.

Transient Thermal and Stress Analysis

The initial start-up of the Mod II engine represents a severe loading condition on the heater head housing. After the flame is ignited the heater tubes are allowed to reach a temperature such that the engine will be self-sustaining once cranked. The gas inside the heater tubes gets hot while the gas inside the heater head housing remains cold. Once cranking of the engine begins hot gas is pumped into the cold head and rapidly heats the inner surfaces of the expansion space. Because the majority of the head remains cold for some seconds, the inner regions are subjected to very high compressive stresses. If the stresses resulting from the transient temperature gradients are too high, thermal ratcheting can occur due to the start/stop cycling of the engine. A finite-element, transient, thermal and stress analysis of the heater-head housing was conducted to assess the severity of this situation.

The geometry analyzed is shown in Figure 4-3 with thermal sleeves inserted in each tube hole. These sleeves have reduced area of contact with the bore surface of the tube hole in the housing and provide some thermal insulation between the hot gas and the housing metal during start-up. Due to the discrete bosses for tube entry holes, the housing is not truly axisymmetric, making necessary a three-dimensional model for analysis. The 3-D model consisted of a $7.5^{\circ}$ arc section shown in Figure 4-5, and the finite elements comprising this model are shown in Figure 4-6. Nodes labeled P1 thru P4 proved to be points of peak compressive stress. The gas temperature history during the start-up transient was assumed to be as shown in Figure 4-7, based on Mod I operating experience. The thermal boundary conditions assumed for the transient thermal analysis are summarized in Figure 4-8.

The output from the transient thermal analysis consists of sets of nodal temperature values at each time-step interval of the analysis. These sets of temperatures are used as direct input to the structural analysis. Figure 4-9 shows the temperature plots at 12 and 25 seconds, respectively, after cranking of the engine begins.

Several stress analyses were run at various times to determine the maximum stresses that occur during start-up. The effective stresses for four critical nodes are plotted versus time in Figure 4-10. As indicated by the graph, the maximum stress in the heater head occurs at approximately 13 seconds. The temperature profile through the head at this time will be virtually the same as that at 12 seconds, shown in Figure 4-9. The stress contour plot at 12 seconds, illustrated in Figure 4-11, shows the stress concentrations around the gas ports.

In order to ensure the life/reliability requirements of the heater heads, a set of design criteria was established. The high-cycle fatigue requirement in the heater head must exhibit a safety factor of at least 1.5 under maximum temperature and pressure conditions. For low-cycle fatigue resistance, the head must exhibit a minimum safety factor of 1.5 against fatigue due to start/stop cycles ($10^4$ cycles required). Generally a minimum safety factor of 1.0 against yielding is required for thermal stresses. However, some yielding can be tolerated if the pressure stresses are small. Therefore, a third criterion will be included on the design. A minimum of a 1.0 safety factor against thermal ratcheting must be exhibited. Thermal ratcheting would occur if during the first half-cycle an area yields and during the second half-cycle the same area yields in the opposite direction. This criterion applies only to thermal stresses, since yielding due to pressure stresses alone is not tolerable.

The conclusions from the analysis are summarized by the safety factors determined for peak-stress points P1 thru P4, given in Table 4-1.

TABLE 4-1
SAFETY FACTORS

| Node | Start/Stop Cycles | Maximum Power Cycles | Thermal Ratcheting R.T. | 800°C |
|---|---|---|---|---|
| P1 | 1.07 | 3.63 | 1.73 | 1.84 |
| P2 | 1.34 | 2.87 | 6.20 | 1.57 |
| P3 | 1.22 | 5.53 | 3.25 | 2.99 |
| P4 | 1.25 | 1.27 | 4.00 | 0.59 |

At one location, P4, the thermal ratcheting criterion is not met. The validity of the results obtained for location P4, however, is questionable. For modeling simplicity the zone was modeled as a sharp point, tending to increase the resulting stresses significantly. The location on the actual heater head has an approximate radius of 2 mm. Actual stress at this location will be significantly lower, due to this radius. With this qualification, the design criteria for high-cycle fatigue and thermal ratcheting are met. The fatigue criterion for the start/stop cycling is not met; however, the analysis has some built-in conservatisms in the assumed thermal boundary conditions, and actual safety factors will be higher.

## Mod II Analysis

Initial characterization of the first-build Mod II engine was achieved early in the next report period, and analysis of the characterization data performed. Code predictions for the first build, duplicating the actual engine configuration and the test conditions run, were used as the basis for comparison of the test data. A configuration summary of the first build, highlighting items affecting performance level, is listed below:

- Engine configuration is close to a BSE. All auxiliaries are externally powered with the exception of the oil pump, water pump, and hydrogen compressor. These three components are an integral part of the drive system and balance arrangement.

- Heater head configuration No. 1 is installed. This heater head is an early design that incorporates two manifolds in the heater casting and a resultant heater tube cage design that connects the front row (closest to combustor) tubes to the regenerator instead of the cylinder side as in previous ASE practice and with the final Mod II configuration 4 design.

- Atomizing airflow rate and combustion excess air ratio (λ) differed from final design values.

- Engine coolant is 50% water/ethylene glycol mixture with engine inlet water temperature set at 50°C. Vehicle operation would be with radiator top tank temperature (engine outlet) equal to 50°C.

The predicted performance of the first build Mod II differs considerably from that of the final Mod II as a result of the above changes and operational differences. The primary reason for the difference is the impact of the heater head configuration. A comparison is shown in Figure 4-12. Configuration No. 1 performance is inferior due to higher conduction losses with XF-818, lower mean working gas temperature due to the heater tube routing, and increased dead volume. Configuration No. 4 heater head has no manifolds, reducing dead volume, heater front row tubes enter the cylinder side, and the housing (for demonstration purposes) is lower conducting HS-31 instead of XF-818. Configuration No. 1 heater head requires a variable set temperature to prevent overtemperaturing of the front row tubes at low loads (Figure 4-13) while configuration 4 does not. The production configuration 4 heaters will be made of a non-cobalt material (NASA/UT) which has material properties similar to HS-31. This material is still in the development stage.

In addition to the heater head configuration differences, EHS performance is not as good as the final Mod II as a result of higher atomizing airflow (1.0 g/s as opposed to 0.8 g/s for the final build) and higher $\lambda$ values at low loads (Figure 4-14). The higher atomizing airflow rate causes a reduction in EHS efficiency, particularly at low loads, due to the introduction of a higher amount of low-temperature air into the combustion zone. Higher combustion airflow rates cause only a small change in preheater performance in the BSE configuration, but become more significant in the full engine system configuration due to the increase in combustion air blower power requirement.

Characterization of the engine water pump in a test rig also indicated that pump power consumption was higher than design values. The comparison of the test data to values used for initial design predictions is shown in Figure 4-15. Code predictions of the first-build performance were adjusted to reflect the actual, measured water pump power.

Comparison of the first-build test data to predictions in power and efficiency are shown in Figure 4-16 and 4-17. The following noteworthy conclusions can be reached from this comparison:

- Efficiency levels, especially at higher mean pressure levels, are very close to predictions.
- At lower mean pressure levels, the efficiency is somewhat lower than predictions. The power output at lower mean pressures is also lower than predicted. The combined effect of these two differences results in achieving predicted efficiency levels at a given power; however, a mean pressure higher than predicted is required.
- Power shortfall relative to design increases approximately linearly with speed up to 2000-3000 rpm and extrapolates approximately to 0 at 0 rpm (Figure 4-18). This indicates that the shortfall is likely caused by higher than anticipated parasitic (friction) losses. The zero intercept indicates that leakage loss predictions are realistic.
- At the higher speeds, the shortfall is nonlinear and begins to decrease with speed, indicating that cycle pumping losses at the higher speeds are less than predicted.
- The shortfall shows pressure dependency increases with increasing mean pressure, up to 9 MPa, and then remains fairly constant up to 15 MPa. The reason for this discrepancy is now being determined.

