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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

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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, multi-fuel capability, low-emissions, and low-noise potential of the Stirling engine made it a prime candidate for an alternative automotive-propulsion system.

ASE Program logic called for the design of a Reference Engine System Design (RESD) to serve as a focal point for all component, subsystem, and system development within the program. The RESD was defined as the best-engine design generated within the program at any given time. The RESD would incorporate all new technologies with reasonable expectations of development by 1984, and which provide significant performance improvements relative to the risk and cost of their development. The RESD would also provide the highest fuel economy possible while still meeting other program objectives.

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

Through the course of the program, the Mod I has been modified and upgraded wherever possible, to develop and demonstrate technologies incorporated in the RESD. As a result, the program followed a "proof-of-concept" development path whereby the Upgraded Mod I design emerged as an improved engine system, proving specific design concepts and technologies in the Mod II that were not included in the original Mod I design. This logic was recognized as having inherent limitations when it came to actual engine hardware, since Mod I hardware was larger and, in some cases, of a fundamentally different design than that of the Mod II.

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

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

The 1983 RESD configuration was changed substantially from previous designs to achieve these goals. The new design uses a single-shaft V-drive, rather than the two-shaft U-drive system of the Mod I; an annular heater-head and regenerator rather than the previous cannister configuration; and a simplified control system and auxiliary components. By these measures, the projected manufacturing cost of the May 1983 RESD was reduced by more than 25% and total engine system weight was reduced by 47% in comparison to the Upgraded Mod I engine, while engine efficiency and power remained approximately the same. This updated RESD has a predicted combined mileage of 41.1 mpg using unleaded gasoline, which is 50% above the projected spark-ignition engine mileage for a 1984 X-body vehicle with a curb weight of 2870 lb.

Since the RESD update in May 1983, 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; 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, etc. This initial optimization was then honed and refined through many successfully smaller iterations, including a preliminary final version presented at the basic Stirling engine (BSE) Design Review in April 1985. 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 (3000-3500 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 internal combustion (IC) baseline for the Mod II evaluation.

The major accomplishments of this report period were the completion of the BSE and SES designs on schedule, the approval and acceptance of those designs by NASA and the initiation of manufacture of BSE components. The 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.

At this time there is a total of five Mod I engines under test in the ASE program with 11,412 Mod I engine test hours accumulated. This is an increase of over 2700 hrs since the previous Semiannual report (84ASE408SA6). Engines No. 6 and 7 are engaged in concentrated endurance tests of high temperature (820°C) heater head operation and seals/piston ring life, respectively. Upgraded Mod I engine No. 10 has been removed from the program and shipped to the NASA/Lewis Research Center for NASA test programs.

The ASE program remains on schedule for the first testing of a Mod II engine in January 1986. During the next Semiannual period attention will be focused on issuing the remaining Mod II drawings for manufacture, completion of the procurement of Mod II BSE hardware, initiation of the procurement of Mod II SES hardware and most importantly the completion of the assembly of Mod II engine No. 1.

I. SUMMARY

Since the inception of the ASE Program in 1978, 13 Quarterly Technical Progress Reports were issued under NASA Contract No. DEN3-32, "ASE Development Program". However, reporting was changed to a Semiannual format in July 1981. This report, the eighth Semiannual Technical Progress Report issued under the contract, and covering the period of January 1 through June 30, 1985, includes technical progress only. In the interest of reporting the Mod II design in a single Semiannual, the technical content concerning the Mod II engine also includes work up to the SES Design Review in August 1985. The Mod II engine is shown in Figure 1-1.

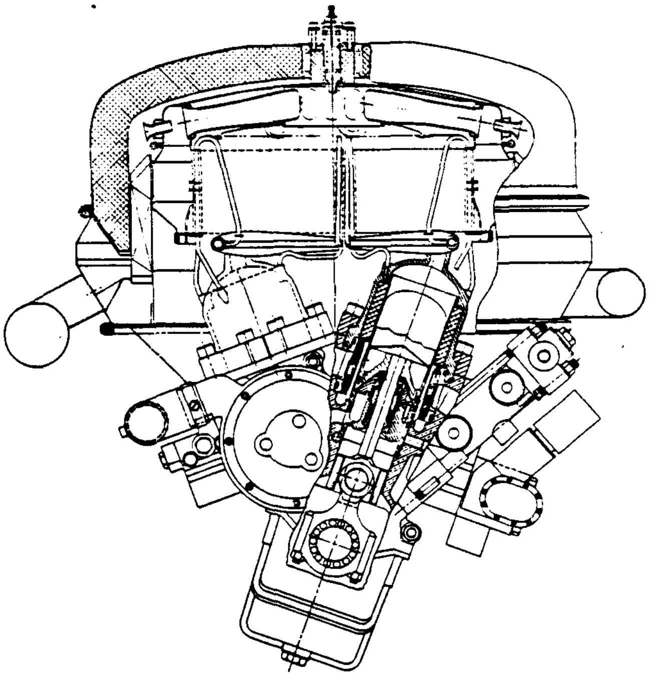

Figure 1-1 Mod II Engine

Overall Program Objectives

The overall objective of the ASE Program is to develop an ASE system which, when installed in a late-model production vehicle, will:

- Demonstrate an improvement in combined metro/highway fuel economy of at least 30% over that of a comparable spark-ignition-engine powered production vehicle, based on EPA test procedures.*

- Show the potential for emissions levels less than: NO $\leq$ 0.4 g/mi, HC $\leq$ 0.41 g/mi, CO $\leq$ 3.4 g/mi, and a total particulate level $\leq$ 0.2 g/mi after 50,000 miles.

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

- Ability to use a broad range of liquid fuels from many sources, including coal and shale oil.

- Reliability and life comparable to current-market powertrains.

- A competitive initial cost and a life-cycle cost comparable to conventionally powered automotive vehicles.

- Acceleration suitable for safety and consumer considerations.

- Noise/safety characteristics that meet the currently legislated or projected Federal Standards.

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

Major Task Descriptions

The major ASE Program tasks are described below:

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

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

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

Task 3 - Technology Familiarization - The USAB P-40 Stirling engine, which was available at the beginning of the program, was used as a baseline for familiarization; to evaluate current engine operating conditions and component characteristics; as a test bed for component/subsystem performance improvement; and, to define problems associated with vehicle installation. Three P-40 engines were built and delivered to the U.S. team members - one was installed in a 1979 AMC Spirit. A fourth P-40 engine was built and installed in a 1977 Opel sedan for testing in Sweden. The baseline P-40 engines were tested in dynamometer test cells and automobiles. Test facilities were constructed at MTI to accommodate the engine test program and to demonstrate technology transfer.

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

Task 4 - Mod I Engine - The Mod I (a first-generation ASE design) was developed using USAB P-40 and P-75 engine technology. The prime objective was to increase power density and overall engine performance.

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

Three Mod I engines were manufactured by USAB and tested in dynamometer test cells to establish their performance, durability, and reliability. Continued testing and development was necessary to meet preliminary design performance predictions. One additional Mod I engine was manufactured, assembled, and tested in the United States by MTI. A production vehicle was procured and modified to accept one of the engines for installation. Tests were conducted under various steady-state, transient, and environmental conditions to establish engine-related driveability, fuel economy, noise, emissions, and durability/reliability.

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

Task 5 - Mod II Engine - The Mod II (the second-generation ASE design) is based on the 1983 RESD, the development experience of the Mod I engine system, and technologies and components developed under Task 2. The goal of the Mod II is to demonstrate the overall ASE Program objectives in an engine/vehicle system. Although postponed in 1981, this task was reinstated during the first half of 1984 as the preliminary design of the Mod II engine system was activated. In the latter half of 1984, this has transitioned into a preliminary detail design of the Mod II.

Task 6 - Prototype Study - Postponed.

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

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

Task 9 - Program Management - Work under this task will provide total program control, administration, and management, including: reports, schedules, financial activities, test plans, meetings, reviews, and technology transfer. This task also provides for an extensive Quality Assurance Program.

Program Schedule

A current schedule of the major milestones and the Mod II Engine Development Program is presented in Figure 1-2. This schedule has been maintained since its inception in the fall of PY 1984 when the Mod II program was reactivated.

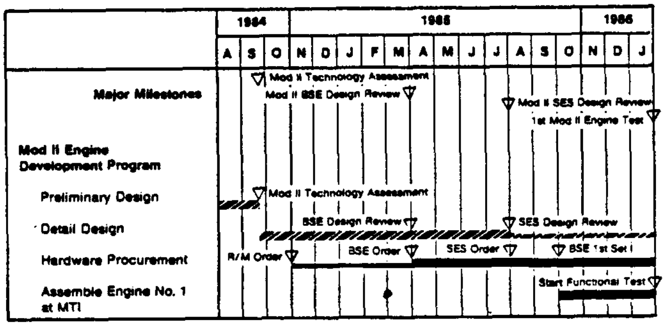

Fig. 1-2 Mod II Engine Development Program Schedule

Program Overview, Status and Plans

The pace of Mod II engine development continued to accelerate in early PY 1985 as the initial detail design was completed and the BSE and SES Design Reviews were successfully completed. Design review projections of Mod II engine performance indicate all of the program goals will be satisfactorily fulfilled. Mod I engine and component development are now focusing more specifically on Mod II design, analysis support, and substantiation.

A summary of the accomplishments achieved in the ASE Program during this report period are presented in the following sections.

RESD

The RESD was revised in May 1983, with a major departure from earlier versions of the engine concept. This revision was motivated primarily by the goal of achieving a manufacturing cost competitive with that of internal combustion (I.C.) engines, while still meeting other program objectives. As reported in 84ASE408SA6, an independent manufacturing cost study of the May 1983 RESD was completed during the first half of 1984 by Pioneer Engineering and Manufacturing Company. This RESD, to a substantial measure, is the foundation on which the Mod II engine design is based. Consequently, no further work has been accomplished on the RESD since early 1984, with program resources being reserved for execution of the Mod II design. The next concerted effort to update the RESD will be in 1986.

Component and Technology Development

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

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

EHS Development

The EHS design concepts for the Mod II were selected at a MTI/USAB/NASA task team meeting early in this report period. The choices were between a conical or Mod I Bill of Material (BOM) nozzle and an exhaust gas recirculation (EGR) or combustion gas recirculation (CGR) combustor. The BOM nozzle (see Figure 5-25) was chosen for the Mod II because of its superior performance and greater operating experience. Lower soot, reduced heater head temperature spread, and adaptability to a center igniter were three of its major advantages. However, it has not yet demonstrated its ability to operate without plugging at the very low fuel and atomizing airflows required for the Mod II. Work will continue on the BOM nozzle to improve its operational capabilities at these conditions. The conical nozzle is considered an alternate design that has demonstrated its nonplugging capability but will be developed further to attain its potential for the required lower soot levels and the reduced heater head temperature spread. After the selection of the nozzle concept, improvements were incorporated in the design to provide a longer life and increased reliability: centered ceramic igniter, streamlined flow paths, better seal joints, and improved tolerances.

The CGR combustor design (see Figure 3-26) was chosen over the EGR design for the Mod II because of higher EHS efficiency, smaller preheater size, lower NO_x emissions, compact packaging and simpler controls. The CGR concept has been under study and development since it was tried on the first Mod I engine and caused unacceptable deformations of the preheater due to thermal stresses. The new tubular-to-elliptical (TTE) design has 12 radial tubes with an ejector for each tube to thoroughly mix the combustion gases and air. Extensive steady state and transient testing on both rigs and Mod I engines was performed in order to demonstrate its readiness for use on the Mod II engine. Examination of test hardware indicated that the TTE design caused no damage to the preheater. Emissions levels, although higher in soot during transients than steady state, were acceptable. These tests also defined the Mod II ignition and start-up envelopes and optimal conditions so that control strategy could be identified.

During this report period rig tests were run with DF-2 diesel grade fuel using the BOM nozzle and a CGR combustor selected for the Mod II engine. These tests indicated that both CO and NO_x emissions were comparable to gasoline under steady-state operation.

The development of a low-cost preheater using a mixed-oxide ceramic matrix was continued with a follow-on contract to Coors Porcelain Company (Coors) to fabricate eight test blocks. Previous testing had indicated that the performance of this design was comparable to the metallic designs. The test blocks will be used to demonstrate the capability of the materials to resist cracking and leakage when exposed to thermal cycling.

Hot Engine System (HES) Development

Annular heater head development for the Mod II focused on demonstrating manufacturability, improving the design and substantiating heat transfer correlations. At this point a total of 31 prototype castings have been manufactured by Howmet and the identified problems were solved for the Mod II castings. Segregation of the XF-818 casting material in the neck region was eliminated by double gating the mold and adjusting mold preheat and metal temperature. Manifold core posts were eliminated from the design when they caused more casting problems than they solved. The design was also changed to increase the height and width of the tube deck to substitute a simpler machining approach for the difficult casting of the neck. These changes will make the Mod II design manufacturable. A new design for the heater head fins was evaluated on a heat transfer rig and the expected improvements were verified. The design makes use of a higher fin density with a novel fin design. An engine test of this design also verified the higher heat transfer coefficient for the rear row (see Figure 5-30).

Materials and Process Development

Materials development focused on selection of materials for the Mod II hardware, on proof-of-concept testing of the integral reduced height and weight piston base/rod/crosshead assembly and a new lightweight fiberglass reinforced plastic (FRP) hydrogen storage bottle. The piston base/rod joint has now withstood 0.47×10^6 cycles at 140% of maximum load fatigue testing which is slightly lower than the goal of 1×10^6 cycles at 160% of maximum load. Design improvements will be incorporated and tested in the next Semiannual period. Favorable results were obtained for the hydrogen storage bottle and it was selected for the Mod II design.

Main Seal and Piston Ring Development

Endurance and life testing of main seals is proceeding on engines No. 6 and 7 using the accelerated duty cycle described in previous reports (see Figure 3-42). This test cycle provides a higher proportion of high speed, high pressure operation than found in normal driving and therefore accelerates seal failure. In the interest of improving the statistical value of the data, the criteria for oil leakage failures was quantified to be less than 0.02 g/hr per seal. Rulon J* Pumping Leningrader (PL) seals were subjected to a 1000 hr endurance test on engine No. 7 with mixed results. Cycles 1 and 2 exceeded the leakage criteria and were removed at 742 hrs without any clear reason for the reduced life. In engine No. 6 HABIA* PL seals in cycles 3 and 4 have exceeded 1176 endurance hours, while cycles 1 and 2 seals have exceeded the leakage criteria and were replaced. Tolerances for critical parts are being reviewed for their effects on piston rod motion. Because of its demonstrated life the HABIA PL seal was selected for the Mod II engines.

Single-solid piston rings (see Figure 3-43) which have advantages in lower parasitic losses and simplified piston dome pressurization are now under endurance test in engine No. 7. Since there have been 1000 hrs accumulated on the accelerated duty cycle without failure the rings will remain in the engine for an additional 1000 hrs. Up to a 2 kW improve-

*Rulon J comes from the Dixon Corporation of Bristol, R.I.; HABIA is taken from a Swedish manufacturer named Habia, although the powder is from DuPont in the United States.

ment in maximum engine power is projected for this design (see Figure 3-44). The potential susceptability of this design to excessive leakage at cold start conditions remains to be investigated.

Control Systems Development

The impact on the Mod II engine design of the SES components (controls and auxiliaries) has increased as the engine optimization has reduced the size of the improvements available from the Stirling cycle portion of the engine. Advanced systems strategy and hardware are responsible for a significant portion of the vehicle mileage improvement. Major new technologies include a pulse width modulated (PWM) digital air/fuel control (DAFC), multiple volume hydrogen compressor, two tank storage system, electrically-driven blower, and a new high efficiency blower.

The interactions of the three major Mod II control systems are shown in Figure 1-3: power loop, fuel loop, and combustion air loop. These complex closed loop controls have required intensive systems and simulation development that have resulted in major technological advances.

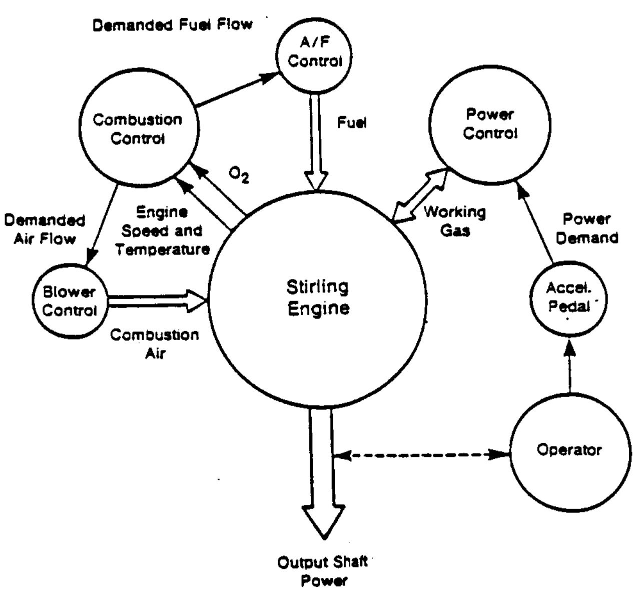

Figure 1-3 Mod II Control System

Goals for the Mod II SES, developed at the Technology Assessment in September 1984, are indicated in Table 1-1. All of the goals can be accomplished or are within reach through implementation in the Mod II design, and most have been demonstrated in component development hardware. However, the overall system performance of the mean pressure control (MPC) remains to be demonstrated in the Spirit as Mod II hardware becomes available. The weight goal was missed by a few kg, but only as a result of incorporating a more complex MPC system than was envisaged at the time the goals were set. Further design improvements are expected to reduce the weight below the goal level. An indication of the progress is that the Mod II design SES is 34% lighter than the previous Upgraded Mod I design.

TABLE 1-1

SES GOALS

ASE Engine	
3	Mod I (USAB) EHS Development
5	Upgraded Mod I (MTI) General Dev.
6	Upgraded Mod I (USAB) Endurance Testing
7	Mod I (USAB) Seals/Ring Dev.
8	Upgraded Mod I (MTI) Transient Vehicle Performance

Mod I Engine Test Program

At the end of this report period there are five active engines in the ASE Program, two in the original Mod I configuration located at USAB and three in the Upgraded Mod I configuration located at MTI and USAB. Table 1-2 summarizes the engines, their purpose and locations. Figures 1-4 and 1-5 indicates the test hours on each engine and the total accumulated test hours, respectively. An increased testing rate was accomplished in this report period to bring the total hours accumulated to 11,412.