Flow tests of the configuration No. 1 heater head were conducted to assess internal pressure drop (see Section II). Tests indicated a higher than anticipated pressure drop in the neck area between the manifolds and housing. Pressure drops in the heater tubes, manifolds, and in the regenerator side of the housing were essentially as predicted. A careful review of the casting geometry revealed that there were no restricting "pinch points" in the flow cross-sectional area in the neck region, therefore it was concluded that the pressure drop increase was most likely caused by swirl. The higher than anticipated pressure drop could be a contributing factor to the power shortfall. The final configuration 4 heater will not have the manifold/neck configuration and therefore will not experience this problem.

Initial test results indicated that tube temperature spreads increased to unacceptable levels at low load (Figure

4-19). Investigation of temperatures in the various quadrants indicated that the "upright" quadrants (Figure 4-20) had decreasing temperature with lower loads while the "inclined" quadrants had temperatures remaining fairly constant. Possible reasons for this occurrence were

- EHS system hardware circumferential nonuniformity
- Nozzle fuel spray pattern nonuniform at low flow rates
- Increased combustion gas recirculation at low loads due to buoyancy effects.

Tests were conducted to assess these factors. Rotation of EHS components (combustor and preheater) showed no change and indicated that EHS hardware uniformity was not the problem. The recirculation effect was evaluated by increasing levels of excess combustion airflow ($\lambda$) thereby increasing the contribution of forced convection through the heater and decreasing the probability of recirculation due to free convection (or buoyancy) effects. As noted in Figure 4-21, higher $\lambda$ values reduced temperature spread, indicating buoyancy effects were contributing to the problem. Nozzle spray pattern was evaluated by using different design nozzles (upgraded Mod I and conical), which have different low-flow spray patterns. The conical nozzle, known to have better low-flow characteristics, improved the low-load temperature spread, similar to the effect of higher $\lambda$. It was concluded that the low-load problem was a combination of both bouyancy and fuel spray pattern and that temperature spread could be minimized by use of the conical nozzle. These results are also discussed in Section II.

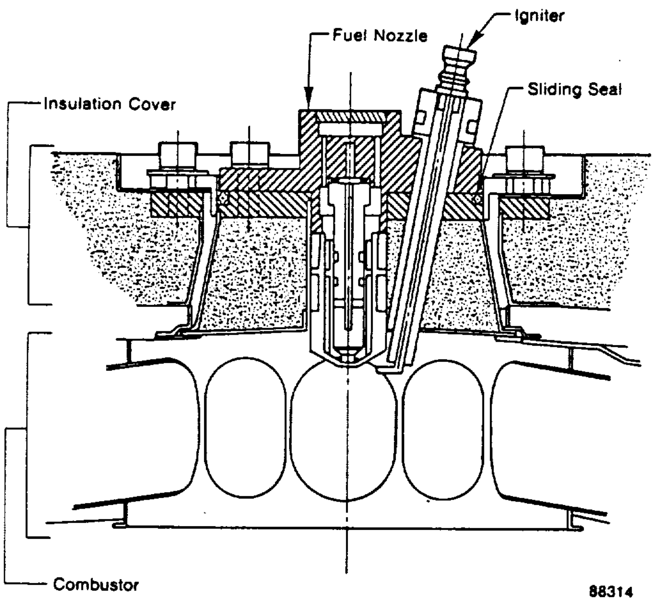



Fig. 4-1 Alternative Support and Seal for 3-Hole Conical Nozzle and Igniter in Mod II CGR Combustor

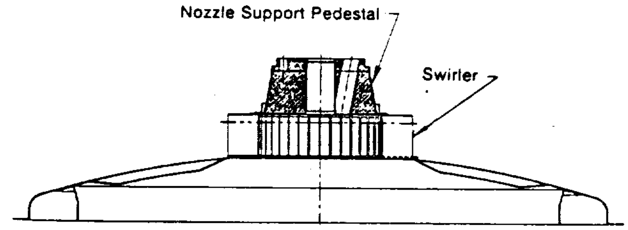



Fig. 4-2 EGR Combustor for Mod II Engine

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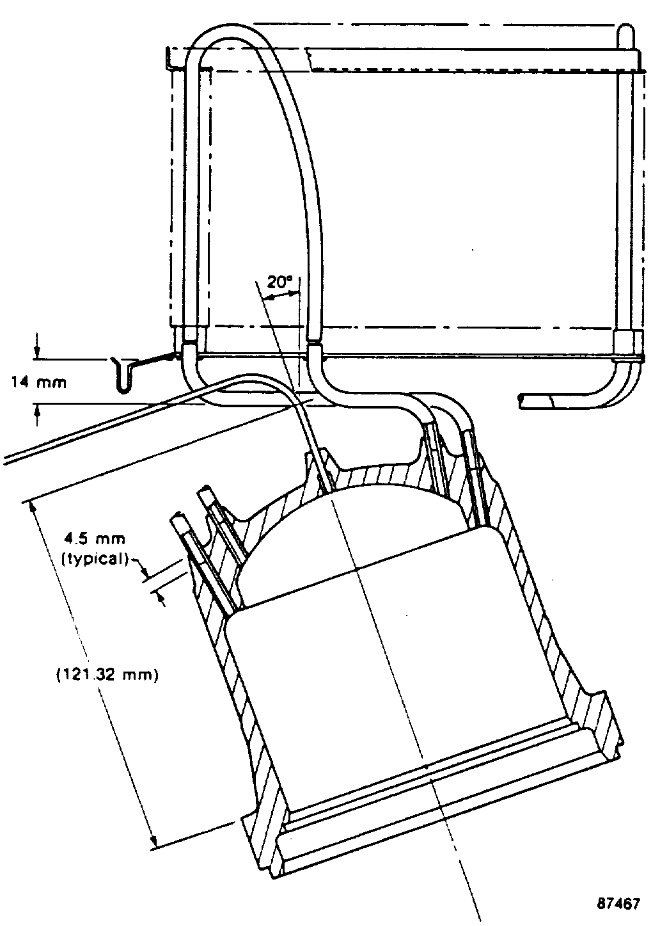