Mod I engine No. 3 has as its principal use the development of the EHS. The substantiation tests for the selection of

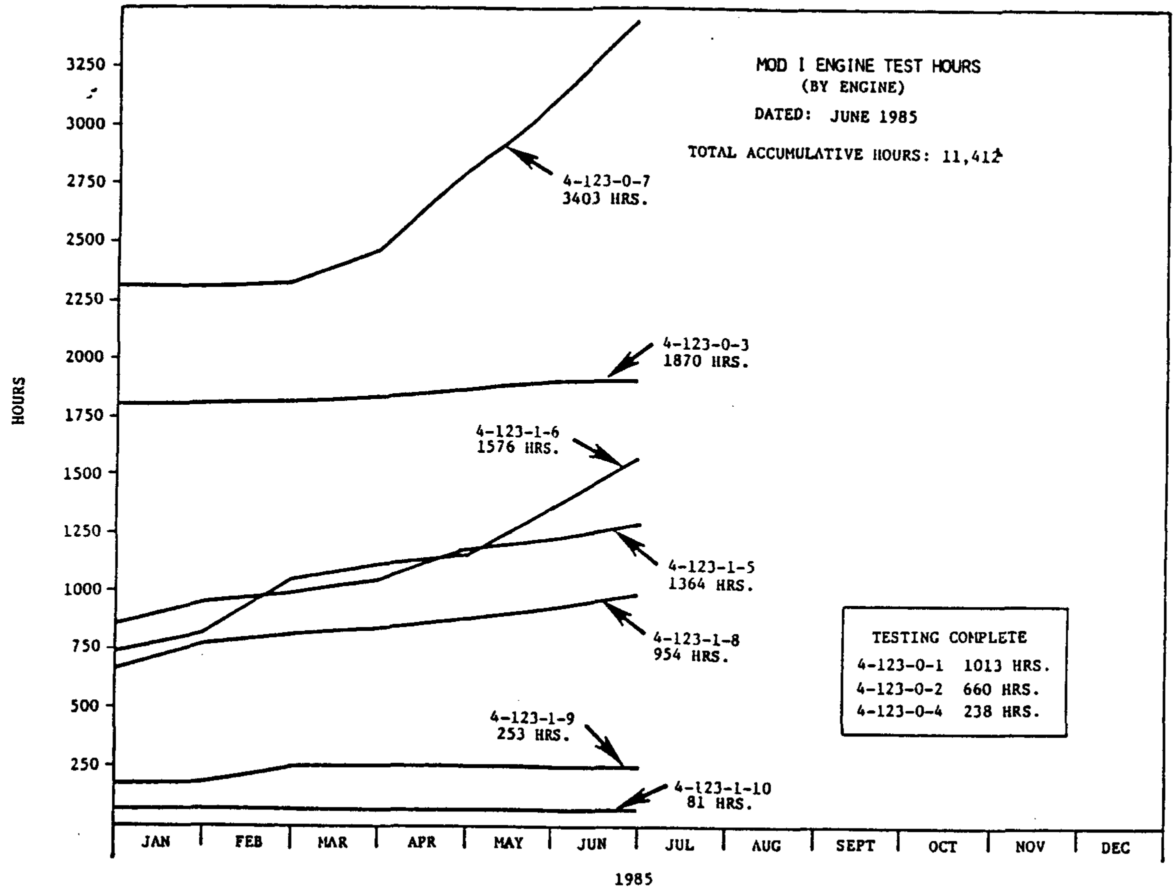

Figure 1-4 Mod I Engine Test Hours

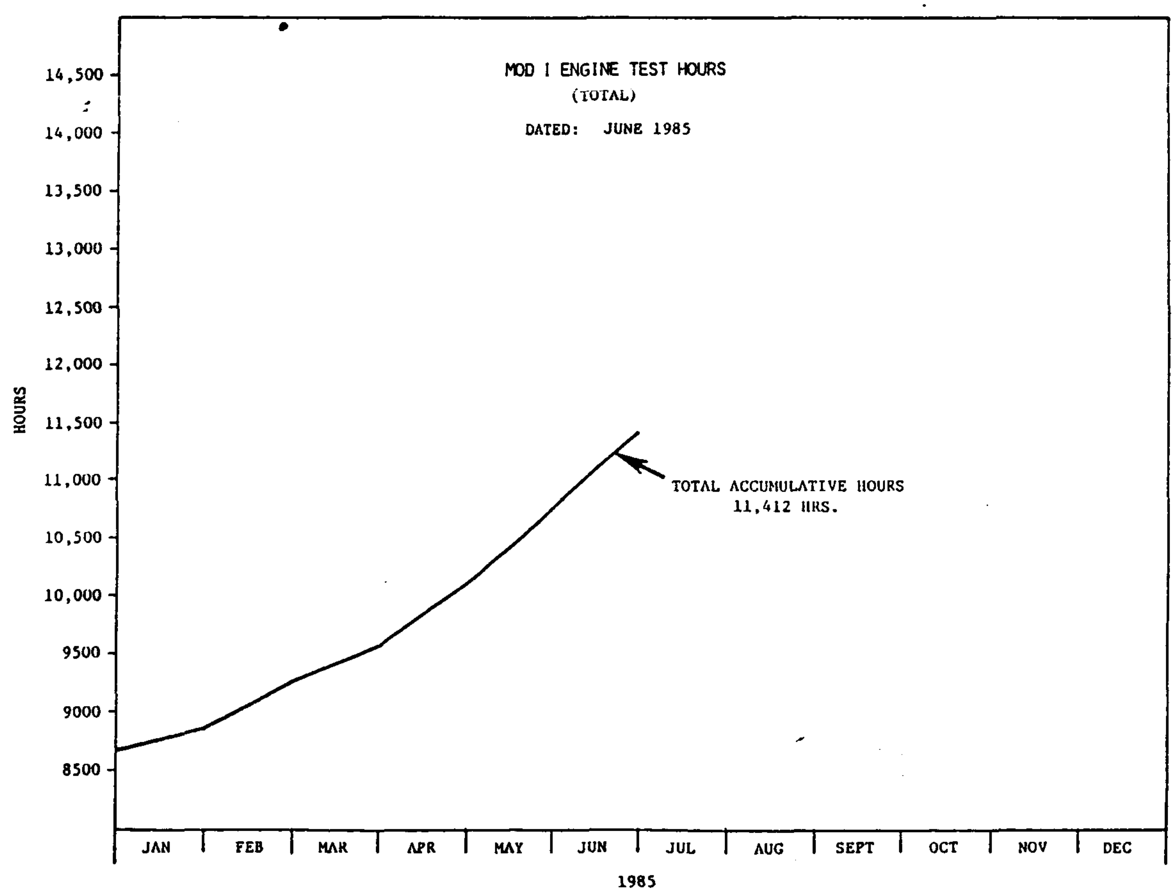

Figure 1-5 Mod I Engine Test Hours

the CGR system along with fuel nozzle and swirlers was performed on this engine.

TABLE 1-2

MOD I ENGINE UTILIZATION

- Idle Fuel Flow Capability of 0.15 g/s
- A/F Rate Control Accuracy of 5%
- Heater Head Temperature Maintained within 10 C
- Idle Speed and Pressure Reduced to 400 rpm at 2 MPa
- Elimination of the Oil Servo System
- Improved Reliability
- Total Weight of 82.0 kg (180.8 lb)

Mod I engine No. 5 located at MTI has been used as the work hours for general engine development; even so it has accumulated 514 hrs. Tests included: an evaluation of single-solid piston rings, baseline data on the Mod II heater head tube and fin heat tansfer, comparative CGR emissions, back-to-back tests of ethylene glycol and water coolants, and definition data for the Mod II appendix-gap code validation.

Upgraded Mod I engine No. 6 has been used for the high temperature (820°C) evaluation of the Mod II heater head brazing technique. By the end of June, 960 hrs of heater endurance had been accomplished without failure. Based on the success to date this test will be extended to 2000 hrs. Engine No. 7 has accumulated over 1100 hrs while being used exclusively for development of the single-solid piston ring and main seals.

The Spirit vehicle has Upgraded Mod I engine No. 8 installed. As described in the previous Semiannual report this engine and vehicle was successfully tested at GM as part of the ITEP Program. The engine accumulated 250 hrs in this period while performing tests of the DAFC, low speed and pressure idle, comparison of 720° and 820°C vehicle operation and the evaluation of hot shutdown techniques for reduction of the cold start penalty (CSP) in CVS tests. The progress of Spirit vehicle testing is indicated in Figure 1-6 where the relative fuel economy performance is shown to increase from -13% in 1984 to +21% in June of 1985.

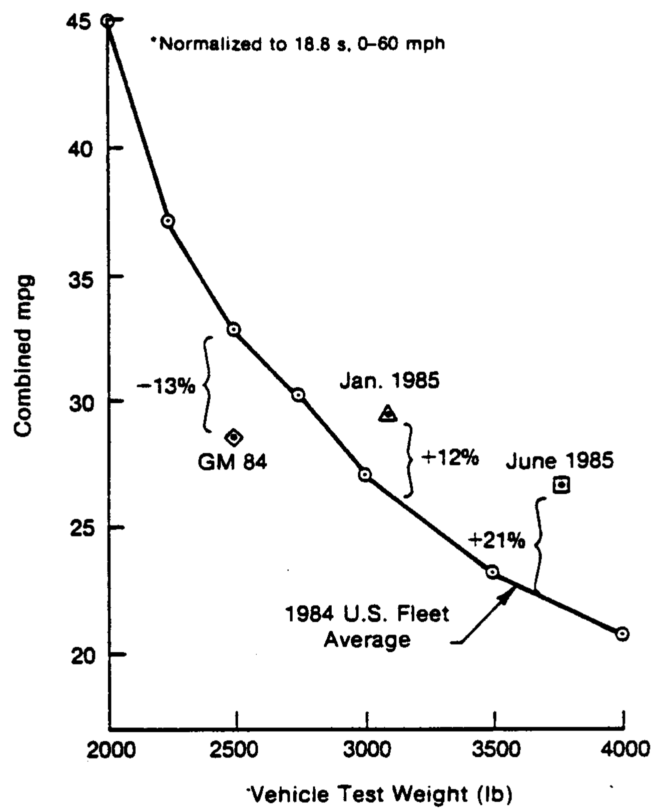

Figure 1-6 Fuel Economy Upgraded Mod I Spirit History

Mod II Engine Development

The Mod II engine is the second and final experimental engine design in the ASE Program. It will be used to demonstrate accomplishment of the ASE Program goals via EPA specified fuel economy and emissions tests by September 1987. The development is being accomplished over a four year period progressing through an RESD, design and analysis, two formal design reviews, hardware procurement, engine assembly, test and development followed by vehicle system assembly, test and development. This report covers the Mod II development period through the SES design review. Although this interval exceeds the normal Semiannual period by two months it was decided that any delay in the issuance of the Semiannual report would be compensated by having a complete self contained report on the Mod II design process. The schedule shown in Figure 1-2 depicts the expected progress of Mod II development.

Historically, engine development has focused on a higher efficiency, lower cost and weight design. The Mod I was the first experimental engine in the ASE Program designed for application in a rear-wheel-drive vehicle. Its development was based on the RESD and prior non-ASE technology, utilizing a four-cylinder, double crankshaft, building block design. This hardware design was followed by the Upgraded Mod I, which used the same basic configuration, but also incorporated nonstrategic materials, operated at 820°C instead of 720°C, and was part-power optimized. These and other improvements resulted in additional power and reduced weight.

In May 1983, a new design concept was selected for the RESD that was to become the basis for the Mod II. The selected design utilized a V-block with a single crankshaft and annular heater head. This concept was selected over several other alternatives because of manufacturing ease, lower weight and cost, and higher power-to-weight ratio. Other concepts which were not as fully developed or had high risk were eliminated, since the Mod II engine design will use only those technologies that can be implemented in hardware by 1987.

The Mod II will be packaged in a 1985 front-wheel-drive Chevrolet Celebrity for the ASE Program demonstration in 1987. This vehicle was chosen as representative of a high sales volume, high mileage vehicle of the appropriate test weight and compartment size for the Mod II engine. Currently, the Celebrity has the highest mileage of any 3125# inertia test weight vehicle by a 20% margin.

The design process has followed an overall logic that has maximized the use of available time and resources. The RESD concepts, V-block, annular heater head, low weight, low cost, and ease of manufacturability were used as the foundation for the Mod II design. The technology derived from the Mod I and component development was then added to provide a preliminary design. Subsequently, this engine design was optimized in each of three power levels, which bracketed the expected power level (i.e., 55, 60, and 65 kW). Finally, the performance for each engine was utilized in the vehicle simulation code to define the optimum drive train and engine size. On the basis of this approach, a 60 KW engine was selected to meet the overall program performance objectives.

In addition to the Mod I engine development experience, vehicle testing, ITEP, and component development have had major impacts on the Mod II design. Vehicle test results provided the confirmation for mpg projections, constant volume sample (CVS) cycle CSP, hot start improvement, and low-speed idle improvements. ITEP testing at GM Research Laboratory pointed to high idle fuel flow as a lucrative area for improving fuel economy, as well as providing confidence in the vehicle performance and cooling system. ITEP involvement by John Deere and Company (Deere) provided assistance in the design of the V-block casting for lower weight and ease of manufacture. Component development tests and anaytical results were critical in the selection of subsystem components (i.e., CGR combustor, heater head, fins, tubes, controls and auxiliaries, etc.).

The design of the BSE was presented at a formal design review at NASA/Lewis Research Center on April 2-3, 1985. It was concluded at that design review that the Mod II design was technically sound, its preliminary performance projections were consistent with the program goals and that it was at an acceptable risk level. This design was then approved by NASA and initial BSE procurement was authorized.

The final Mod II engine cross section is depicted in Figure 1-1. The delineation and function of major BSE subsystems is the same as that used in previous engine designs. In other words, the EHS converts fuel flow to heat flux; HES contains the hot working gas and converts heat flux to a pressure wave acting on the piston; the cold engine system (CES) contains the cold working gas, seals, and transfers piston motion to a connecting rod; and the engine drive system (EDS) converts the reciprocating rod motion to rotary motion.

The major SES control systems and auxiliaries consist of the combustion control system that includes air/fuel (A/F) ratio regulation, atomizing air supply, and heater head temperature control; the combustion air supply system that includes an alternator, blower motor, and blower; the mean pressure control (MPC) system that includes working gas storage, compressor, control valve, and check valves; and the digital engine control (DEC) that contains the logic required to operate the BSE and SES.

Subsequent to the BSE Design Review, there were several updates and improvements to the BSE that have altered its design and performance. A major change occurred in the HES design when the heater head was reoptimized, utilizing an upgraded analysis for the appendix gap losses, and for a single manifold with the front row heater tubes attached to the expansion space. Figure 1-7 compares the single and two manifold heater head designs. This design change resulted in lower front row tube temperatures and increased cycle efficiency. Additionally, the EHS components were modified for ease of manufacturing. As a result of these and other modifications, the BSE weight was reduced by over 11 kg.

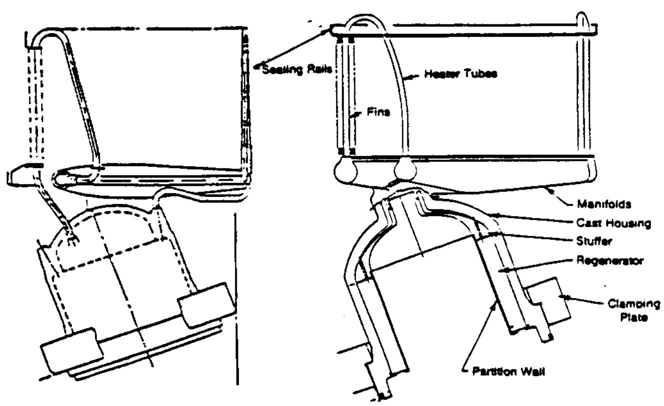

Figure 1-7 Comparison of Double and Single Manifold Heater Head Designs

The design of the SES was presented at a formal Design Review at NASA/Lewis Research Center on August 7-8, 1985. At this review, the final design Mod II engine performance projections were presented and determined to exceed the program goals. Also, the design was acknowledged to have adequate mechanical integrity and be achieveable at an acceptable risk level. The final design engine map is shown in Figure 1-8. While this engine map may not seem to be different from earlier Mod I designs, there are two key differences. First, the idle operation point has been reduced to 400 rpm and 2 MPa to both reduce parasitic losses and improve fuel economy during the large portion of the CVS cycle when idling occurs. Second, the maximum efficiency point (38.5%) occurs at a lower rpm which matches more closely with the Average Operating Point (AOP) of the CVS cycle. Both of these improvements increased combined fuel economy although they increased the demands on such system components as the A/F control and the MPC.

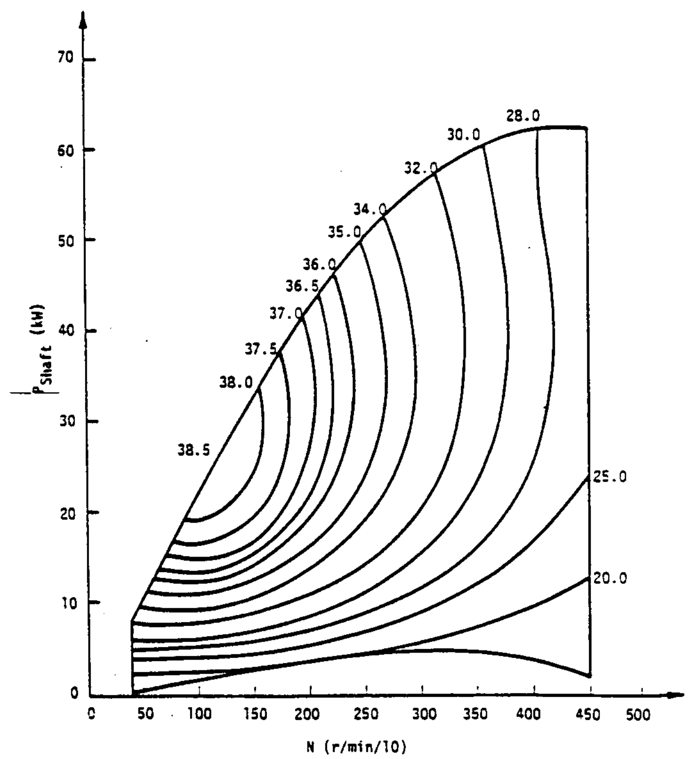

Figure 1-8 Mod II SES Performance Prediction

As can be seen in Table 1-3, the Mod II has met both its power and weight goals. The power density has shown steady improvement (over a 45% improvement) since the Mod I engine was built and is approaching the RESD value.

TABLE 1-3

MOD II POWER/WEIGHT SUMMARY

	Mod I	Upgraded Mod I	RESD	Mod II Goal	Mod II Actual
Weight, kg (lb)	312 (688)	312 (688)	173 (382)	200.9 (442.9)	199.3 (439.4)
Power kW (hp)	53.8 (72.1)	63.5 (85.1)	60 (80.5)	60 (80.5)	62.3 (83.6)
Weight/Power, kg/kW (lb/hp)	5.80 (9.54)	4.9 (8.1)	2.91 (4.78)	3.35 (5.50)	3.20 (5.26)

Mod II vehicle performance shown in Table 1-4 indicates the steady vehicle mileage improvement as well. The projected mileage is 40.9 mpg or 32% over that of the IC-powered Celebrity. Potential improvements to the mileage that were considered to be of excessive risk for inclusion in the predictions could increase the mileage to 42.7 mpg. The resulting ASE Program development progress is shown in Figure 1-9. The progress is dramatic especially in the case of fuel economy when it is compared to the U.S. average at the same vehicle weight.