Fig. 4-3 Mod II Configuration No. 4 Heater Head with 3-Piece Tubes

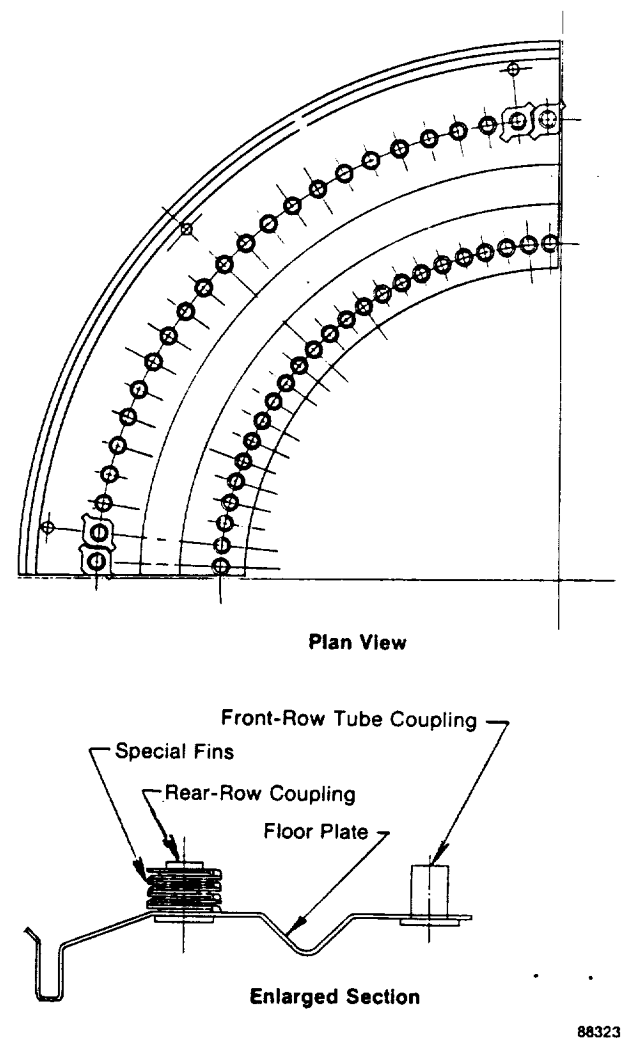



Fig. 4-4 Floor Plate for Configuration 4 Heater Head Quadrant

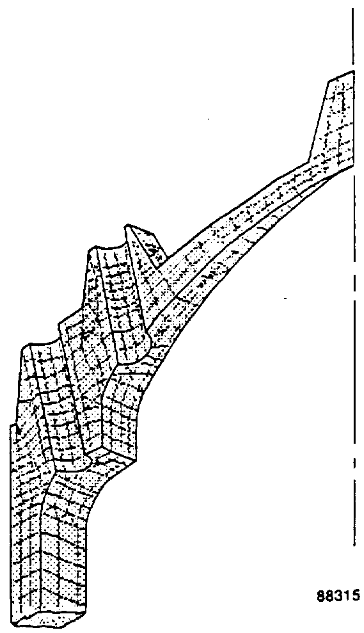



**Fig. 4-5 Mod II Configuration No. 4 Heater Casting Finite Element Model, 3-D View (Tubes, Sleeves, and Ring Removed)**

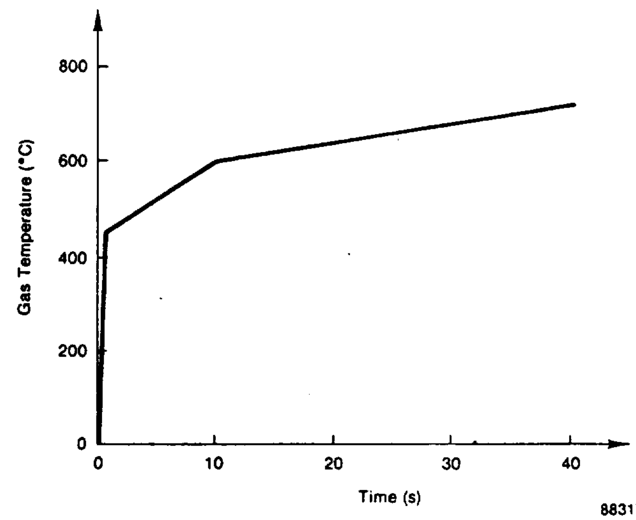



**Fig. 4-7 Mod II Gas Temperature History during Start-up Transient**

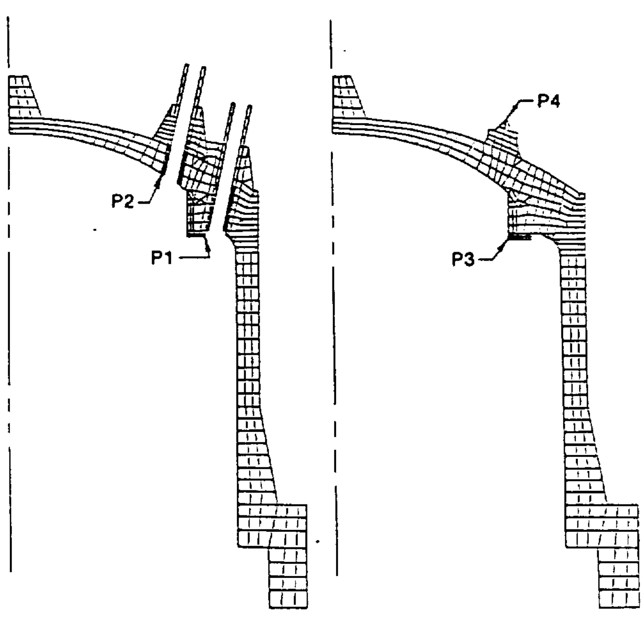



**Fig. 4-6 Mod II Configuration No. 4 Heater Casting Finite Element Model, Sectional Views**

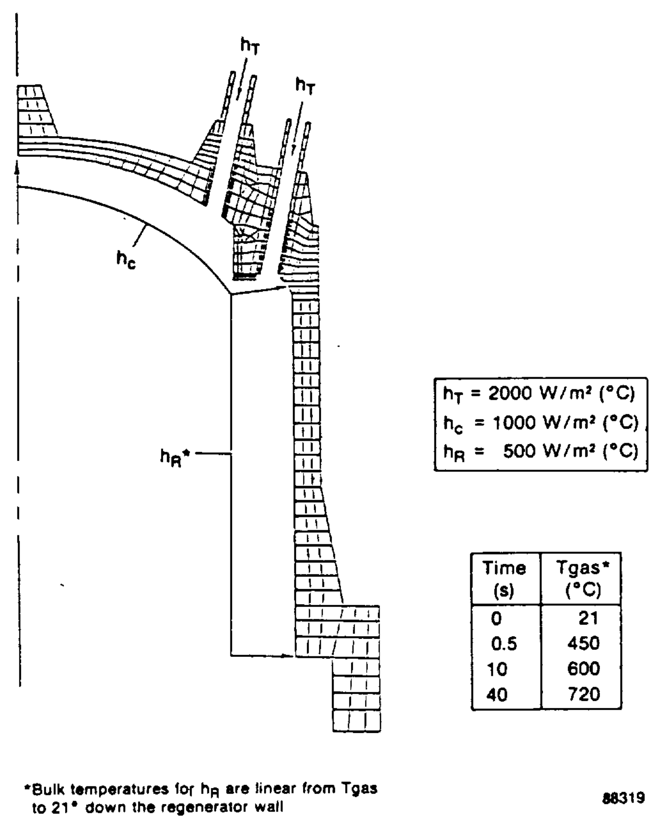



**Fig. 4-8 Mod II Configuration No. 4 Heater Casting Thermal Boundary Conditions**

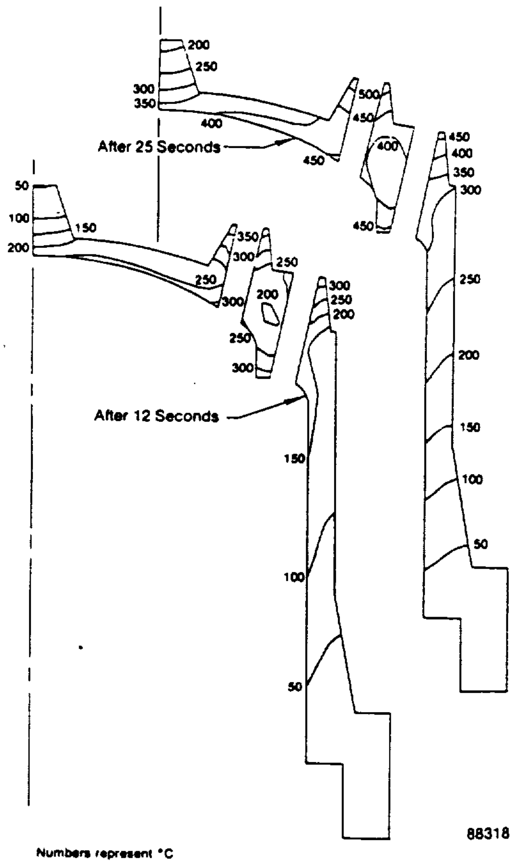



Fig. 4-9 Mod II Configuration No. 4 Heater Casting Transient Temperature Profile

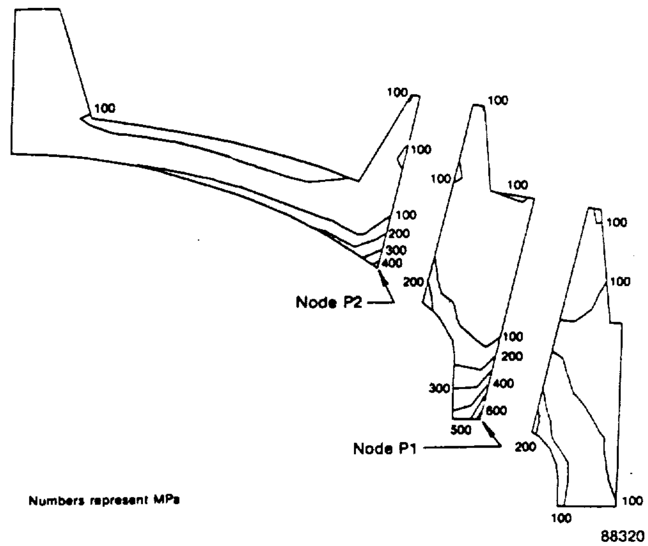



Fig. 4-11 Mod II Configuration No. 4 Heater Casting Effective Stress Distribution at 12 sec

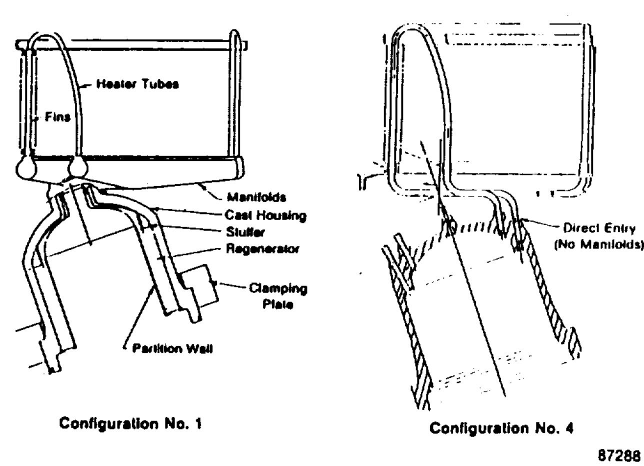



Fig. 4-12 Configuration No. 1 vs No. 4 Heater Head Comparison

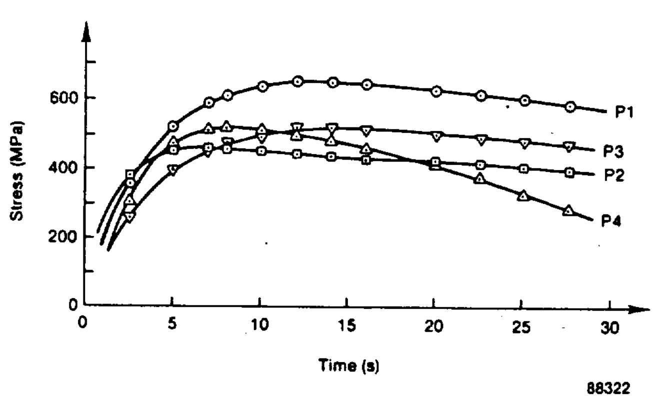



Fig. 4-10 Mod II Configuration No. 4 Heater Housing Stress vs Time for Four Critical Nodes

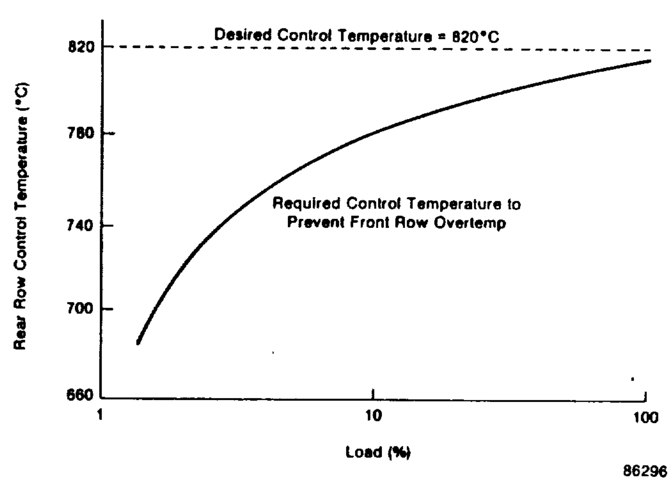



Fig. 4-13 Mod II Configuration No. 1 Heater Head Front Row Temperature Restriction