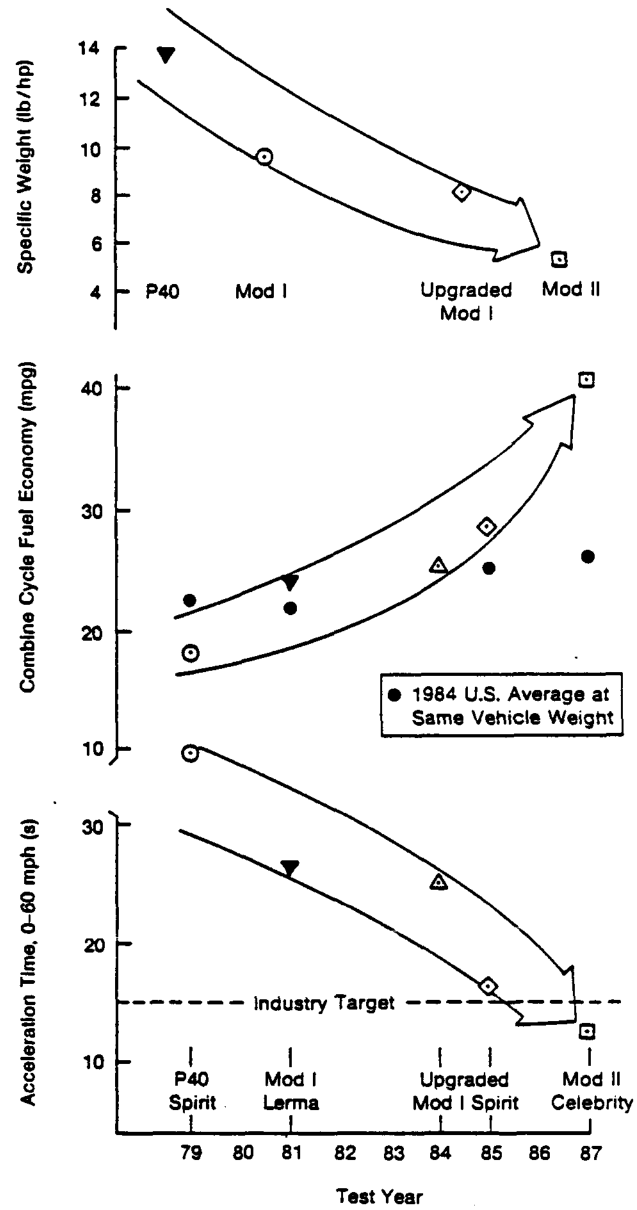

Figure 1-9 ASE Program Development Progress

TABLE 1-4

MOD II VEHICLE PERFORMANCE SUMMARY

Mileage	SI	Mod I	Upgraded Mod I	RESD	Mod II	% Improvement (SI-Mod II)
Vehicle	Celebrity	Lerma	Spirit	Phoenix	Celebrity	
Inertia Weight, kg (lb)	1360.1 (300)	1700.1 (3750)	1473.9 (3250)	1427.2 (3125)	1427.2 (3125)	
EPA Combined Cycle, mpg	31.0	23.5	28.8	40.1	40.9	32
Acceleration, sec	15.0	26.1	16.2	15.0	12.4	
0-60 mph	-	-	-	9.7	8.6	
50-70 mph	-	-	-	4.5	4.1	
0-100 ft						

Shown in Figure 1-10 is a comparison of the projected Mod II engine performance and the U.S. fleet average mileage. Since the IC Celebrity has 20% better fuel economy than the fleet average, the 32% improvement of the Stirling Celebrity over the IC Celebrity is indicative of a 50% or better improvement over the U.S. fleet average.

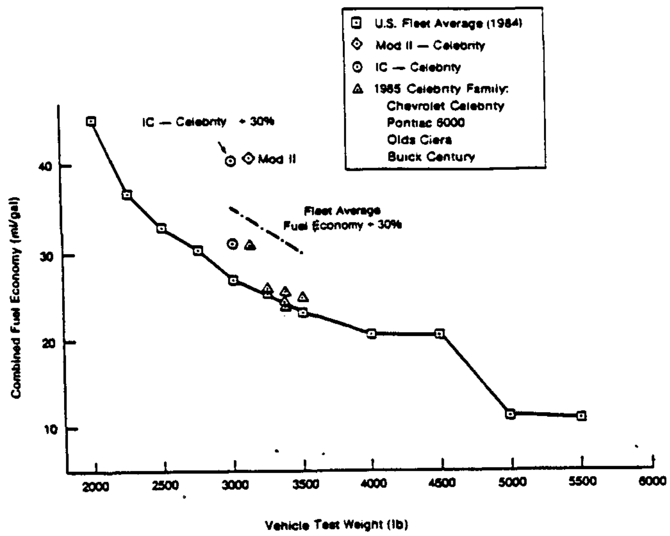

Figure 1-10 Comparison of Spark Ignition and Mod II Fuel Economies

The Mod II engine will have three distinct hardware configurations throughout its development as the technology and modeling mature. The first build will be as a BSE for test cell development at MTI and USAB. The vehicle build will consist of the first build hardware, but will be configured as an SES utilizing the cast-iron block if it is available. The first build is already being procured for a January 1986 engine start-up.

The final hardware build, which will be used to demonstrate the program goals, will also be an SES configuration, but will have the single manifold heater head design. It will also contain all other BSE and SES improvements ready for engine test.

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II. REFERENCE ENGINE AND SYSTEM DESIGN DEVELOPMENT

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

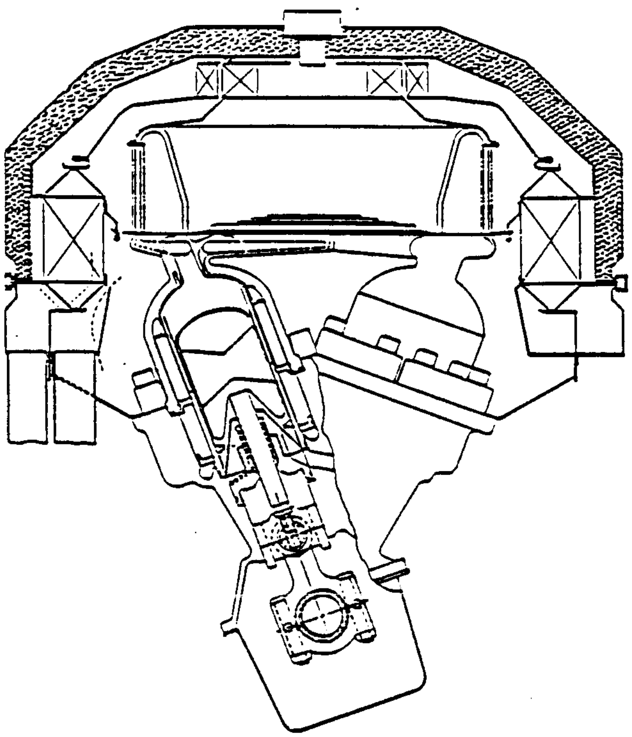

Figure 2-1 V-4 Mark V RESD

During the first half of 1985, all available program resources were devoted to the Mod II engine design effort. Therefore, no further work was done on the RESD during this report period. The next concerted effort to update the RESD will occur in 1986.

III. COMPONENT AND TECHNOLOGY DEVELOPMENT

Introduction

Component and technology development is directed towards advancing Stirling engine technology in the areas of combustion, heat exchangers materials, seals, EDS and auxiliaries. Durability/reliability, performance, cost, and manufacturability are all critical considerations in the development process as defined and guided by the RESD task.

The concentrated development of components for the Mod II engine continues during this report period. A portion of the effort directly supported the design of the Mod II engine and the final BSE and SES design reports with both design and analysis and component subsystem testing. The remainder of the effort concentrated on development components identified either as initial Mod II or RESD.

EHS

The primary goal of the EHS is low emissions while maintaining high efficiency for a 35:1 fuel turndown ratio in a minimum volume.

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

Activities during the first half of 1985 were concentrated on the Mod II where CGR was selected over EGR. Specifically, the 12-tube TTE combustor and BOM internally air atomized nozzle with integral center igniter were chosen and Mod II drawings completed. Performance and durability of both combustor and nozzle were addressed during rig, engine, and vehicle testing. The igniter and combustor were evaluated for improved durability. Emissions, soot, and reduced atomizing airflow performance were addressed as well. A major effort was conducted to identify and correct nozzle-to-nozzle atomizing airflow variations. RESD activities were conical nozzle and ceramic preheater development. An effort was begun to develop a system optimized for diesel fuel.

Fuel Nozzle/Igniter Development

The objectives of nozzle/igniter development are to reduce atomizing airflow to 0.4 g/s (Mod II goal), eliminate plugging, extend igniter life and manufacture nozzles which will limit flow variations so that repeatable emissions and heater head ΔTs may be obtained.

Previous work has led to two development paths. The BOM nozzle, used in the Upgraded Mod I (Figure 3-1)*, has a relatively high atomizing airflow ($\sim$1.0 g/s) and is prone to plugging and cracking of the ceramic portion of the igniter. Performance was acceptable with this nozzle, when clean, however, it was suspected that variations in atomizing airflow among nozzles was affecting the repeatability of data.

The conical nozzle, representing the second path, has 0.4 g/s atomizing airflow (an efficiency benefit), is nonplugging, but tends to produce more soot (preheater plugging) and large heater head temperature variations (an efficiency and durability detriment). During this report period, the BOM nozzle was selected for the Mod II based on superior performance, while development of the conical nozzle was continued as an alternate.

*Figures can be found at the end of this section beginning on page 3-19.

The development of the BOM nozzle included cold-flow tests to determine atomizing airflow and nozzle-to-nozzle repeatability, extending igniter life, and formulating plans to reduce atomizing airflow. The conical nozzle development concentrated on improving spray stability and droplet momentum in order to decrease soot and heater head temperature variation to BOM levels.

Initial Mod II development began with the determination of the atomizing airflow characteristics of existing BOM nozzles (Figure 3-2). The wide flow variation, present at all atomizing airflows tested (0.3-1.0 bar), is unacceptable for both engine and combustor performance. The reasons for the variation were tolerances that were too large resulting in ±16% geometric flow area variation, ambiguous swirler dimensioning (Figure 3-3), and leakage caused by press fit construction. The hot to cold difference of the current BOM nozzle (Figure 3-2) indicates that atomizing airflow in an engine environment is substantially lower* than previously assumed and, hence, a lower performance penalty due to cold atomizing air will be incurred with the initial Mod II build. Efforts to reduce atomizing airflow will involve decreasing the number of holes.

BOM nozzle plugging was evaluated during steady-state (engine No. 5) and transient (engine No. 8, Spirit vehicle) conditions. It was concluded that there is a strong correlation with the amount of time at idle. The severity of the effect will be determined by instrumenting a nozzle with thermocouples (Figure 3-4) and testing in the performance rig. Options which are being considered include: water cooling, air cooling, and thermal barrier coating of the nozzle tip.

An extended life center igniter was designed and evaluated during this period (Figure 3-1). The original Mod I pronged igniter (Figure 3-3) failed frequently in only a few hours when soot formed on the ground electrodes and the ceramic sleeve cracked leading to internal shorting. In addition, there is evidence that the nozzle orifices nearest the two prongs plugged more readily than those further away. The center igniter uses a thicker wall, high temperature ceramic (Omegalite 450) and a concentric ground electrode. A total of 125 hrs and 395 starts have been achieved without failure.

The Mod II nozzle/igniter design (Figure 3-5), completed during this report period, addresses the short comings of earlier BOM designs. Specifically the mixed atomizing A/F flow path was streamlined to eliminate dead areas where carbon can form and cause orifices to plug, the igniter sleeve to nozzle tip joint was tapered to eliminate leaks, a high tempertaure ceramic center igniter with proven durability was used, helical, as opposed to straight, swirl slots were selected to reduce flow variation among nozzles and dimensional tolerances were tightened to reduce flow variation to ±5%.

Development of the nonplugging conical nozzles continued both as an alternate for Mod II and for alternate fuels. During the current report period efforts were concentrated on improving heater tube temperature variation and reducing soot. Atomizing air swirler geometry was changed from helical to straight slots leading to increased flow stability**, durability during assembly and ease of manufacturing. Following this a high pressure ratio atomizing air sleeve, using straight slots, was designed and fabricated. The purpose of the higher pressure (Figure 3-6) was to increase

* Heat conduction into the nozzle results in an increase in atomizing air temperature causing flow to decrease at constant pressure.

**Helical slots led to a bi-stable spray pattern which adversely affected soot emissions.

fuel droplet momentum to levels closer to the BOM nozzle and, hence, reduce soot levels with the CGR combustor. The effect of the straight slot high pressure ratio modifications on soot was not as anticipated (Figure 3-7) indicating the need for additional development. Both nozzle and CGR combustor modifications were evaluated in the spray rig. The latter includes an external turbulator and internal guide vanes (Figure 3-8). Free-burning and performance rig testing of these concepts is planned during the remainder of 1985.

CGR Combustor System Performance

The objective of combustor system performance is to meet the program emissions goals (g/mi):

NO_X:	0.4
CO:	3.4
HC:	0.41
Particulates:	0.2

with soot-free combustion (Bacharach $\leq$ 2), low heater head temperature variation (ΔT1 and ΔT2 $\leq$ 50°C) over a 35:1 fuel turndown ratio. In order to meet the NO_X requirement, recirculation of combustion products has been utilized (EGR and CGR).

During this report period CGR was selected for Mod II and a specific design was identified. In recognition of the possibility of performance degradation with time when integrated with the control system in an engine environment. An extensive transient and steady-state test program of the selected CGR combustor was conducted using rigs, engines, and the vehicle. This approach also served to establish the performance variation among combustors and engines in order to establish the margins needed to meet program goals. Specific tests were conducted to optimize Mod II CGR idle and start-up conditions and establish the feasibility of using a lambda sensor with the DAFC.

CGR was chosen over EGR for the Mod II design because of higher EHS efficiency, smaller preheater size, lower NO_X emissions, more compact packaging, and simpler controls.

The 12-tube TTE CGR combustor (Figure 3-9) was selected based on superior emissions (Figure 3-10) and heater head temperature profile, (Figure 3-11), relative to EGR and all other CGR combustors*. Extensive steady-state and transient tests were conducted to determine durability and performance of the TTE combustor with BOM nozzle. Based on these results, design changes were identified. Two versions of the combustor (TTE 1 and TTE 2) were tested in Upgraded Mod I engines No. 5 and 6 to determine relative performance. Although these combustors differ in the inner swirl radius (Ri) (Figure 3-12), they give similar performance. Engine-to-engine comparison of emissions (Figures 3-13 through 3-15) reveals equivalent soot and CO, although not NO_X. Some of the variation in NO_X emissions may be due to the effect of lambda (Figure 3-16) or alignment of the ejectors. Based on an average of the data (Figure 3-15) CVS cycle NO_X emissions may exceed the program goal of 0.4 g/mi at certain fuel flows. A definite statement cannot be made due to uncertainties in the prediction technique** and the effect of higher Mod II heater head ΔP, compared to Mod I. Reductions in NO emissions, if needed, will be accomplished by optimizing mixing tube diameter or substitution of exhaust gas for atomizing air as the result of development tests during the remainder of 1985. As part of a Mod II CGR evalu-

* Ribbon and 42-tube tubular CGR combustors, at their current development state, exceed the Mod II goals for soot.

**The prediction code, completed during this period, integrated steady-state engine data over the Mod II vehicle urban CVS cycle. The Mod II cycle is defined in terms of time at each fuel flow.

ation, a Mod I version of the TTE combustor was tested at 720°C in engine No. 3 and the performance rig. NO_x emissions were lower in the rig (Figure 3-17) as expected due to the unidirectional flow in the rig heater head causing lower front row tube temperatures.

Extensive Spirit vehicle tests of the CGR/BOM nozzle system were conducted over the road and on the chassis dynamometer at the New York State Environmental Conversation Department test facility in Albany. The goal was to determine the impact of transients and system interaction (A/F control, atomizing air, temperature control) on soot and CO emissions and to minimize ignition delay (HC emissions). Transient response and soot were evaluated during acceleration, deceleration, and the CVS hot 505 cycles. In the former case there was virtually no difference between flow changes at the blower discharge and ejector entrance or apparent time lag between air and recirculated flow. Transient soot was higher than obtained during test cell tests on engine No. 8 (Bacharach No. < 1), although acceptable (Bacharach No. < 2 is the Mod II goal) if atomizing air pressure is equal to or greater than 0.5 bar. Soot is of concern because of preheater plugging.

The discrepancy in the soot data between vehicle and engine is believed to be due to the following characteristics of BSE No. 6 engine.

- Lack of idle running, i.e., less nozzle plugging
- Decoupling of the air blower from the engine, i.e., more uniform lambda
- Use of a shop air supply for atomizing air, i.e., constant airflow.

The comparative data illustrates the system (vehicle) interactions which can degrade performance. In addition to the previous effects, soot is influenced by nozzle fuel leakage caused by poor sealing and reduced atomizing airflow. Thus, differences in the condition or flow characteristics of BOM nozzles (Figure 3-2) will cause variations in soot levels illustrating the need for the improved quality control of the Mod II nozzle.

Idle emissions and igniter tests were run to establish Mod II ignition and start-up conditions. CO (Figure 3-18) and soot emissions determined a minimum idle atomizing air pressure of 0.5 bar and lambda of 1.35. Using vehicle and performance rig data, and assuming a Mod II atomizing airflow characteristic represented by the average of the data shown in Figure 3-2, an operating envelope was determined (Figure 3-19). Optimal ignition conditions were also defined (Figure 3-20).

RESD CGR combustor activities other than direct Mod II support consisted of performance rig testing of the 42-tube tubular and 48 guide-vane combustors. CGR levels were too low to meet NO_x emissions requirements while soot was unacceptably high. Analysis indicated that the low CGR levels were due to inadequate mixed flow cross sectional area (Figures 3-21 and 3-22). Good correlation between hot and cold flow data is evident and forms the basis of the CGR prediction code completed during the report period.

In support of DAFC development, a Bosch lambda sensor was tested in the performance rig. The sensor was mounted in the exhaust (Figure 3-23) and is maintained at a minimum temperature of 300°C by internal heaters. A strong dependence of sensor output on flow rate was found at 720° and 820°C (Figure 3-24) heater head set temperatures. This characteristic is unacceptable due to the wide variation of exhaust flow rates in the engine.

<u>CGR Combustor Durability</u>

The objective of combustor durability efforts is to demonstrate a life of 3500 hrs for 10,000 starts. Experience with earlier CGR and EGR combustors indicates life far less than this goal, several hundred hours being typical. In order to improve the life of the TTE CGR combustor, a deep-drawn combustor shell was fabricated and tested, oxidation-resistant coatings were applied to an EGR combustor for evaluation and alternate materials considered for evaluation. The first activity addresses warpage and cracking while the last two address high temperature oxidation. Results of the durability evaluation were incorporated into the Mod II design.