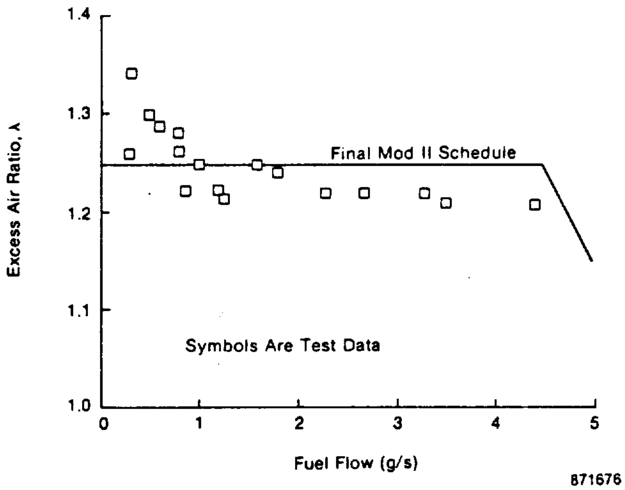



**Fig. 4-14 Mod II Engine Lambda Comparison**

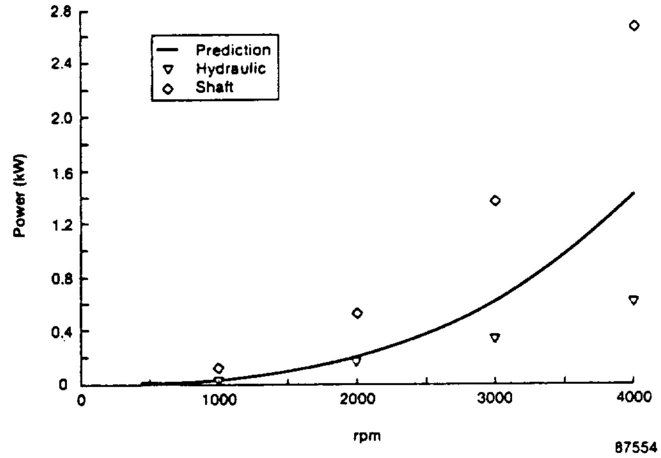



**Fig. 4-15 Mod II Water Pump Losses**

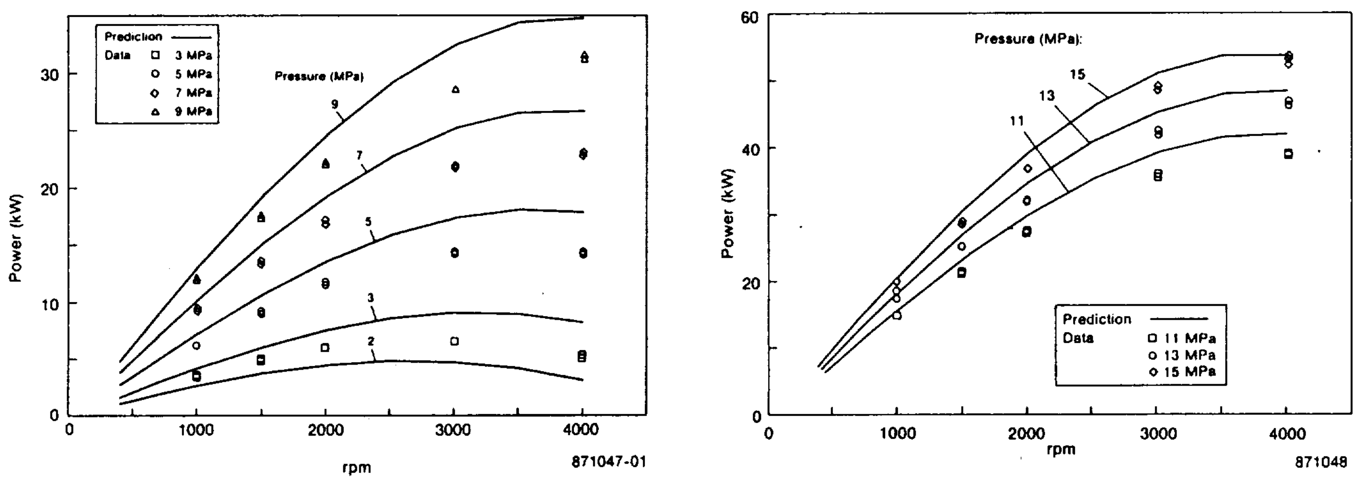



**Fig. 4-16 Mod II BSE Power Levels**

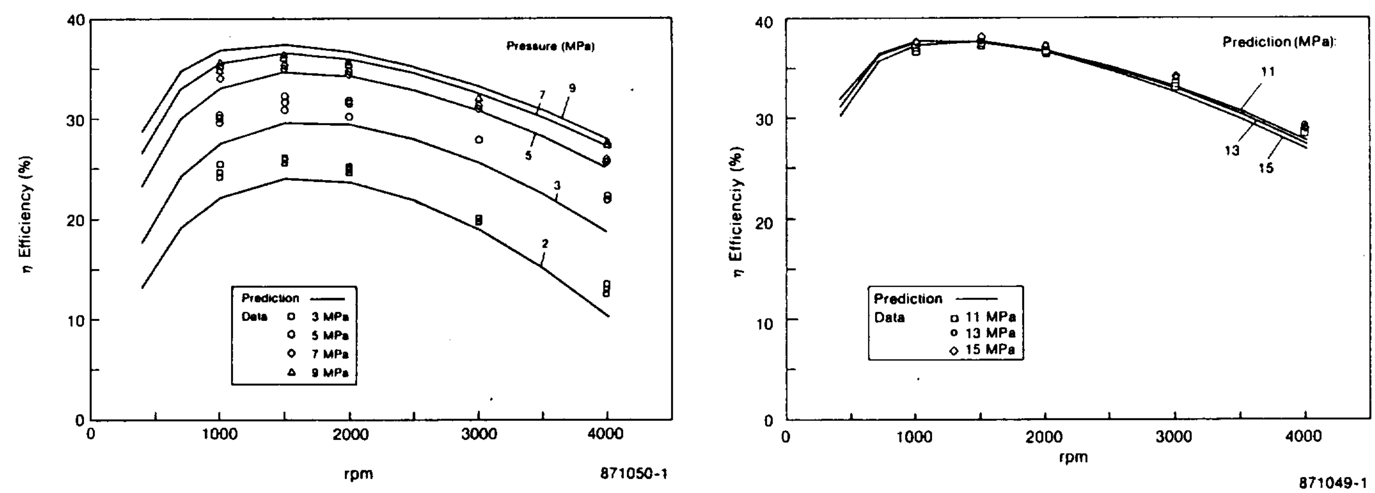



**Fig. 4-17 Mod II BSE Efficiency**

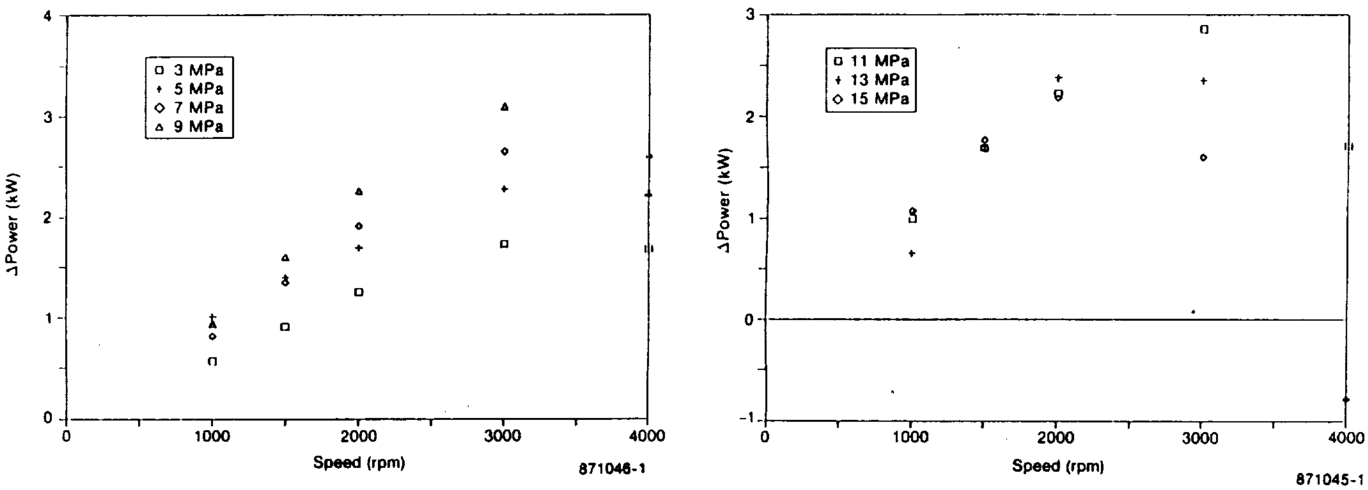



**Fig. 4-18 Mod II BSE Power Deviation from Predictions**

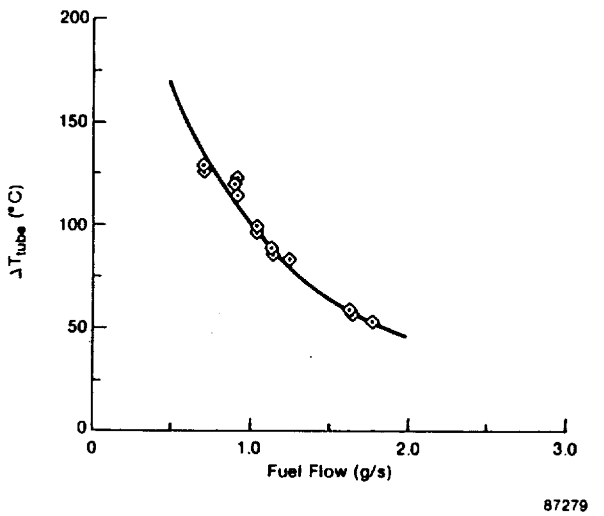



**Fig. 4-19 Mod II Engine Heater Temperature Spread at Low Load**

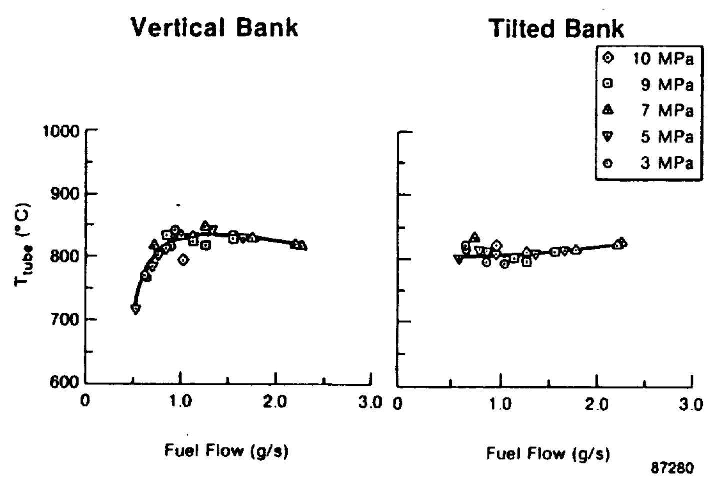



**Fig. 4-20 Mod II Engine Heater Quadrant Tube Temperature Comparison**

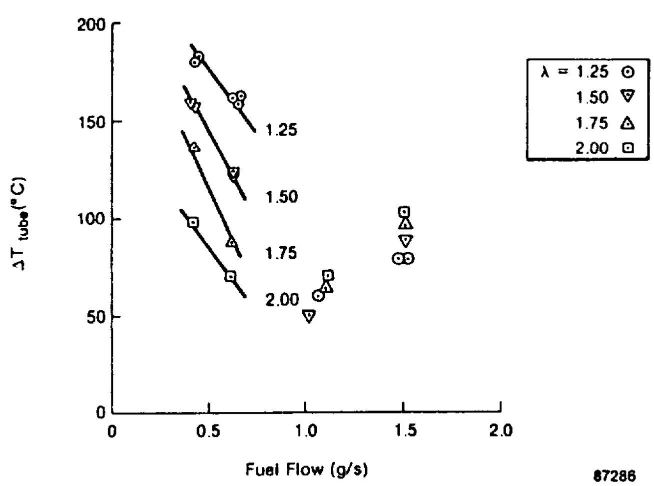



**Fig. 4-21 Effect of Lambda on Heater Head ΔTs**

# V. TECHNICAL ASSISTANCE

## NASA Technology Utilization Program

During this report period, installation of an upgraded Mod I engine in an Air Force multistop van was completed. The van was released to the Air Force for in-service evaluation and 319 hours of operation and 1200 miles were accumulated.

Efforts required to modify the van for the upgraded Mod I engine involved (1) removal of the stock diesel engine, (2) modification of the front crossmember to accommodate the deeper upgraded Mod I engine (Figure 5-1),* (3) installation of a Chrysler 3-speed automatic transmision, in place of the stock GM transmission to take advantage of the experience obtained during the ASE program, (4) installation of gas-fired heaters for windshield defogging and personnel heating, and (5) minor driver compartment modifications for Stirling-specific instrumentation and controls. The engine control and monitor were mounted in the rear compartment of the van to provide monitoring and diagnostic capabilities without changing the driver compartment area.

In preparation of the vehicle, particular attention was paid to safety. Design of the hydrogen working gas system included separation of hydrogen and gasoline lines, isolation of ignition sources, ventilation of the engine compartment to prevent hydrogen accumulation, protection of the driver, protection of the hydrogen system from collision, and minimization of potential hydrogen leakage. An extensive hazard analysis was performed on the van before release to the Air Force (MTI Report 86ASE526HA1).

Training and familiarization sessions were held with Air Force personnel at the initiation of van operation. Fire and safety personnel were briefed on the van and its systems, drivers were trained, and the Military Equipment Evaluation Personnel (MEEP) were trained to perform routine service operations.

The van (Figure 5-2) was put in service at Langley Air Force Base on 9 September. Accumulation of in-service experience since then is shown in Figure 5-3. The van is assigned to an aircraft maintenance unit and is used to transport personnel and their equipment to the flight line to service aircraft. Operating time can be 18 hours per day or more depending on aircraft flight plans. A recording device (tachograph) was mounted in the vehicle to record engine speed, vehicle speed, and miles traveled. These parameters are recorded 24 hours per day, 7 days per week. The recordings are removed once per week during regular van service. As shown in Figure 5-4, the duty cycle has extended periods of idle time and driving periods characterized by highly cyclical speed variations. There is essentially no constant speed cruise operation.

In-service operation of the vehicle has shown the basic engine integrity to be excellent. Operational problems have been associated with components other than the basic engine. Major problems have been

- Personnel Heaters. Cabin fumes were eliminated by changing air ducting.

*Figures are at the end of this section, beginning on page 5-3.

- Control Interference from Walkie-Talkies. Solved by installation of external antenna and mounting adapter.

- Cooling Hose Cut By Alternator Belt. Resulted in loss of coolant and engine overheating; replacement of O-ring seal in cylinder liners and engine returned to service.

- Watchdog Timer Engine Stalls. Engine control watchdog timer problems caused inadvertent engine shutdowns; problem was resolved and stalls eliminated.

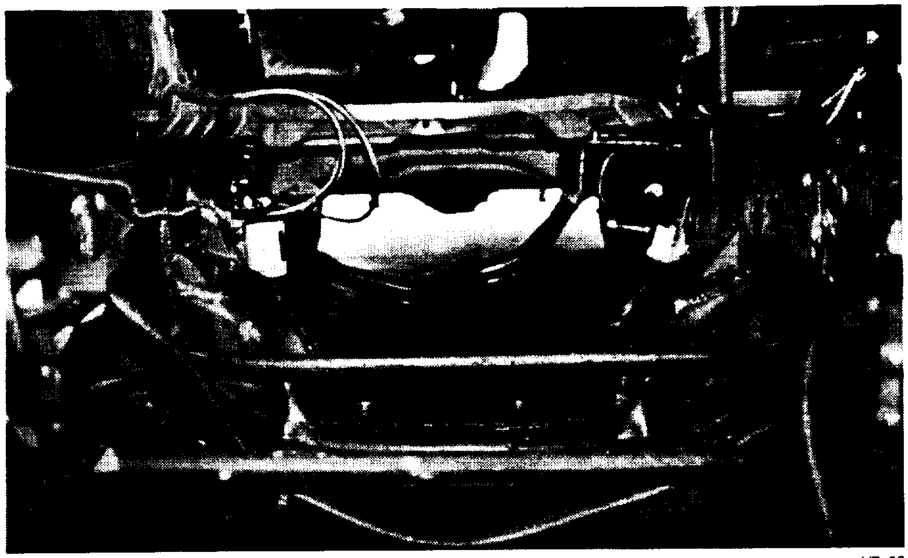
Other than the need for O-ring replacement caused by the coolant line incident, no service has been performed on the basic engine.

At completion of 400 hours of in-service operation, the fuel will be changed from unleaded gasoline to JP-4 to obtain an evaluation of the alternative fuel capability of the engine. No changes to the engine or control system are required to operate with JP-4. An additional 400 hours of operation on JP-4 are planned followed by 200 hours operation on diesel fuel. This will complete the first phase of the Air Force Van Program.

Two additional phases are planned for continuation of the van program.

Phase II - Install Upgraded Mod I in a Lighter Vehicle and Conduct Tests at Environmental Extremes. The purposes of this phase are to demonstrate acceptable over-the-road performance and to evaluate the environmental operating capability of the Stirling engine. (The current Phase I vehicle has a 75-hp upgraded Mod I engine replacing the 145-hp stock diesel engine with resultant sluggish performance. The van is considered adequate for flight-line use, where on-base speeds are limited to 35 mph and flight-line operation limited to 15 mph.) The environmental evaluation will be accomplished by stationing the van at different Air Force bases during different seasons.

Phase III - Evaluate the Final ASE Engine in a Vehicle. The first two phases are designed to evaluate feasibility of the Stirling concept. This third phase, which will occur after the ASE program, will provide an evaluation of a production-type ASE (Mod II) for eventual Air Force and postal service use.



V7-59

Fig. 5-1 Van Engine Compartment Showing Modified Crossmember

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Fig. 5-2 Air Force Multistop Van with Upgraded Mod I Engine Installed

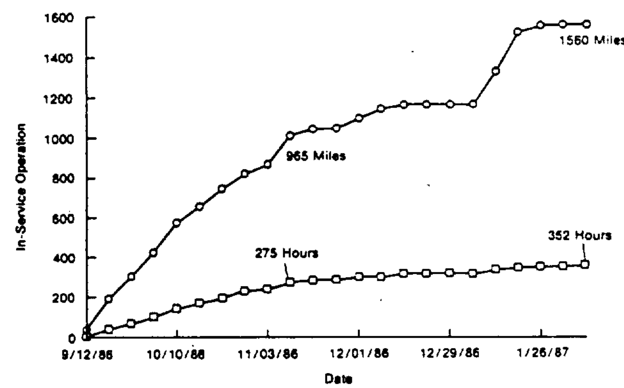


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Fig. 5-3 Air Force Van In-Service Hours and Miles

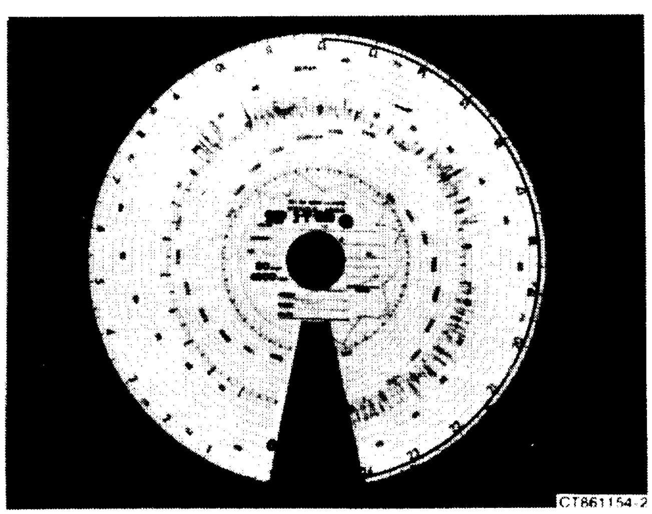

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Fig. 5-4 Typical Tachograph Plot. Outer Spikey Line Illustrates Vehicle Speed