CGR combustor durability was assessed under steady-state (engine No. 5, engine No. 3 and combustor performance rig) and transient (engine No. 6, Spirit vehicle) conditions. Earlier Upgraded Mod I versions of the TTE combustor featured welded construction (flanged ejector ring to shell) and tended to crack along the weld as well as warp the shell. Attempts to weld repair the combustor in engine No. 5 were unsuccessful in that NO_x emissions were found to increase to an unacceptable level. A different result was found with a similar combustor in engine No. 6 where no degradation in performance occurred after 250 hrs of transient duty cycle, in spite of severe cracks and warpage. Nevertheless, continued cracking will eventually result in loss of CGR (high NO_x and soot) and is unacceptable.

Cracking was addressed by utilizing a single deep drawn piece to eliminate the weld joint. This change appears to be successful based on limited experience. The warpage phenomenon is being addressed by design and material changes. In the former case, one possibility consists of a shell using textured (diamond pattern) stainless steel. Material changes being considered consist of high temperature Cabot 214 and oxidition-resistant coatings.

Attempts to fabricate an Upgraded Mod I deep drawn combustor shell from Cabot 214 have, thus far, been unsuccessful due to material tearing. Oxidation-resistant coatings have been evaluated on an EGR combustor (Figure 3-25). Only the plasma-sprayed coatings (Amdry 962 and 995) survived the initial 4-5 hrs of engine testing. These coatings have attained a total of 121 hrs of operation without evidence of degradation. Testing will continue based on engine availability.

The Mod II CGR combustor (Figure 3-26) incorporates several design features to improve durability, i.e., reduce cracking and warping. These include a deep-drawn shell, rib stiffeners, allowance for the combustor to grow axially via bellows on the fuel nozzle, textured 310 stainless steel material, the use of folded tabs in place of weld joints where possible and conical as opposed to a flat center section. Application of oxidation resistant coatings will remain as a back-up since it adversely affects CSP.

Durability improvement will continue through 1985. Specific activities consist of measuring Upgraded Mod I CGR combustor surface temperature to use in verifying stress levels of the Mod II design, deep drawing of combustor shells from Mod II textured stainless steel and Cabot 214, and oxidation-resistant coating and testing of an Upgraded Mod I CGR combustor.

Alternate Fuels Development

One of the goals of the program is multifuel capability with the proviso that satisfactory performance must first be obtained with unleaded gasoline. The long term objecting of this task is to ascertion the effect of alternate* fuels on EHS performance and durability goals and implement design changes, if needed, to accommodate them. During this report period the most difficult of the alternate fuels, diesel grade DF-2, was tested. The assessment of the degree of difficulty is based on viscosity (atomization), aromatic content (soot), sulfur (preheater corrosion), and volatility (ignition and start-up) considerations.

*Gasohol, diesel (DF-2), kerosene, and home heating oil

Baseline diesel fuel tests, using a TTE CGR combustor and BOM nozzle, were run in the performance rig. CO and NO_x emissions were comparable to gasoline (Figures 3-27 and 3-28). Soot emissions were inconclusive due to a suspected fuel nozzle leak.

Design of several diesel fuel CGR combustors was completed. The modified TTE combustor features external air swirlers and/or internal guide vanes to enhance atomiziation and mixing, respectively. The designs were based on results of the conical nozzle spray tests using plastic models (Figure 3-8). Free-burning and performance rig development will be conducted during the remainder of 1985. A flammability code was written to aid in the design of combustors for alternate fuels.

Ceramic Preheater Development

A follow-on effort was initiated with Coors, to fabricate eight mixed-oxide preheater matrix blocks. The object was to demonstrate the capability of producing several blocks which meet the Mod II leakage requirement. Tooling modifications were completed and the preheater rig refurbished in preparation for steady-state and thermal transient evaluations. Fabrication was begun with delivery of the first test section scheduled for July.

The susceptability of mixed oxide to sulfuric acid corrosion was determined by immersing samples in varying % molal solutions (Figure 3-29). Analysis will continue in the latter part of 1985 to determine representative solution strengths in the preheater passages.

Hot Engine System Development

The primary goal of this task is to develop a manufacturable annular heater head for Mod II that performs with high efficiency and to develop heat transfer correlations to predict performance. Another goal is to evaluate a ceramic partition wall in an annular P-40 engine.

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

Efforts during the first half of 1985 centered on Mod II prototype heater head manufacture, engine evaluation of hybrid Mod I/Mod II heater head performance, and basic heat transfer studies using the double mantle rig (DMR).

Mod II Heater Head Development

Prototype heater head (MTI drawings No. 1013D001 and 1013D002) development continued throughout this report period. Following the identification of tooling changes required for the Mod II head (MTI drawings No. 1019D001 and 1019D002), it was decided not to do a trial brazing of the 1013 heads since the timing would interfere with the actual Mod II heater head build which utilizes the 1019 design. This task is considered complete with the machining of fatigue test specimens.

A total of 31 prototype heater head castings have been made by Howmet. An initial problem of metal segregation was solved by double gating and adjustments to the mold preheat and metal temperatures. In the later case preheat and metal temperatures were first lowered, with little effect on metal segregation, and then increased so that preheat was higher and metal temperature equal to the initial levels. The final temperatures and the change from single to double gating (bell and manifold) was successful as demonstrated by sectioning of two castings.

Another prototype casting was sectioned through the manifold plane and end wall thicknesses greater than 3 mm confirmed. Two castings were also received for manifold fatigue tests, and machining was completed. Design of flow test fixtures for the Mod II heater head and regenerator were started.

Design changes were identified for the Mod II heater head. These were elimination of manifold core posts and increasing the height and width of the manifold tube deck compared to the prototype design. The first change is based on persistent breakage problems during casting while the second was made in recognition of the difficulty of casting the geometry of the deck. This region will now be machined.

Heater Head Heat Transfer

In order to determine the heat transfer from combustion gas to the heater tubes, specifically the impact of geometry changes from Mod I to Mod II, a Mod I/Mod II heater head and DMR test sections were fabricated. Testing was completed during this report period.

The Mod I/Mod II heater head was tested back-to-back with a Mod I head in an Upgraded Mod I engine. The Mod I/Mod II head consisted of 3.75-mm O.D. tubes (Mod II size at the time of fabrication) and Mod II fins, although configured to the Upgraded Mod I geometry by the cylinder and regenerator housing. Analysis of the results indicated a significant improvement in performance with the Mod II fin and that the front row heat transfer correlation of the Upgraded Mod I is valid for the reduced tube diameter and increased gap of the Mod I/Mod II head. Overall, comparing Mod I/Mod II to Mod I, the front row heat transfer is not as effective, Figure 3-30 (higher combustion gas temperature between rows in spite of lower tube temperature), however, this is more than compensated for by the rear row fins as indicated by lower combustion gas temperature after the heater head (Figure 3-31)*. The net result is a more effective Mod I/Mod II head in spite of the smaller size tube and larger gap of the front row causing a reduced front tube row temperature.

The first-order code was used with the two-pass heat exchanger code incorporated in order to predict Upgraded Mod I and Mod I/Mod II heater tube, combustion gas, and working gas temperatures. After validations against baseline Upgraded Mod I temperature results, the Mod I/Mod II geometry was run (Figures 3-32 and 3-33). Front row heat tansfer correlation factors did not change significantly going from the Upgraded Mod I geometry to the Mod I/Mod II. The increase in the correlation factor for the rear row indicates the impact of the improved fin. The heat transfer coefficient increased $\sim$20% (Nusselt No. increased $\sim$30% at equivalent Reynold's Number). Measured working gas temperature between rows are somewhat lower than calculated due to radiation errors of this thermocouple.

Front and rear Mod I and prototype Mod II heater tube test sections (Figures 3-34 and 3-35) fabricated during the previous Semiannual report period, were tested in the DMR heat transfer rig. Three additional Mod II front row geometries were then fabricated and tested (Table 3-1) in order to encompass the pitch/diameter ratio of the first build double-manifold (configuration No. 1) and single manifold (configuration No. 3) Mod II heater heads. The rig simulates the heat transfer process from the combustion gas to the heater tubes by using a heater tube test section having concentric tubes, with water flowing through the inner tube

*Tube No. 14 - the combustion gas temperature at tube No. 2 shows the effect of increased Mod I/Mod II quadrant bypass caused by a larger gap.

TABLE 3-1

DMR FRONT AND REAR ROW GEOMETRIES

ID	No. Tubes	Measured Pitch (mm)	OD (mm)	Pitch/ OD	Hydraulic Diameter (mm)	Length (mm)	Flow Area (m^2)	Heat Transfer area (m^2)	
Front Test Sections									
215	17	5.62	4.50	1.25	1.43	95	0.00181	0.0228	Mod I
217	15	6.29	3.75	1.68	3.24	95	0.00363	0.0168	Prototype Mod II
299	12	7.85	4.50	1.74	4.27	98.5	0.00396	0.0166	
300	13	7.23	4.50	1.61	3.48	96.9	0.00344	0.0178	Configuration
301	15	6.30	4.50	1.40	2.29	98.9	0.00267	0.0210	No. 3 Mod II
Rear Test Sections									
216	9	10.33	4.50					0.0110	Mod I Style Fins
218	12	7.62	3.75					0.0120	Mod II Style Fins

Hydraulic diameter = $(4/\pi)$ (pitch - OD)
Flow area = (Pitch - OD) (length) (No. tubes)
Heat transfer area = π(OD) (length) (No. tubes)

and stagnant gas layer in the annulus. Gas composition ranges from air to helium and sets the heat flux for a given tube temperature.

Prior to evaluating the three Mod II sections, improvements were made to the rig to provide a more uniform combustion gas temperature (laminar flame burner) and correct an error in combustion gas exit temperature measurement. The error was due to radiation loss by the thermocouples and was corrected by using aspirated thermocouples. The result of these changes was reduced data scatter and altered test results as demonstrated by a repeat test of the Mod I and prototype Mod II sections (Figure 3-36). Data scatter was minimal for all test sections following these improvements (Figure 3-37). Analytical efforts indicated that the theoretical heat transfer correlation*, previously used to evaluate front row DMR data, should be modified to account for a single, as opposed to multiple, tube rows. The correlation:

$$Nu = [0.5 \frac{\text{Tube Pitch}}{\text{Tube OD}} - 0.39] Re^{0.6} Pr^{1/3}$$

must be multiplied by 0.5. Comparison of measured to predicted (for pure convection) Nusselt Number (Figure 3-38) reveals a factor of two difference. The discrepancy is due to combustion products instead of air, radiation, and possibly reactions on the surface of the tubes.

Evaluation of the front row results yields different results depending on whether Nusselt Number (Figure 3-38) or heat transfer coefficient (Figure 3-39) is used due to the effect of hydraulic diameter. Using heat transfer coefficient, the results are as expected, i.e., decreasing pitch yields increased heat transfer for a constant diameter tube. One would also expect increasing diameter for the same pitch to yield a higher heat transfer coefficient. This is not the case, however; the curves (301 versus 217) are so close that the difference lies within the data scatter. Comparison of the various Mod II front row geometries to Mod I (215) reveals similar levels for two of the sections (217 and 301) while the remaining two are lower (Figure 3-39).

Rear row heat transfer is significantly greater for Mod II (Figure 3-40) confirm-

*Kays, W.M., and London, A.L., "Compact Heat Exchangers", Chapter 7, pp. 146-147, McGraw Hill, Third Edition, 1984.

ing the Mod I/Mod II engine test results. Overall the DMR results predict higher heat transfer coefficients (front and rear) than the engine tests. The differences could be due to the proximity to the flame (surface reactions), straight versus involute tubes or front-to-rear row spacing. The influence of tube spacing and number of rows will be evaluated during the next report period.

Materials and Process Development

The goal of this task is the utilization of low cost and low strategic element content materials in the ASE, capable of surviving 3500 hrs of automotive duty cycle exposure. Additionally, this task functions to provide materials related support to the Mod II design and RESD component development activities. Accomplishments during the first half of 1985 included:

- Further fatigue testing of piston base/rod/crosshead assembly
- Testing of a M-CrAlY coated combustion cover
- A review of the Mod II engine for hydrogen compatibility
- Fatigue-proof testing of a hydrogen storage bottle.

Activities planned for the next reporting period include:

- Investigating alternative flamestones
- Hydraulic fatigue testing of Mod II heater head castings of XF-818
- Further proof testing of the piston base/rod/crosshead assembly
- Hydraulic fatigue testing of Mod II cast iron V-block simulative of actual engine operation cycle
- Hydraulic fatigue proof testing of control components
- Enhancing the fatigue properties of XF-818 via HIPing.

Piston Base/Rod/Crosshead Fatigue Test

This task has as its goal the development of a fatigue resistant piston base/rod/crosshead assembly. During this reporting period, two additional fatigue tests were conducted on modified piston rod/base assemblies.

As a review, the initial assembly test reported in the previous Semiannual consisted of a straight line fit between the piston base and piston rod (see Figure 3-41a). Failure occurred at 3.4×10^6 cycles at 100% of full load. See Table 3-2 for the test cycle. Following this failure, two design modifications were tested. The first had a 5° chamfer on the piston base, Figure 3-41b, and the second had an undercut on the piston rod, Figure 3-41c. The 5° chamfer configuration failed after 0.47×10^6 cycles at Step 3, the 140% load level. The undercut piston rod configuration failed after $.4 \times 10^6$ cycles at Step 2, the 120% load level. The numbers seemed to indicate that the 5° chamfer design was more durable, however, a failure analysis of both indicated that no further improvements could be made to the 5° chamfer design. The 5° chamfer design failed by fretting fatigue at the last point of contact (area a in Figure 3-41b). The undercut design failed by fatigue initiating on the radius at the top of the undercut, (area b in Figure 3-41c), a stress raiser, and no fretting was evident.

It was determined that the stress raiser could be reduced by increasing the radius at Area B from .05 to 0.10 mm. These are currently being manufactured and are planned for testing in late summer 1985.

TABLE 3-2 - TEST CYCLE

Piston Base/Rod Crosshead

Step No.	No. of Cycles	% of Full Load
1	10	100
2	10	120
3	10	140
4	10	160
5	10	180

M-CrAlY Plasma-Spray Coated Combustor Covers

An Upgraded Mod I combustor cover was coated with two M-CrAlY plasma-sprayed coatings in an effort to enhace oxidation life of the 310 stainless steel component.

The coated combustor was run for 113.7 hr (including 40 start-ups) in engine No. 5 at MTI. On tear down, the two M-CrAlY coatings appeared to be in great shape. No heavy oxidation, spalling or cracking at the coatings were noted. The combustor cover will continue to be used as a spare.

The coatings compositions for the M-CrAlY are:

a. Amdry 962 - 22 Cr - 10 Al - 1Y - Ni
b. Amdry 995 - 32 Ni - 21 Cr - 8Al - .5Y - Co

These oxidation protection coatings will be applied to a new combustor designs to enhance service life.

Hydrogen Compatibility

Dr. Anthony Thompson, a leading expert in hydrogen/metal interaction, will review the Mod II engine and make any recommendations for change, if necessary. No areas of critical concern were pointed out in the study. Areas designated for further attention were the Inconel 718 dome, the AISI 4130 piston base, and the Nitralloy 135M piston rod. All of these material/ components have seen the Stirling engine high pressure environment for several thousand hours and no failures have been encountered which were related to the hydrogen.

Proof Fatigue Testing - Hydrogen Storage Cylinder

Lightweight, fiberglass wound, aluminum high-pressure air cylinders were subjected to dynamic fatigue testing by internal pressurization with hydraulic fluids on an MTS closed loop fatigue testing machine.

Two cylinders were fatigue tested: 1) a 34.48 MPa rated cylinder which was subjected to an equivalent driving cycle and then 10 million cycles with a pressure cycling between 16.9 to 19.5 MPa, and no damage was detected; and, 2) A lighter, cylinder which was rated for 20.69 MPa was subjected to 10 million cycles with pressure cycling between 18 to 20 MPa, and no damage was detected. This cylinder will be burst tested during the next report period.

Cold Engine System (CES) Development

The primary objective of CES activity is to develop reliable, effective, long life rod seals and piston rings. Development activity during the first half of 1985 was directed at the evaluation of PL seals in engines and at the testing and development of a single piston ring system in a motored engine and in running engines.

Seal and piston ring development will continue during the remainder of 1985 mainly through engine tests.

Main Seals

Endurance and life testing of main seals has been carried out in Mod I engines No. 6 and 7. Both engines operate on an automated CVS type duty cycle shown in Figure 3-42.

In engine No. 6 HABIA PL seals are being tested. The seals in cylinders No. 3 and 4 have now completed 1176 endurance hours without failure. After 270 hrs oil leakage occurred past the seals in cylinders No. 1 and 2; there was no obvious reason for these failures. The replacement seal in cylinder No. 3 has now completed 906 hrs without failure. In cylinder No. 4 the replacement seal failed due to oil leakage after 769 hrs, a new seal was installed and this has now completed 137 hrs.

In engine No. 7 Rulon J PL seals were subjected to a 1000-hr endurance test. After 742 endurance hours the rate of accumulation of oil in the absorption filters for cylinders No. 1 and 2 exceeded 0.02 g/hr. Inspection showed that both seals had failed, oil had entered the cycles and was present on the piston rings; there was no obvious reason for the seal failures. The failed seals were replaced and the test continued until a total of 1017 endurance hours was achieved. After completing the test the engine was torn down for inspection. Cylinders No. 1, 2, and 3 were dry, although there had been oil leakage in cylinder No. 4. The seal was wet with oil and some oil had passed the cap seal. There was a thin oily film on the lower surface of the piston, however, it had not reached the piston ring. There was no major damage to the seal in cylinder No. 4, although it had extruded between the rod and the seal seat in a nonuniform manner which was unusual. Extrusion of the lower tip of the seal had occurred over $\sim 180°$ while the remainder showed no sign of extrusion. Inspection of the relevant parts did not reveal an explanation for this.

Previous tests had shown that excessive crosshead clearance produced premature seal failures. Since the crosshead clearance only indirectly effects the lateral motion of the crosshead this indicated that lateral and/or angular motion of the piston rod relative to the main seal was an important factor in determining seal life. In an engine, the rod motion is also affected by the piston clearance, the length of the piston rod and the alignment of the crosshead guide and cylinder liner axes. At the present time it is not possible to specify limits for acceptable rod motion relative to the seal, although in an engine design, every effort should be made to minimize these. As part of the exercise, a study was made of the P-40, Mod I (with steel crosshead guides) and Mod II engine designs to determine the maximum possible motions of the piston rods in the plane of the main seal. This showed that there were no major differences in the possible lateral and angular motions of the piston rods in the three engine designs.

Piston Rings

In the motored engine an attempt was made to carry out a 500-hr endurance test of single-solid piston rings of the type shown in Figure 3-43. After 300 hrs of motoring, mainly at 2000 rpm/13 MPa high cycle-to-cycle pressure difference developed. Inspection showed that oil had leaked into all four cycles and there was some severe contamination of the piston rings. The test was terminated at this time.