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# VI. QUALITY ASSURANCE

## Quality Assurance Overview

The status of the ASE Program Quality Assurance Reports (QARs) as of 31 December 1986 is presented below:

| | |
|---|---|
| Open QARs (pending further analysis and/or NASA approval: | 179 |
| Closed QARs (total to date) | 2090 |
| P-40 QARs | 272 |
| Mod I QARs | 626 |
| Upgraded Mod I QARs | 961 |
| Preliminary Mod II QARs (durability rig hardware) | 49 |
| Mod II QARs | 323 |
| Total QARs in system | 2269 |

Program QAR activity for the second half of 1986 is as follows:

| | |
|---|---|
| New QARs (for last six months) | 208 |
| P-40 QARs | 0 |
| Mod I QARs | 0 |
| Upgraded Mod I QARs | 88 |
| Preliminary Mod II QARs (Durability Rig) | 0 |
| Mod II QARs | 115 |

## Mod I QAR Experience

A summary of trend-setting problems documented via the QAR system is presented in Table 6-1* and Figures 6-1 through 6-4. 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 30 June 1983 are shown in comparison with the results of this reporting period and that of previous semiannual report periods.

Table 6-2 is a summary of the operating times versus failures for all active ASE Program Mod I/Upgraded Mod I engines.

TABLE 6-2
OPERATING TIMES VERSUS FAILURES
AS OF DECEMBER 31, 1986

| Engine No. | Operation Time (hr) | Mean Operating Time to Failure (hr) |
|---|---|---|
| 3 | 2376 | 132 |
| 5 | 1995 | 154 |
| 6 | 4060 | 508 |
| 7 | 4480 | 1120 |
| 8 | 1217 | 304 |
| 9 | 393 | 56 |
| 10 | 156 | 52 |

*Table 6-1 is on the following page.

TABLE 6-1

MAJOR PROBLEMS SUMMARY

| Established Prior to 6/30/83 | | % of Total | Reports from 7/1/83-12/31/83 | % of Total | Reports from 1/1/84-6/30/84 | % of Total | Reports from 7/1/84-12/31/84 | % of Total | Reports from 1/1/85-6/30/85 | % of Total |
|---|---|---|---|---|---|---|---|---|---|---|
| Moog Valve | 19 | 76.0 | 1 | 4.0 | 1 | 4.0 | 1 | 4.0 | 1 | 4.0 |
| Heater Head | 13 | 40.8 | 3 | 9.4 | 3 | 9.4 | 5 | 15.6 | 6 | 18.7 |
| Check Valves | 8 | 32.0 | 3 | 12.0 | 2 | 8.0 | 4 | 16.0 | 3 | 12.0 |
| Combustion Blower | 12 | 41.3 | 6 | 20.7 | 6 | 20.7 | 3 | 10.3 | 1 | 3.5 |
| Fuel Nozzle | 13 | 20.0 | 6 | 9.2 | 13 | 20.0 | 5 | 7.7 | 5 | 7.7 |
| Igniter | 5 | 50.0 | 1 | 10.0 | 0 | -- | 2 | 20.0 | 0 | -- |
| Preheater | 7 | 22.6 | 7 | 22.6 | 1 | 3.2 | 7 | 22.6 | 1 | 3.2 |
| Atomizing Air Comp./Servo-Oil Pump | 6 | 50.0 | 1 | 8.3 | 2 | 16.7 | 1 | 8.3 | 2 | 16.7 |
| Combustor | 7 | 18.4 | 4 | 10.5 | 2 | 5.3 | 4 | 10.5 | 5 | 13.2 |
| Flameshield | 6 | 24.0 | 3 | 12.0 | 0 | -- | 10 | 40.0 | 4 | 16.0 |
| PL Seal Assy. | 8 | 24.2 | 3 | 9.1 | 8 | 24.3 | 7 | 21.2 | 1 | 3.0 |
| Crankcase/ Bedplate | 0 | -- | 2 | 40.0 | 0 | -- | 1 | 20.0 | 2 | 40.0 |
| Piston Rod | 0 | -- | 3 | 30.0 | 4 | 40.0 | 1 | 10.0 | 1 | 10.0 |

| | Reports from 7/1/85-12/31/85 | % of Total | Reports from 1/1/86-6/30/86 | % of Total | Reports from 7/1/86-12/31/86 | % of Total |
|---|---|---|---|---|---|---|
| Moog Valve | 2 | 8.0 | 0 | -- | 0 | -- |
| Heater Head | 0 | -- | 2 | 6.3 | 0 | -- |
| Check Valves | 2 | 8.0 | 3 | 12.0 | 0 | -- |
| Combustion Blower | 1 | 3.5 | 0 | -- | 0 | -- |
| Fuel Nozzle | 13 | 20.0 | 9 | 13.9 | 1 | 1.5 |
| Igniter | 2 | 20.0 | 0 | -- | 0 | -- |
| Preheater | 5 | 16.1 | 3 | 9.7 | 0 | -- |
| Atomizing Air Comp./Servo-Oil Pump | 0 | -- | - NO LONGER USED - | | | |
| Combustor | 12 | 31.6 | 1 | 2.6 | 3 | 7.9 |
| Flameshield | 2 | 8.0 | 0 | -- | 0 | -- |
| PL Seal Assy. | 6 | 18.2 | 0 | -- | 0 | -- |
| Crankcase/ Bedplate | 0 | -- | 0 | -- | 0 | -- |
| Piston Rod | 1 | 10.0 | 0 | -- | 0 | -- |

| | | | |
|---|---|---|---|
| Mod I hours accumulated prior to 6/30/83 | - 2689 | % of hours | - 16 |
| Mod I hours accumulated from 7/1/83 to 12/31/83 | - 1881 | % of hours | - 11 |
| Mod I hours accumulated from 1/1/84 to 6/30/84 | - 1848 | % of hours | - 11 |
| Mod I hours accumulated from 7/1/84 to 12/31/84 | - 2263 | % of hours | - 14 |
| Mod I hours accumulated from 1/1/85 to 6/30/85 | - 2731 | % of hours | - 16 |
| Mod I hours accumulated from 7/1/85 to 12/31/85 | - 2787 | % of hours | - 17 |
| Mod I hours accumulated from 1/1/86 to 6/30/86 | - 1623 | % of hours | - 10 |
| Mod I hours accumulated from 7/1/86 to 12/31/86 | - 766 | % of hours | - 5 |
| TOTAL | 16588 | | 100 |

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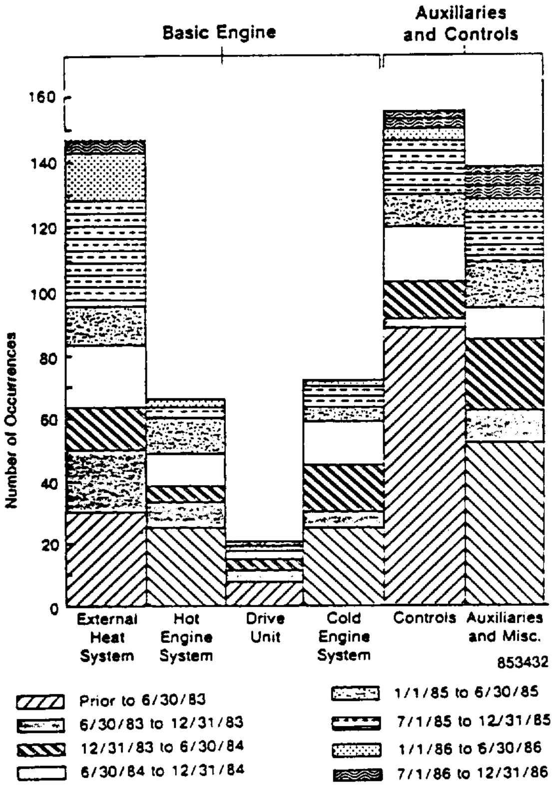