Single-solid rings of the design shown in Figure 3-43 were also tested in engine No. 5. After carrying out a back-to-back performance test with the BOM split-solid rings described in the previous Semiannual the rings remained in the engine while other development tests were carried out. The single rings continued to perform well for 164 hrs when a large pressure imbalance developed. It was subsequently found that in one cylinder some foreign material had entered the appendix gap and been trapped near the upper rider ring. The foreign material had then scored the cylinder liner and damaged the piston ring. All four piston rings were removed at this time. During the 164 hrs of operation wear (loss of weight) of the piston rings was only 0.5-0.8%. After replacing the scored cylinder liner new single-solid piston rings were installed. After break-in cycle-to-cycle pressure differences exceeded the normal accepted limit and this did not improve with further operation. To allow the cycle pressures to equalize a small leak path was provided from each cycle to Pmin by introducing split shim washers between the piston feet and domes. After this the differences in mean cycle pressures were well within acceptable limits and mainly less than 0.2 MPa. However at higher pressures this modification did appear to result in slightly reduced engine power as shown in Figure 3-44. The rings continued to perform consistently for 104 hrs before they were removed to allow other tests to take place. During the 104 hrs wear of the piston rings was less than 0.7%.

During March cold start tests were carried out with the Spirit TTB engine. The starts were carried out at ambient temperature after soaking outside overnight. Over the test period ambient temperatures at start ranged between -12°C and +12°C with the majority close to 0°C. Initially problems were encountered with the hydraulic servo-control system which responsed too slowly to control the engine resulting in a false start. This problem was overcome to some extent by modifying the normal start procedure, although it did not completely eliminate false starts. During the starts strip chart recordings were made of the following:

- Air throttle postion
- Engine speed
- Blower speed
- Heater tube temperature
- Battery voltage

and event records of key on, starter on, and rec. drive (recommended drive away light on). The time from key on to rec. drive was used as a measure of the sealing capacity of the piston rings.

The first series of starts was carried out with BOM split-solid piston rings. With temperatures above 0°C, the start times varied from 90 to 114 sec with an average of 99 sec. Below 0°C, the start times ranged from 59 to 114 sec with an average of 93 sec. There was no significant correlation between start time and temperature and overall, the start times were similar to those experienced at higher ambient temperatures. From this it would appear that the lower temperatures did not have an adverse effect on the sealing capacity of the split-solid piston rings.

A second series of cold start tests was carried out with single-solid piston rings of the form shown in Figure 3-41. With temperatures above 0°C, start times varied from 76 to 140 sec with an average of 98 sec. Below 0°C, times varied from 72 to 168 sec with an average of 115 sec. Again, there was no strong correlation between temperature and start time. Above 0°C, there was no significant difference between start times for split-solid and single solids. Below 0°C there was some indication that the engine took slightly longer to start with the single rings, however, this could have been influenced by the operation of the hydraulic servo-control system.

In order to improve the sealing capability of the single-solid ring, particularly at cold temperatures, the design was modified to increase the interference between the ring and the cylinder liner. To assess the overall effect on performance, rings of the modified design were tested in engine No. 5. To give a direct comparison, the split shim washers were retained between the piston feet and domes. At 15 MPa, the engine power appeared to be slightly higher than with the original single rings as shown in Figure 3-45. At lower pressures, the differences were in the same order as those which might be seen from day-to-day and build-to-build. The rings performed consistently for 90 hrs before they were removed for other tests.

The original single-solid ring design was also tested in engine No. 7 for more than 1000 hrs using the CVS endurance cycle shown in Figure 3-42. This test was carried out with no split shim washers under the domes so that there were no pressure equalizing leak paths. During the test performance was monitored at the 2500 rpm, 15 MPa condition. A large variation in cycle-to-cycle pressure difference was seen during the test, as shown in Figure 3-46, although the variations appeared to be quite random. Cycle-to-cycle pressure differences as high as 1.5 MPa would not normally be tolerated, however, in this case they did not excite any abnormal vibrations and did not have significant effect on the power developed by the engine as shown in Figure 3-47. During the test, there was little variation in the power developed and no sign of any progressive deterioration. Approximately 60 hrs before the end of the test the cycle mean pressures were manifolded to eliminate the cycle-to-cycle pressure differences. This did not have any adverse effects and if anything, there was a slight increase in the engine power; in Figure 3-47 the last three points were taken after the mean pressures were manifolded. At the end of the test the piston rings showed no signs of deterioration and were reinstalled in the engine for another 1000-hr test run.

Engine Drive System Development

Problems reported earlier on the Mod II preliminary design block casting in ductile iron proved unsurmountable for Motor Castings Company, Milwaukee, Wisconsin. Severe porosity in critical gas passage areas could not be eliminated. The tooling was given to John Deere Foundry of East Moline, Illinois and they successfully poured a good casting in March 1985 with porosity eliminated. It was, however, to late from a funding priority standpoint to evaluate the rolling element drive.

The Motor Castings Company block which was machined proved unuseable due to porosity leakage from gas paths into oil gallerys. The durability rig effort was discontinued. Deere Foundry has accepted an order for the Mod II block and will be the vendor for the actual engine block in cast iron.

Control System/Auxiliaries Development

The major goals of this task are the development of the engine control and auxiliaries sytems. Specific control systems goals include the development of a highly flexible DAFC with a low combustion-air pressure drop and low idle fuel flow; development of a simplified mean pressure control that does not require a servo-oil actuator; development of a high-efficiency combustion air blower and development of logic to control the engine actuators optimally.

Priority was given to the effective coordination of critical combustion and mean pressure control tests on the MTI cell and vehicle engines. These tests provided a wealth of data which was applied to improve Mod II control logic.

Toward the end of this half year period, initial agreements were formed to acquire a vehicle chassis dynamometer as used in standard EPA certification. This dyno will allow further significant control and component developments to be made at MTI.

Combustion Control

Testing of the DAFC was carried out along with other vehicle performance checks at the Mercedes Automotive Emissions Laboratory in Ann Arbor, Michigan. The January version used an Hitachi mass airflow meter and a micropump gear-type fuel metering pump.

The system performed well on the vehicle with the exception of some undesired and uncontrolled lean A/F ratios. One error source was determined to be dynamic cavitation of the pump, something not occurring in steady state. Vehicle acceleration from idle required a sudden acceleration of the near stationary fuel in the vehicle's tubing. The fuel pump momentarily cavitated, resulting in a leaness at the most undesired time. During this testing, it was also determined that the pump's fuel flow-to-motor rpm characteristic was found to be too sensitive to such changing downstream conditions as the nozzle heating up. However, other functions of the DAFC performing well were the airflow meter and the DEC logic for keyboard adjustability. To make the system fully operational, it requires only effective isolation of the fuel flow controller from shifts in upstream and downstream conditions. Several different components were procured for this purpose.

Extensive testing was carried out to select the optimum fuel flow actuator for Mod II. Eight different configurations were tested and reviewed against seven selection criteria (see Figure 3-48). A pair of Ford automotive fuel injectors manifolded together exceeded all others in the selection criteria. The fluid and electrical systems were then designed and prototyped (see Figure 3-49). Pressure regulators maintain a constant ΔP across the injector orifices, ensuring insensitivity to the changing external conditions. The pressure and electronic driver timing was then optimized to permit both adequate control resolution at the low idle point and sufficient fuel flow for a hard acceleration. This required a metering system capable of operating precisely and reliably over a 35:1 turn-down ratio.

The dual Ford injectors are operating as fuel flow valves to the BOM nozzle. Each valve is pulsed open for an identical fraction of their 100 ms periods. However, the second valve's pulse starts half a period after the firsts. The net effect is a 20 Hz pressure wave to the nozzle, which is too fast to effect emissions as determined by combustor exhaust sampling. Engine testing of this new system is planned in the cell during the second half of 1985.

Also noted at the January vehicle tests were the effects on combustion control of the DAFC's reduced airflow restriction. The default blower map (originally set up for the K-Jetronic A/F control) supplied more excess air than was necessary for temperature control. Consequently the air throttle was used excessively to reduce the excess air. This, in turn, resulted in increased oscillations of airflow, temperature, and emissions. During March, a more appropriate Mod I blower map was tested and developed in the cell. It took advantage of the DAFC's less restrictive air meter while utilizing the real characteristics of the Mod I air throttle. Blower power wastage and excess air throttle activity were reduced.

A new improvement in Mod I blower speed control was developed at the January Mercedes testing. Formerly, the blower increase solenoid added hydraulic fluid to the variator faster than the DEC could effectively monitor the changing blower speed. Consequently the blower speed control frequently produced undesired oscillations. A throttling valve was installed between the hydraulic pump and the speed increase solenoid. The restricted flow enabled the DEC to catch up with the increased blower speed before it overshot. Thus a much smoother blower speed control was effected with an even tighter error band.

The January vehicle tests bore fruit in another area as well. An intermittent problem with initial combustion ignition was traced to excessive atomizing air at start-up. The airflow was sufficient to blow out the ignition. An immediate solution was to bleed off the excess atomizing air at light off. This permitted repeatable rapid ignitions. Once good ignition was attained, the bleed was closed. Sufficient atomization was then available for clean emissions and low differential temperatures across the heater heads. This same strategy will be incorporated into Mod II system design. An appropriate solenoid valve has been selected.

With Mod II there will be neither a separate upstart motor nor an hydraulic system. Consequently, an electrically-driven atomizing air compressor will be necessary. Two 12 VDC motor driven compressors were procured for testing on Mod I engines. As the pumps are a positive displacement type driven by near constant speed motors, they are virtually constant flow devices.

At the nozzle, there are back pressure interactions with the fuel flow. As the fuel flow increases, the atomizing air back pressure also increases. With a pressure regulated atomizing source, the airflow decreases as the system back pressure increases. However, with a constant flow device this does not occur. The constant flow method may result in lower, more stable, emissions during vehicle urban cycling.

The two compressors were tested in the Spirit vehicle with EGR and CGR combustors on urban cycles at the nearby ENCON test facility. The smaller, rotary vane compressor, providing atomizing airflow of 0.5 g/s, produced clean emissions with the EGR combustor. The larger diaphragm type compressor, providing 1.0 g/s, was necessary for clean emissions with the CGR compressor. Figure 3-50 shows a strip chart recording of fuel and atomizing air pressure changes while atomizing airflow remains stable.

To ensure long term calibration of the A/F ratio system, several lean exhaust gas oxygen sensors have been under consideration. These will be positioned in the Mod II engine's exhaust to monitor lambda. As the system's mechanical parts wear, the sensors will feedback correcting information to adjust DEC maps. The sensors will also provide use with a low-cost, transportable, and nearly real time record of exhaust conditions. Prototype units are being procured from NGK and Bosch.

Control System Analysis

Several new control algorithms were identified as necessary for maximization of the Stirling engine's potential in the automotive application. Each of these algorithms will have sensor input to actuator output relationships that will need to be optimized. The optimization is done most productively by computer simulation. In preparation for Mod II work, a simulation for Mod I is being developed and tested against actual engine performance.

Existing simulation codes and engine test data inputs were reviewed for validity and consistency. Standard documentation methodology was applied to the simulation for the engine process and its control. Several further test data inputs were identified as necessary for effective modeling, and a test plan was devised for data acquisition of these from actual engine performance.

Control Engine Support

Throughout this reporting period the MTI Engine Test Cell and Spirit Test Vehicle were maintained at a high level of operation for program development and demonstration purposes. Numerous engine tests were facilitated by wiring and calibration of specialized sensors and recording equipment. Some of the major tests included: characterization of the Raised Appendix Gap for ASE Code development; several trips to the ENCON facility for mean pressure and combustion control development; two trips to Mercedes Emissions Lab for vehicle performance verification; and several cold start, idle and CGR combustor tests on the vehicle in the MTI garage. DEC programming changes were also implemented during this time in support of developmental needs and findings.

A DEC was prepared for NASA's use with engine No. 10 at their Lewis facility. This was a major preparation which included: assembling and populating printed circuit boards and control panels; constructing wire lists and cables for the NASA cell-to-DEC communications; and wiring in, checking out, and calibrating the new installation.

Mean Pressure Control

Analysis of the Spirit testing done in January has revealed that there is a substantial fuel economy penalty on the Urban CVS cycles due to limited hydrogen compressor capacity at low engine speeds. Optimization of the Mod II compressor size was initiated to find the best balance between steady-state compressor parasitic losses and pump-down capacity. A substantial fuel economy penalty due to limited compressor capacity was found. Until now, perfect pressure response by the Mod I engine was assumed and has produced little fuel economy prediction error. However, with the higher gas mileage of the Mod II, the sensitivity to this assumption has increased.

Several alternative hydrogen pump-down and storage systems were evaluated for fuel economy, complexity, weight, and risk.

The selected system will consist of a multiple capacity compressor and two storage tanks. The compressor size was optimized for maximum fuel economy and detailed design initiated. Design of the Mod II Pmax and Ptank power control blocks was completed.

Testing on Upgraded Mod I engine No. 5 was also performed to determine the cause of the engine overspeeding during high speed shifts in the Spirit vehicle. A large orifice solenoid valve was connected between the Pmax and Pmin manifolds on the engine. Opening of the valve at any operating point should have caused the engine output power to drop to zero, indicating adequate short circuiting flow. Initially, the flow was inadequate and the engine output power would not drop to zero at 9 MPa and 2500 rpm. The check valves were then replaced with the latest design (4-18180). The test was then conducted over many operating points from 2500 to 4000 rpm, at 3 to 15 MPa. This was done at both 720° and 820°C mean rear-row tube temperatures. The short circuiting flow was adequate at all tested points except 9 and 11 MPa at 400 rpm and 820°C. Investigation revealed that the hydrogen flow passages in the tested engine are not as large as the most recent design (used in the Spirit).

A set of the latest design cycle check valves (4-18180) was then installed in the Spirit for evaluation. The improvement in the check valve flow rates seen in the test cell in March did prove to be the solution to the Spirit engine overspeeding. Use of these check valves resulted in good controllability of the engine at all speeds and load conditions.

The goals for Mod II auxiliaries and controls require an improvement over the bulk, cost, and reliability of the Mod I hydraulic system and and Moog electro-hydraulic actuator for the PCV. Toward that end, an electrically-driven PCV was substantially developed during this first half year. A bench testing facility was established for control and endurance testing. The new actuator was put under maximum expected load for 200 hrs of endurance operation. The motor gearbox combination appeared to hold up well with no significant wear to the gears, but with some wear to the motor brushes. However, the final drive gears (rack and pinion) did show excessive wear. In part, this was due to one-time-only grease lubrication. The test was repeated on a second set of gears for 100 hrs of endurance with improved lubrication. Still, there was excessive wear. An improved set of hardened gears were ordered during this time to be tested later.

In parallel with the endurance testing was controllability development. Three different combinations of gear boxes were tested for small signal response and slew rates. An improvement in small signal response over the hydraulic actuator was quickly attained. (Figure 3-51 shows response of the most durable gearbox.) However, improvements in slew rate (large movement velocity) was still needed. This was then effected by using a new 8 VDC motor while reducing pressure imbalance in this system. A new (commercial) electronic motor drive was procured and further modified to improve controllability. The net result was an electrically-driven PCV accurate to ±0.05 mm, with improved small signal response, and improved slew rate (Figure 3-52).

Mod II Electronics Development

The Mod II instrumentation list (for controls development and test cell performance use) was reviewed and updated. The Mod II DEC block diagram and input/output list were updated.

The wiring list for the Mod II DEC and controls cabling was started. This list will define all cables and wires in the system and will include all the modifications and updates. This list will form the basis of all Mod II wiring documentation. As more sensors and actuators are added to the engine, the changes needed to support them become increasingly complex.

Prototype hardware modifications were designed and installed in the bench DEC. These include provisions for the cold junction thermistor, upgrading two +10 VDC reference circuits, and interfacing needed to support the new high level pressure transducers. All of these changes proved successful. Software modifications to adjust the scaling of the pressure transducers were also accomplished.

DEC hardware changes to allow Mod II blower speed and battery charge regulator control were designed, prototyped, and documented. Software was written to support the Mod II atomizing air compressor and activate the alternator on/off signal. The new combustion control subroutine was developed. This version (Mod II) uses a Proportional, Integral, and Derivative (PID) controller to directly drive the blower through the brushless DC motor controller without an air throttle. Blower speed is based on the ΔT between temperature set and rear mean temperature. Guards and limits for the blower demand and the actual blower speed were installed, and testing will occur in July.

Auxiliary Development

Mod II Alternator/Blower Motor Development

Initially, the Mod II alternator speed was selected as 32,000 rpm to provide the highest efficiency and ease of construction due to its similarities of design with the 32,000 rpm blower motor. A high speed test bed with load banks was prepared for mapping alternator performance. The alternator was designed, fabricated, assembled, and balanced. Initial testing indicated an 89% bench efficiency.

On the engine, however, a speed increaser would be needed to turn 4000 engine rpm into 32,000 alternator rpm. A commercial traction ball increaser was procured and evaluated. The size and efficiency of the device was unsatisfactory. A planetary gear-type speed increaser was also designed and evaluated, indicating a higher efficiency and reliability. However, due to the weight and size penalty of the 8:1 speed increaser options, a 16,000 rpm alternator was further investigated. All indications were that a 4:1 speed increase could be effectively carried out using poly-V belts and pulleys with comparable net efficiency.

A 16,000 rpm alternator was then selected for Mod II. The alternator was designed and a prototype was fabricated. Acceptance and performance tests were completed in June. Efficiencies in the 90% region were obtained with 1.5 kW output on both the battery charge and blower power coils.

Mod II Blower Development

Mapping of ΔP versus flow for various speeds of the Mod I Blower was carried out on the MTI test rig. The anticipated Mod II combustion air requirements indicated a redesign of the Mod I blower impeller will be necessary to provide the higher Mod II airflows across the CGR combustor's higher pressure drops. Two turbomachine consultants were contacted for evaluating potential blower efficiencies. An efficiency of 80% (up from 68%) was indicated as possible for our potential motor speeds. The Mod II blower point was then selected as 36,000 rpm, 184 scfm at 96 in. of water pressure rise. The projected blower efficiency is 80-83% as illustrated in Figure 3-53.

By June, The Mod II blower motor design and drawings were in progress. As well, the prototype brushless motor drive was assembled and undergoing preliminary tests. A blower backplate was designed and fabricated to permit the prototype 36,000 rpm motor to be fitted to a Mod I type blower.