**Fig. 6-1 Major Mod I Engine Systems Failures and Discrepancies through December 31, 1986**

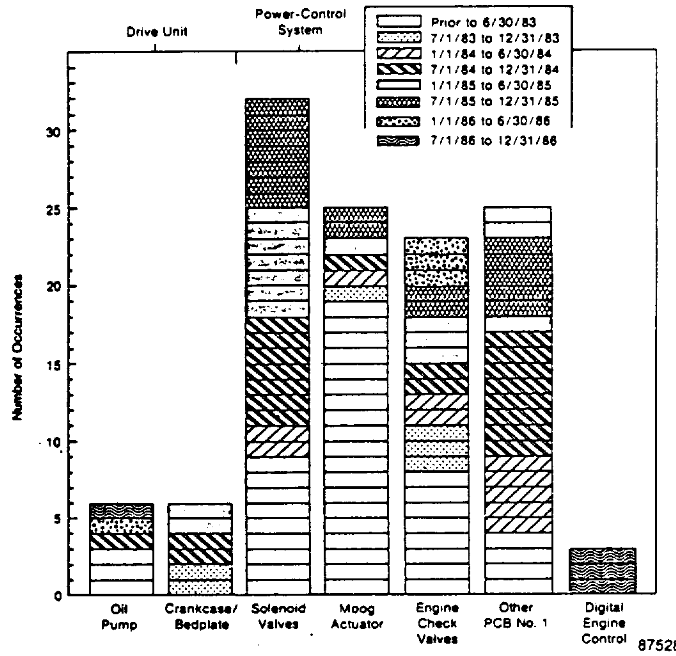


**Fig. 6-2 Mod I Drive Unit and Power Control System Failures and Discrepancies through December 31, 1986**

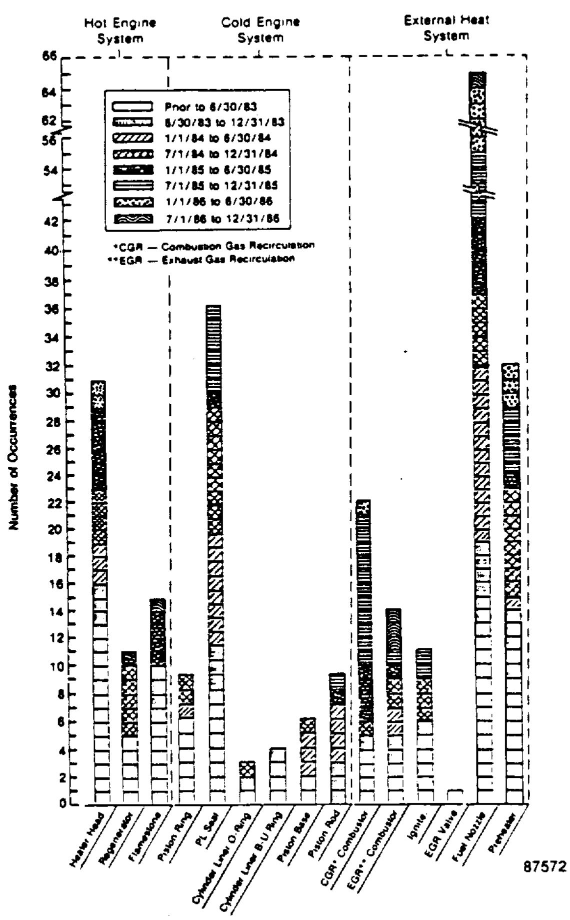


**Fig. 6-3 Mod I Hot Engine, Cold Engine, and EHS Failures and Discrepancies through December 31, 1986**

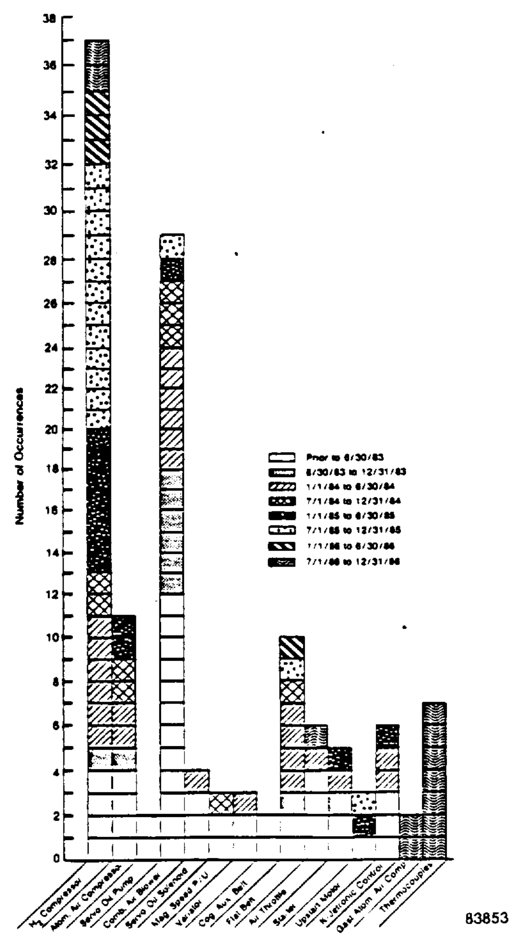


**Fig. 6-4 Mod I Auxiliaries and Miscellaneous Failures and Discrepancies through December 31, 1986**

| | | |
|---|---|---|
| 1. Report No. NASA CR 180839 | 2. Government Accession No. DOE/NASA/0032-30 | 3. Recipient's Catalog No. |
| 4. Title and Subtitle Automotive Stirling Engine Development Program Semiannual Technical Progress Report for Period 1 July - 31 December 1986 | | 5. Report Date February 1988 |
| | | 6. Performing Organization Code |
| 7. Author(s) R. Farrell, C. Hindes, R. Battista, M. Connelly, M. Cronin, R. Howarth, A. Donahue, E. Slate, R. Stotts, R. Lacy, M. Simetkosky, D. Karl, J. Meacher, A. Richey, and J. Mundy | | 8. Performing Organization Report No. 87SAE555SA11 |
| 9. Performing Organization Name and Address Stirling Engine Systems Division Mechanical Technology Incorporated 968 Albany-Shaker Road Latham, New York 12110 | | 10. Work Unit No. |
| | | 11. Contract or Grant No. DEN3-32 |
| 12. Sponsoring Agency Name and Address U.S. Department of Energy Conservation and Renewable Office of Vehicle and Engine R&D - Washington, D.C. | | 13. Type of Report and Period Covered Semiannual Technical 1 July - 31 December 1986 |
| | | 14. Sponsoring Agency Code |

15. Supplementary Notes

Semiannual Technical Progress Report prepared under Interagency Agreement DE-AI01-85CE50112, NASA Project Manager - William Tomazic, Power Technology Division, NASA/Lewis Research Center, 2100 Brookpark Road, Cleveland, OH 44135

16. Abstract

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

This report summarizes the study of high-power kinematic Stirling engines for transportation use, testing of Mod I and Mod II Stirling engines, and component development activities. Mod II development testing was performed to complete the development of the basic engine and begin characterization of performance. Mod I engines were used for Mod II component development and to obtain independent party (U.S. Air Force) evaluation of Stirling engine vehicle performance.

| | | | |
|---|---|---|---|
| 17. Key Words (Suggested by Author(s)) automotive Stirling engine, Mod I engine, upgraded Mod I engine, seals, reference engine, NASA Technology Utilization van, Mod II engine, heater head, emissions, and alternative fuels. | | 18. Distribution Statement Unclassified - unlimited Star Category 85 DOE Category UC-96 | |
| 19. Security Classif. (of this report) Unclassified | 20. Security Classif. (of this page) Unclassified | 21. No. of Pages | 22. Price* |

* For sale by the National Technical Information Service, Springfield, Virginia 22151

MTI-17983

NASA-C-168 (Rev. 6-71)