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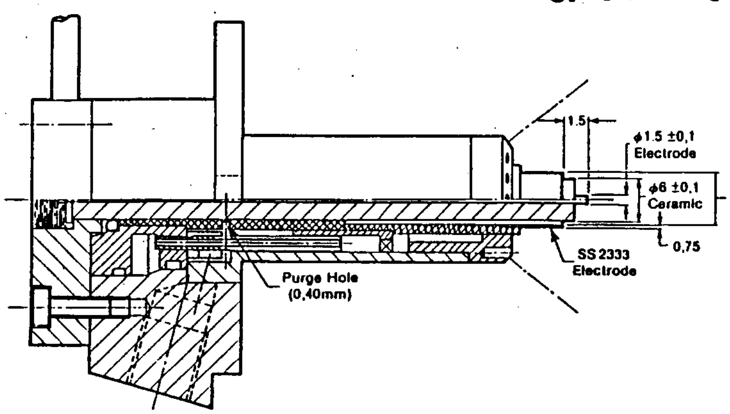


Figure 3-1 Modified BOM Nozzle with Concentric Center Electrode

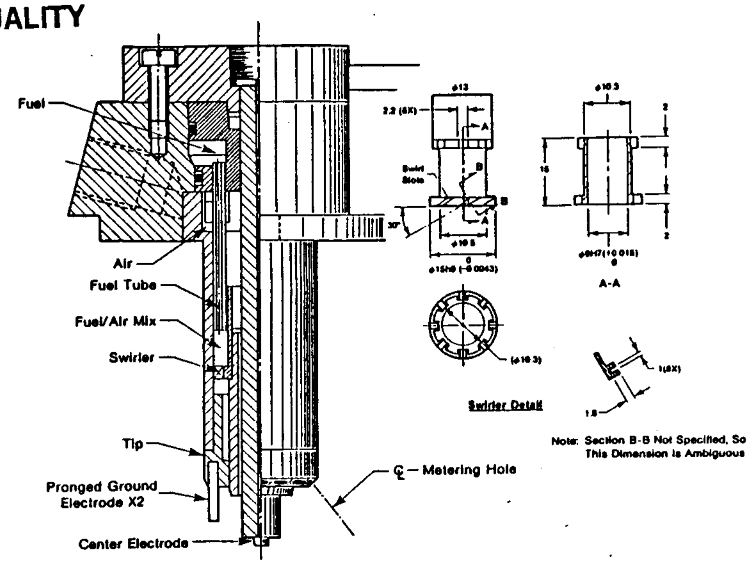


Figure 3-3 Mod I BOM Fuel Nozzle Showing Upgraded Mod I Swirler Detail

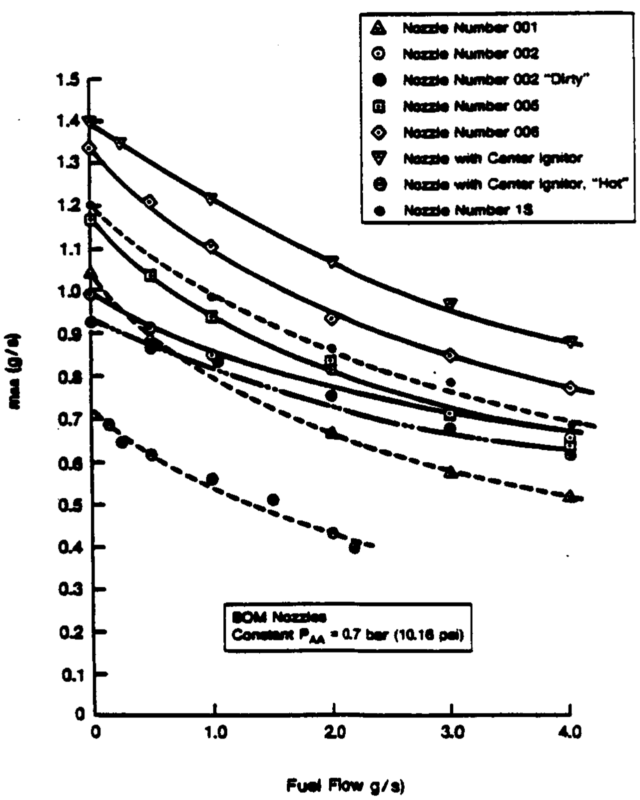
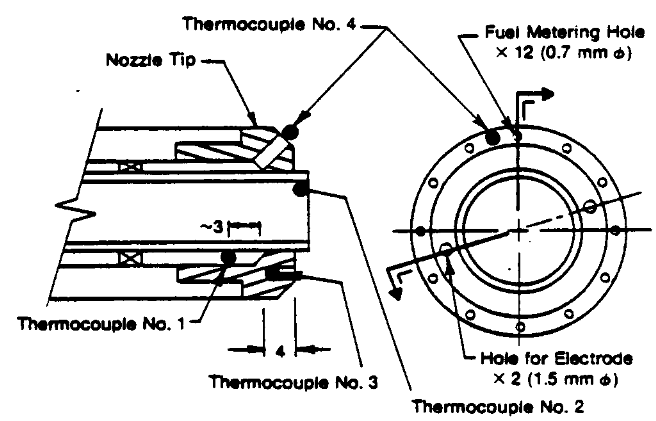


Figure 3-2 Atomizing Airflow variation of BOM Fuel Nozzles



Thermo-couple No.	Description
1	Fuel/Air Mixture
2	Center Sleeve
3	Metal Temperature between Fuel Holes (Shielded)
4	MetalTemperature — Pressed into Electrode Hole

Figure 3-4 BOM Nozzle Thermocouple Locations

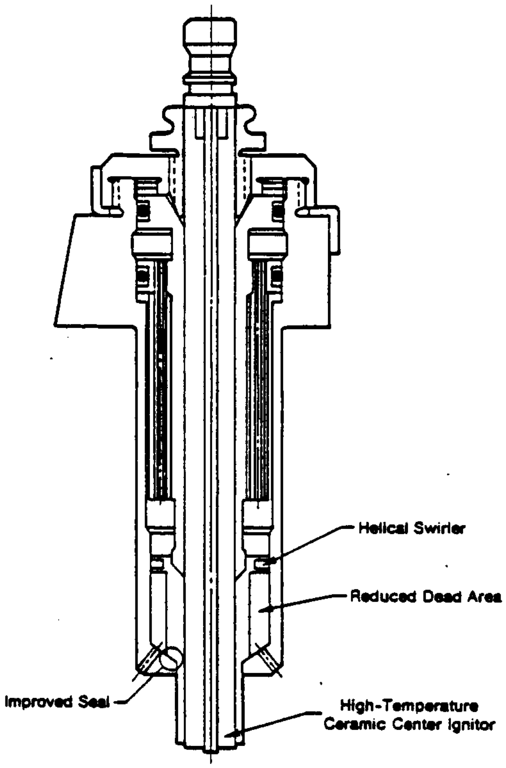


Figure 3-5 Mod II Nozzle/Igniter

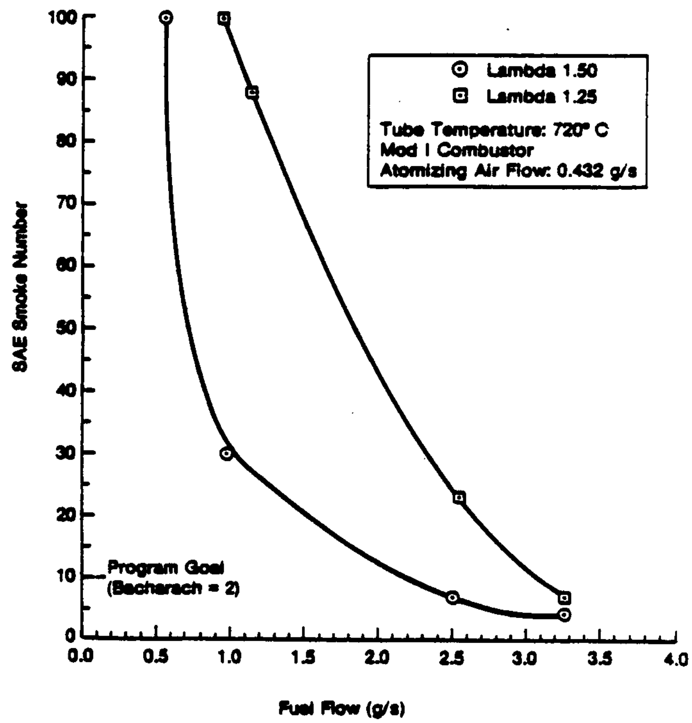


Figure 3-7 Soot Emissions from High ΔP Conical Fuel Nozzle and TTE Combustor in EHS Performance Rig

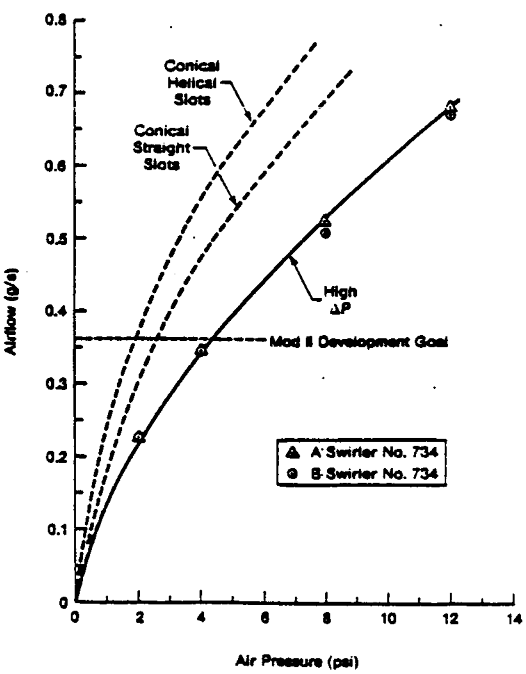


Figure 3-6 Conical High ΔP Atomizing Airflow Characteristics

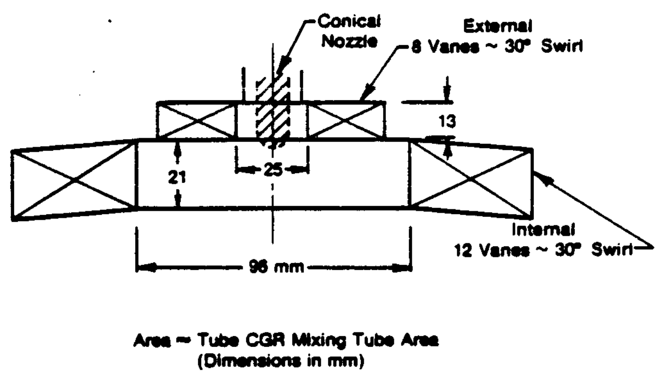


Figure 3-8 Modifications to Improve Conical Nozzle Performance

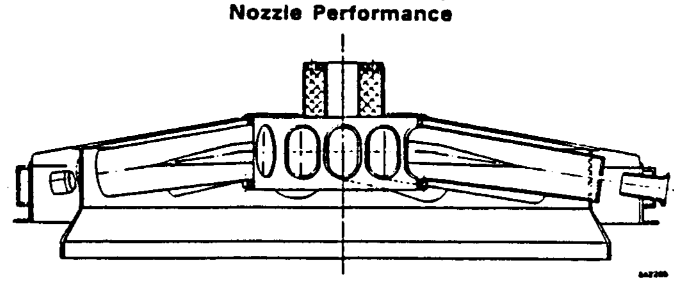


Figure 3-9 Upgraded Mod I Tubular CGR Combustor

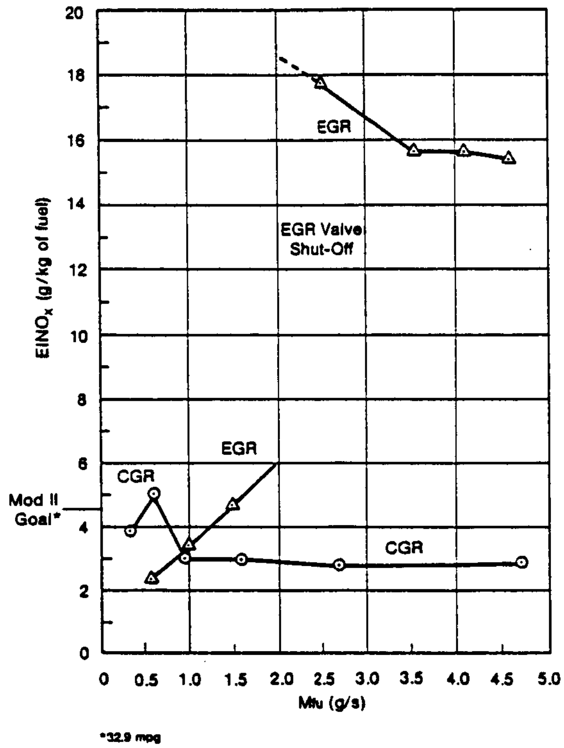


Figure 3-10 Comparison of EGR/CGR NO_x Emissions at 820°C - Engine Data

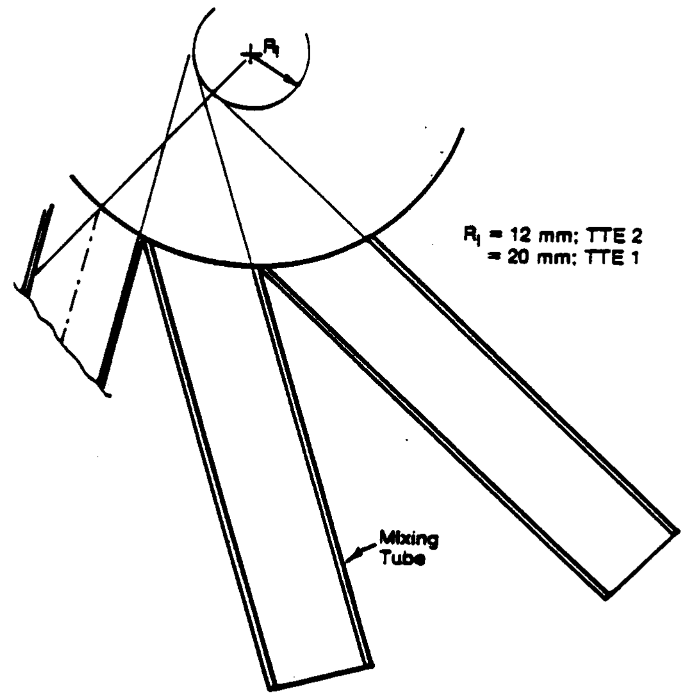


Figure 3-12 Definition of CGR Combustor Parameter, Ri

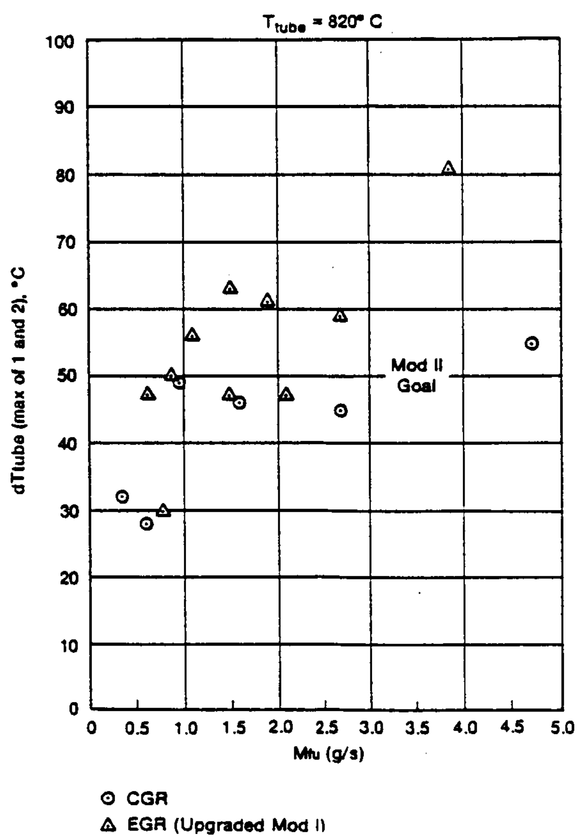


Figure 3-11 Comparison of EGR/CGR Heater Tube Temperature Variation at 820°C - Engine Data

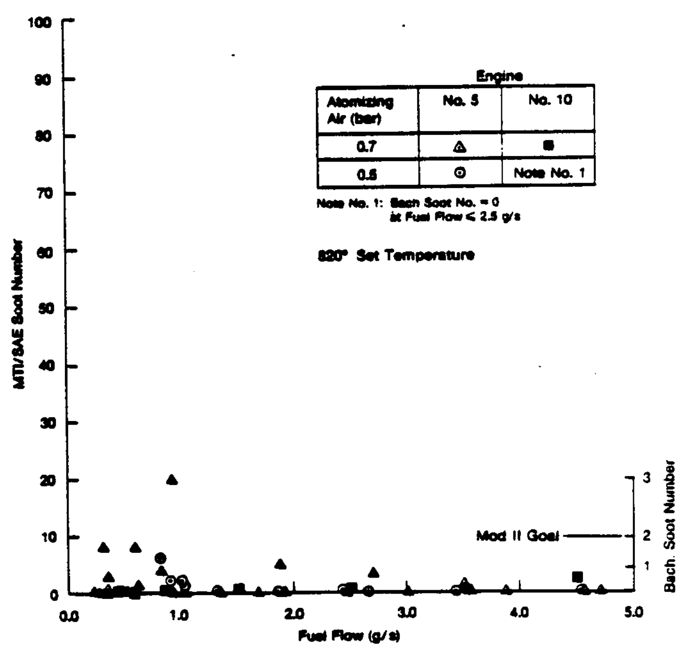


Figure 3-13 Upgraded Mod I TTE 1 CGR Combustor Soot Number

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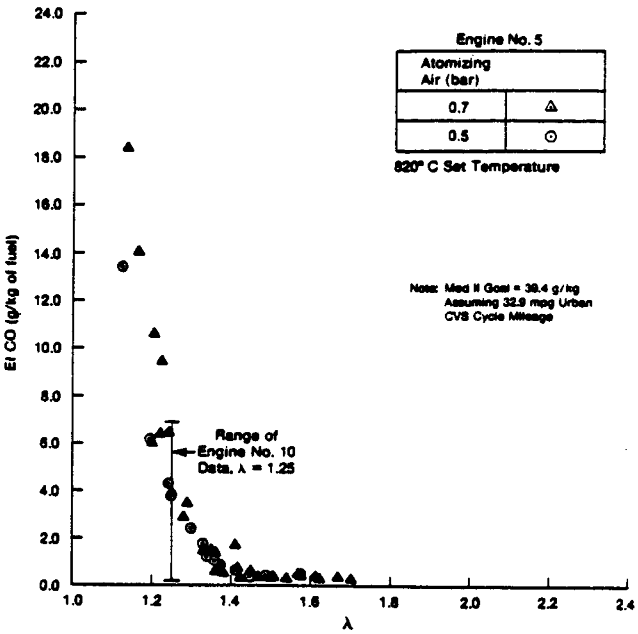


Figure 3-14 Upgraded Mod I TTE 1 CGR Combustor CO Emissions

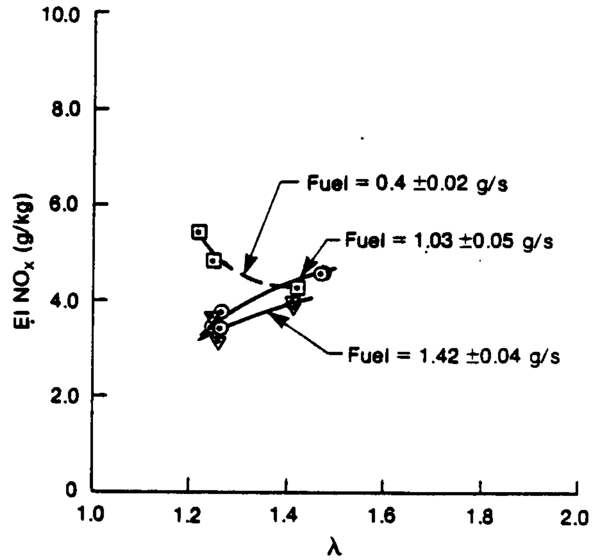


Figure 3-16 Effect of λ on TTE 2 Combustor NO_x Emmissions

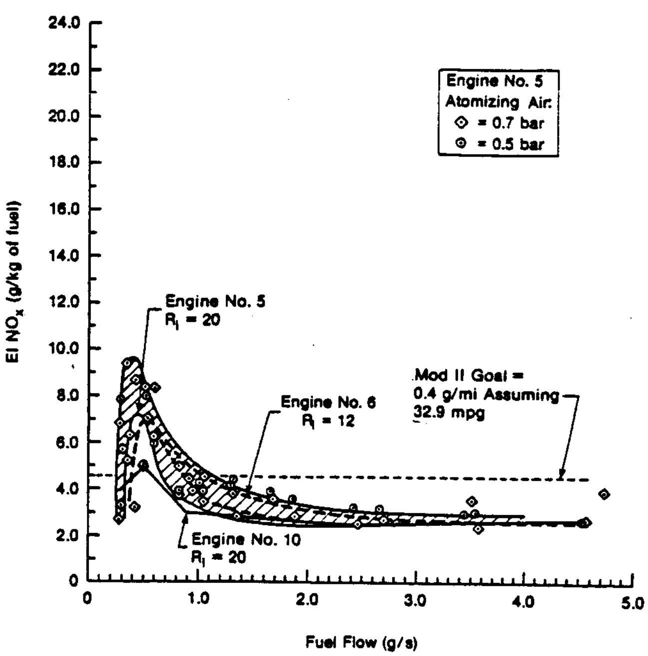


Figure 3-15 Comparison of NO_x on Emissions 5, 10, and 6 with TTE CGR Combustor

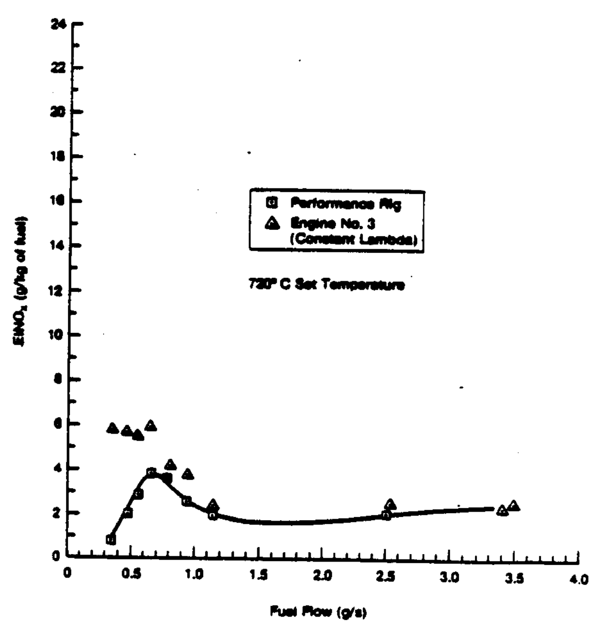


Figure 3-17 Mod I TTE 2 CGR Combustor NO_x Emissions Comparison

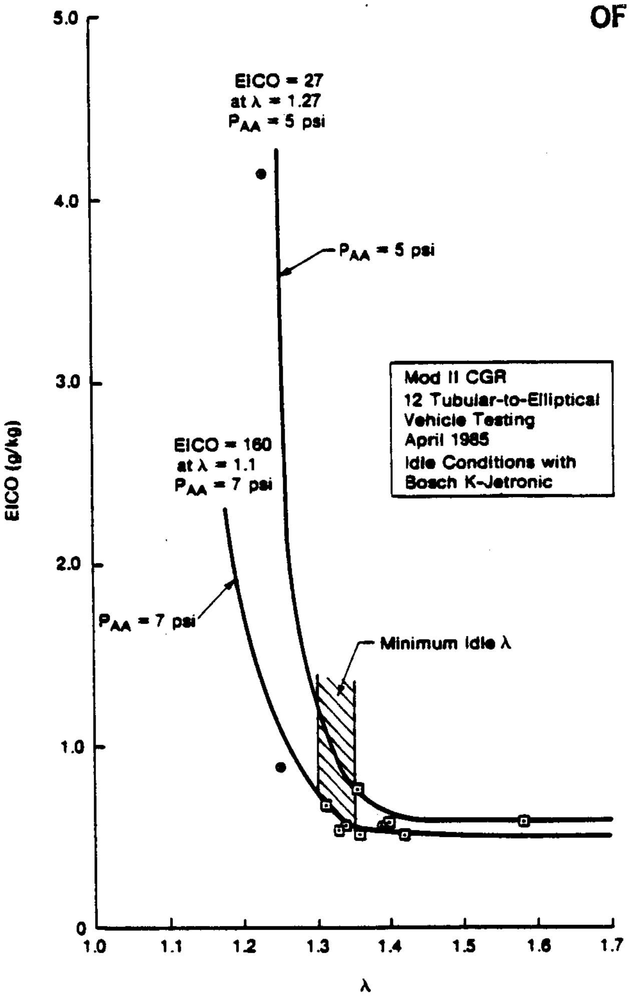


Figure 3-18 Vehicle Idle CO Emissions

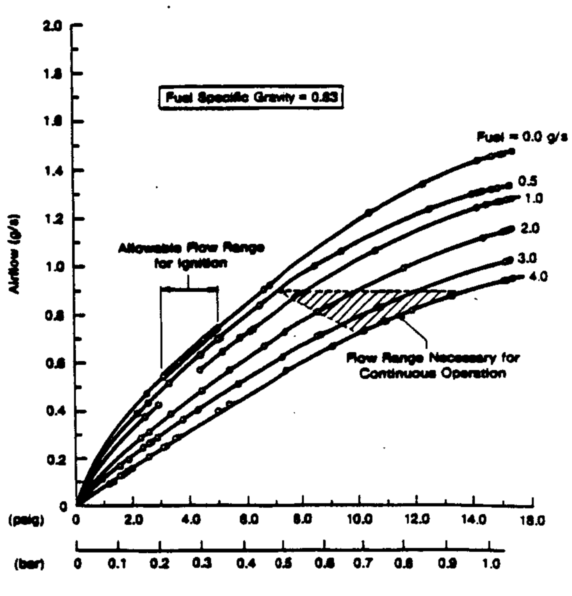


Figure 3-19 Mod II Atomizing Airflow Reqiurements

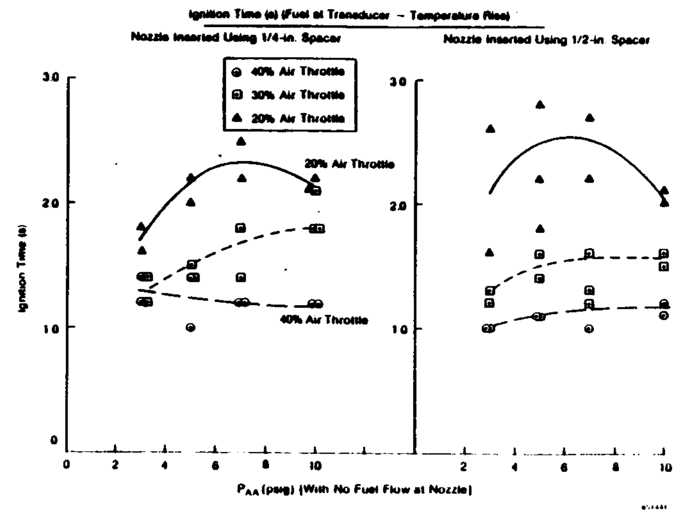


Figure 3-20 Ignition Times for TTE CGR Combustor in Spirit Vehicle

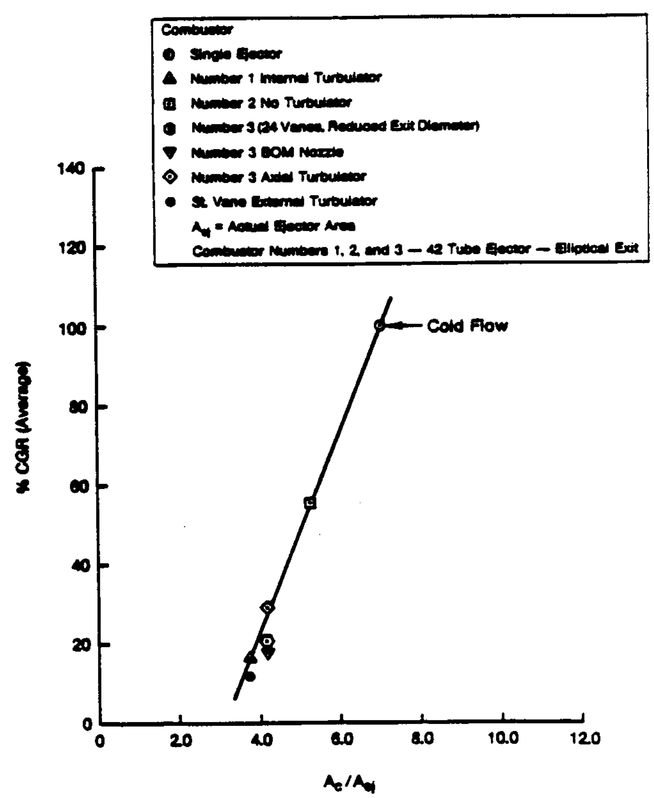
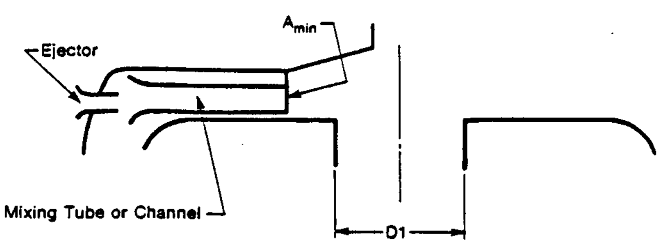


Figure 3-21 Effect of Mixed Flow Exit Area on CGR



A_{min} = Minimum Mixing Tube or Channel X-Sectional Area — Based on Hydraulic Diameter

A_1 = $(\pi/4)\ D_1^2$

A_c = $1/(\sqrt{[1/A_{min}^2] + [1/A_1^2]})$

Figure 3-22 Definition of Mixed Flow Combustor Area

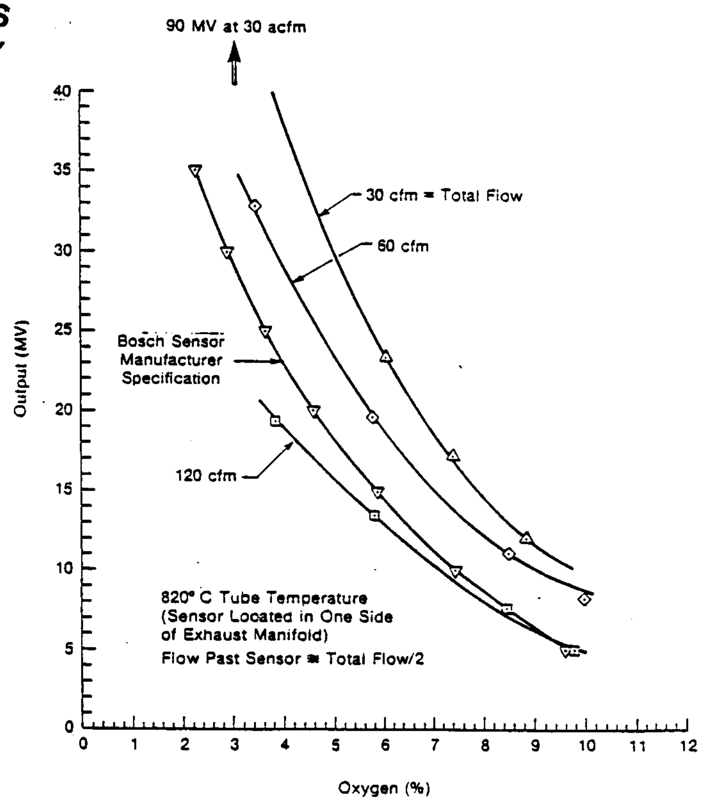


Figure 3-24 Bosch Lambda Sensor Output versus O_2%

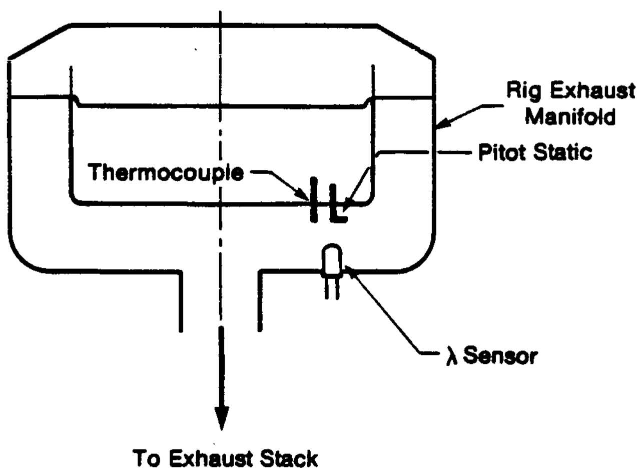


Figure 3-23 Relative Location of Bosch λ Sensor in Performance Rig Exhaust

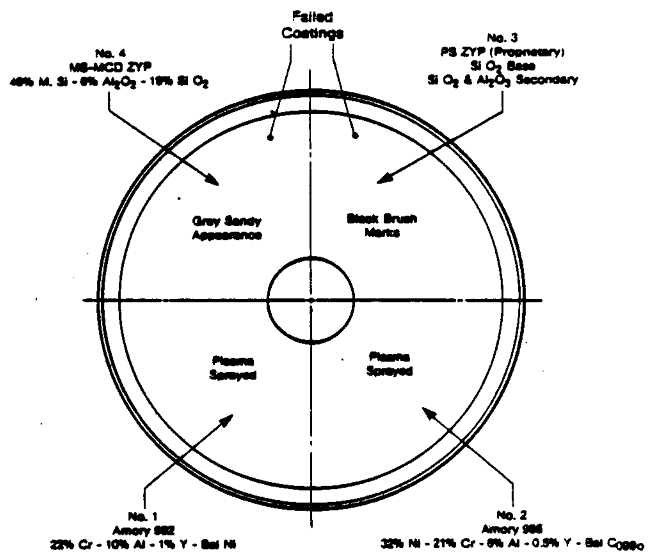


Figure 3-25 Identification of Oxidation-Resistant Coating (view from bottom)

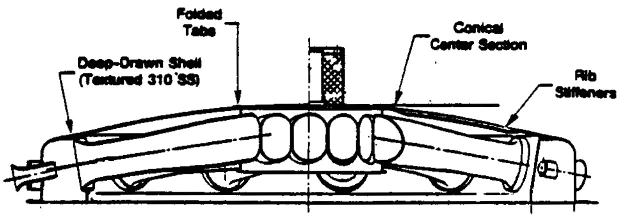


Figure 3-26 Mod II CGR Combustor

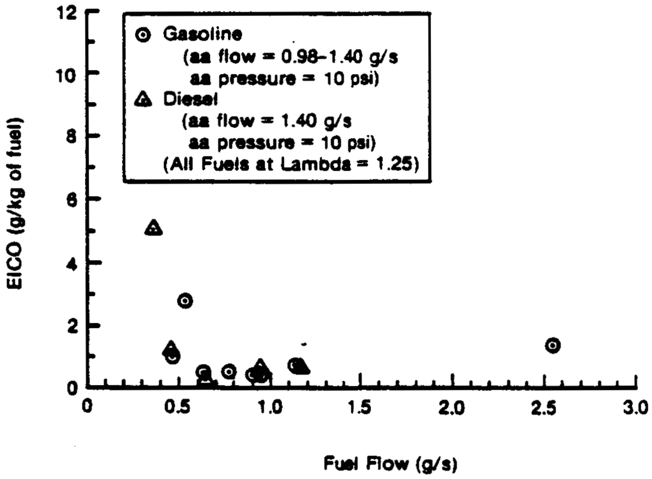


Figure 3-27 CGR Combustor CO Emissions, Gasoline, and Diesel Fuel - Data Points at 820°C

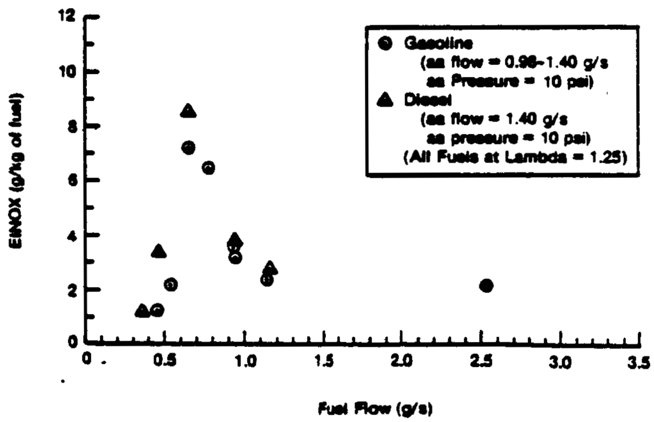


Figure 3-28 CGR Combustor NO_x Emissions, Gasoline and Fuel - Data Points at 820°C

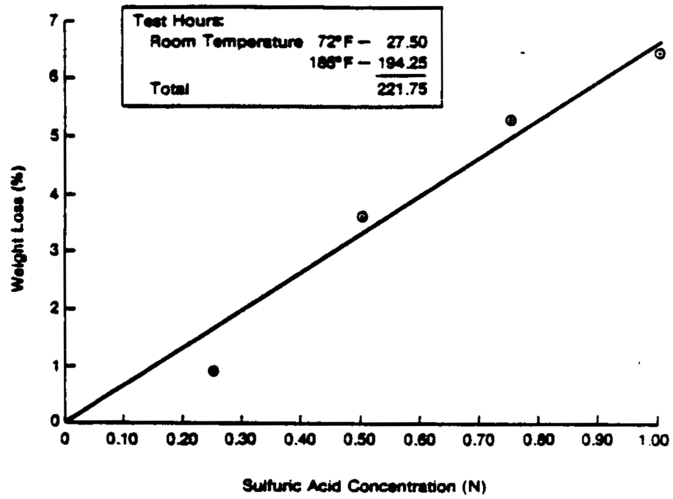


Figure 3-29 Mixed Oxide Corrosion Test Results

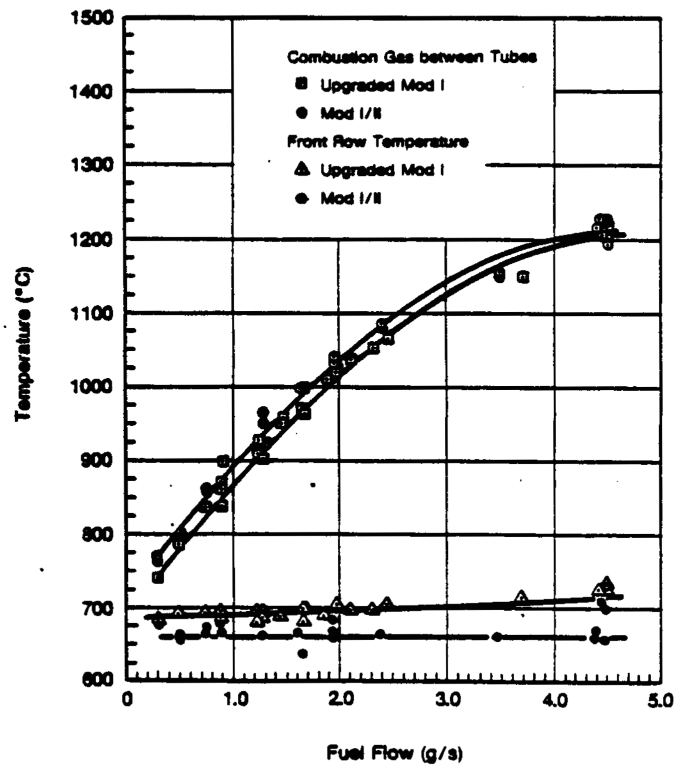


Figure 3-30 Heater Heads (Baseline and Mod I/Mod II), 720°C

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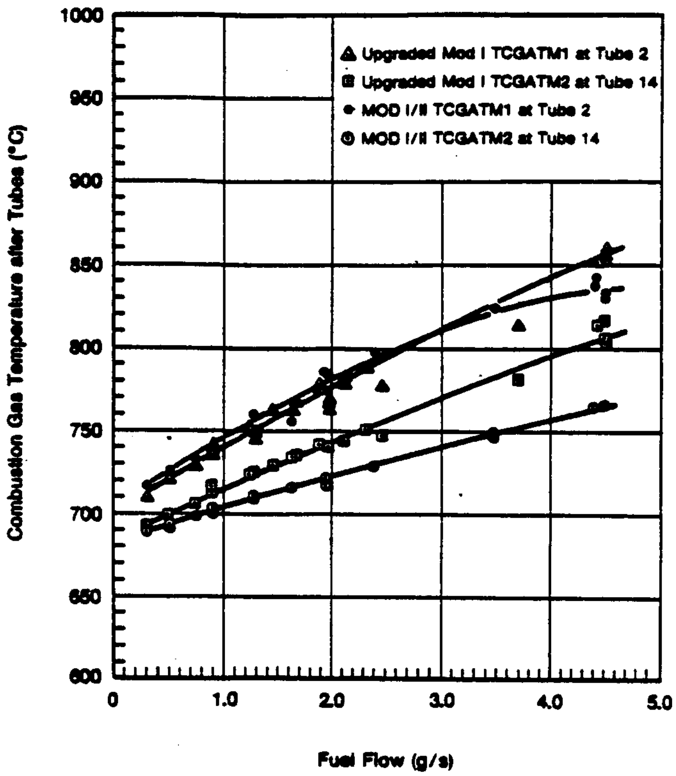

Figure 3-31 Heater Heads (Baseline and Mod I/Mod II), 720°C

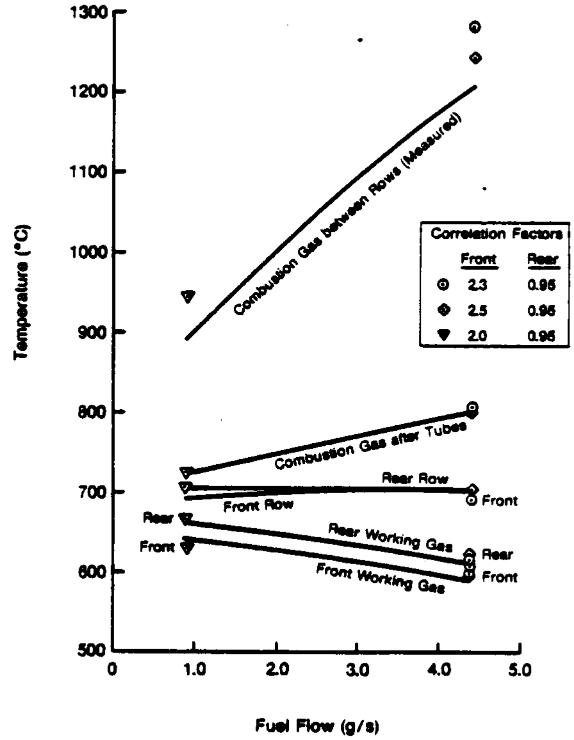

Figure 3-33 Mod I/Mod II Correlation Factors

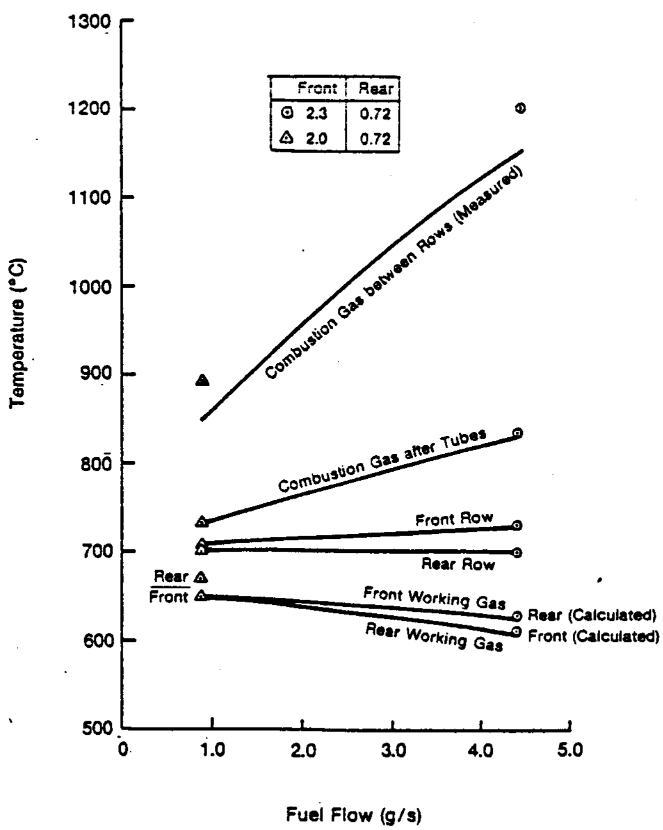

Figure 3-32 Baseline Upgraded Mod I Correlation Factors

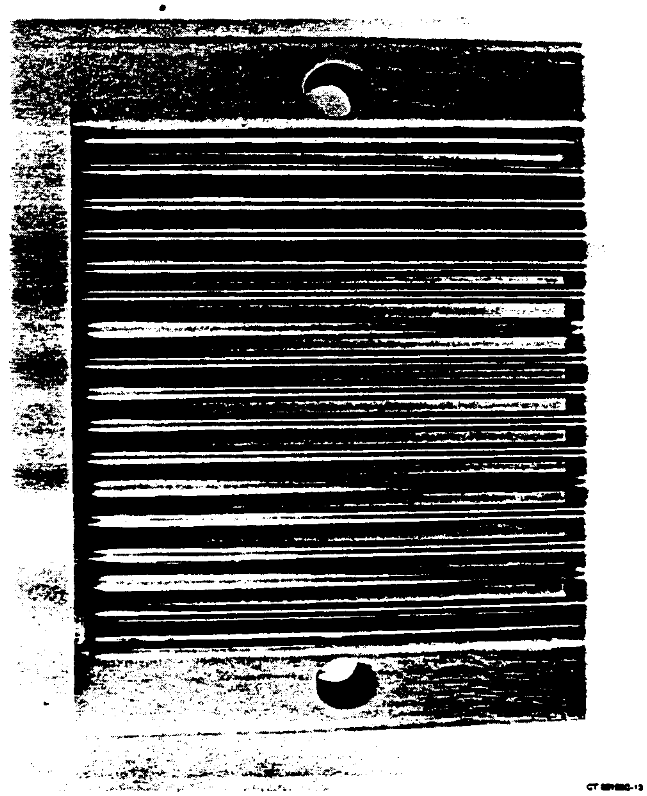

Figure 3-34 DMR Mod II Front Row Test Section

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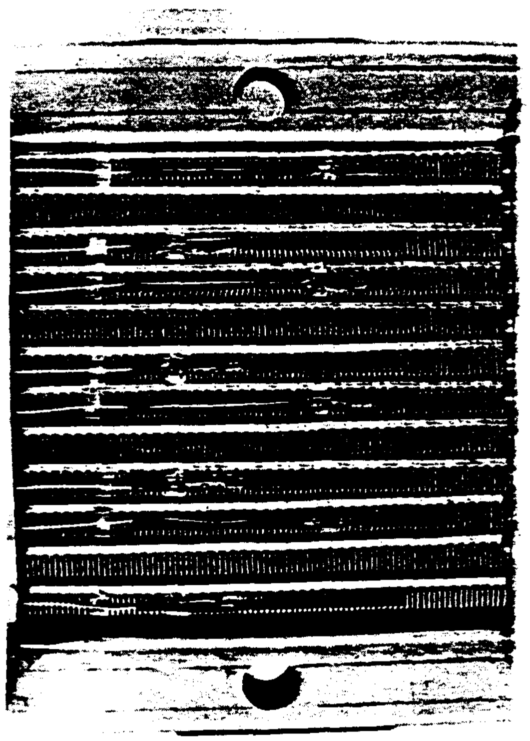

Figure 3-35 DMR Mod II Rear Row Test Section

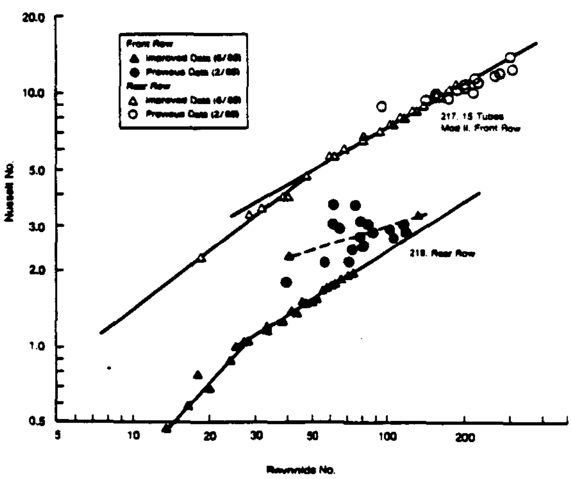

Figure 3-36 Comparison of DMR Test Data

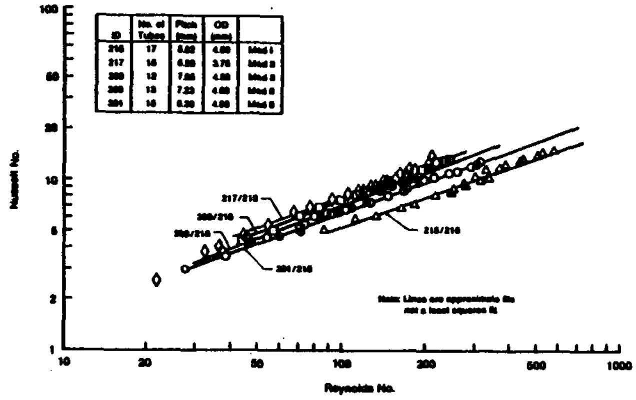

Figure 3-37 Mod II DMR Testing (Actual Data) Front Row (Using Tube Diameter as the Length Parameter for Nusselt and Reynolds Numbers)

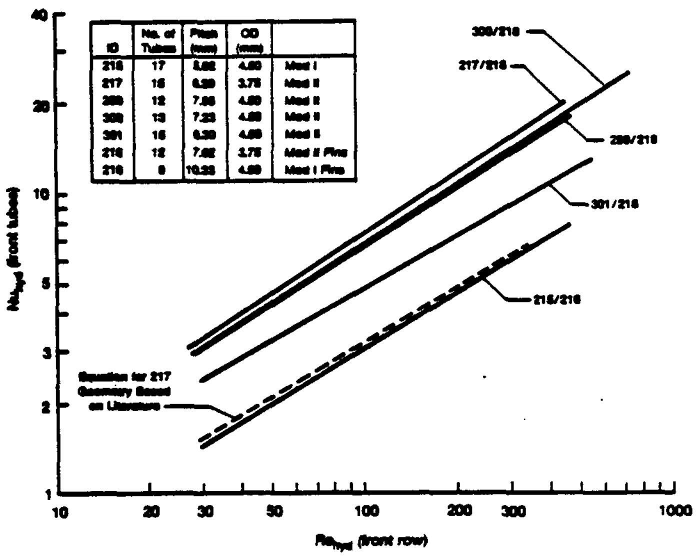

Figure 3-38 Nusselt Number (based on hydraulic diameter) versus Reynolds Number (hydraulic Diameter)

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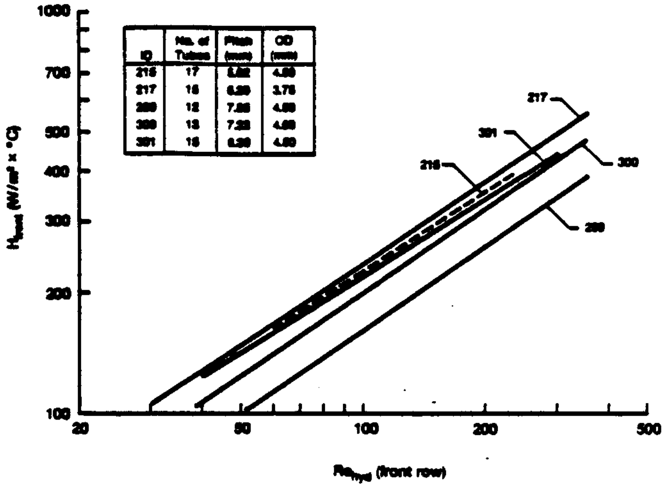

Figure 3-39 Heat Transfer Coefficient versus Reynolds Number (Hydraulic Diameter)

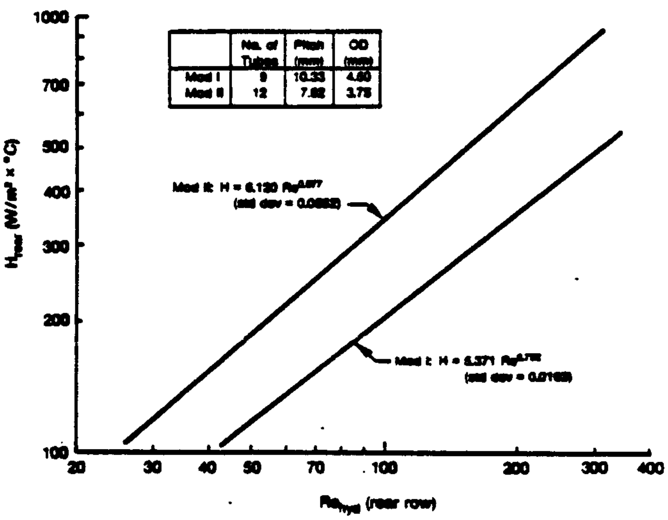

Figure 3-40 Heat Transfer Coefficient versus Reynolds Number (Hydraulic Diameter)

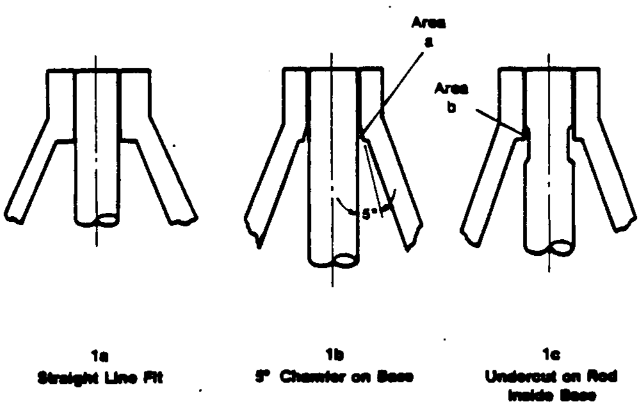

Figure 3-41 Piston Base/Rod Joint

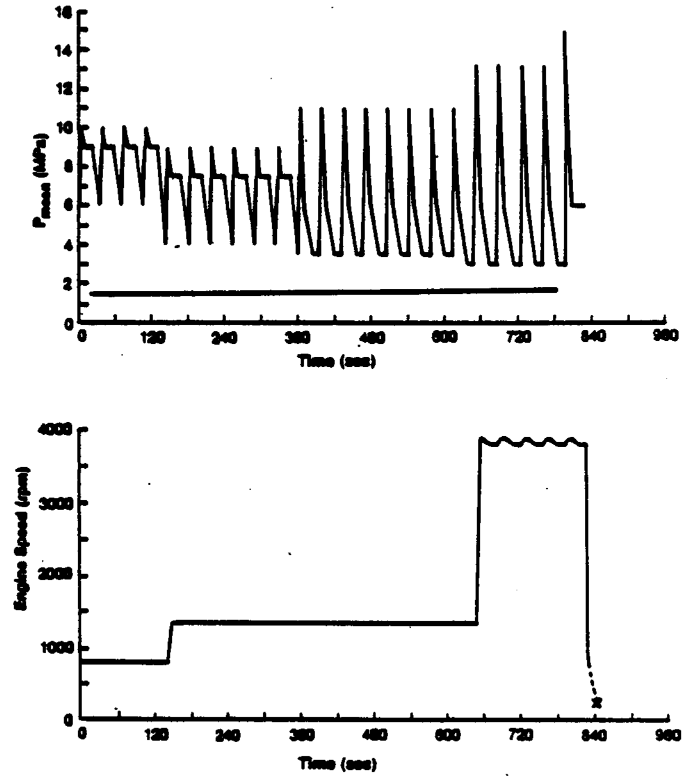

Figure 3-42 CVS Endurance Cycle for Engines No. 6 and 7

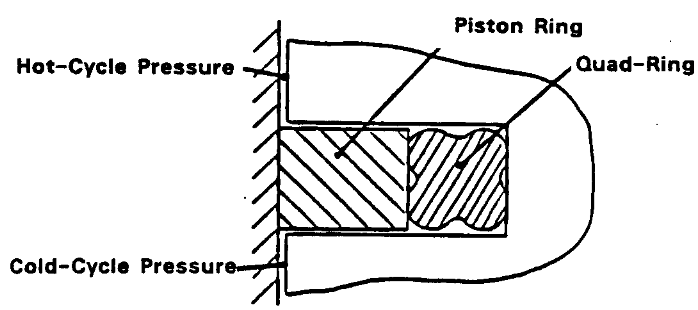

Figure 3-43 Single-Solid Piston Ring

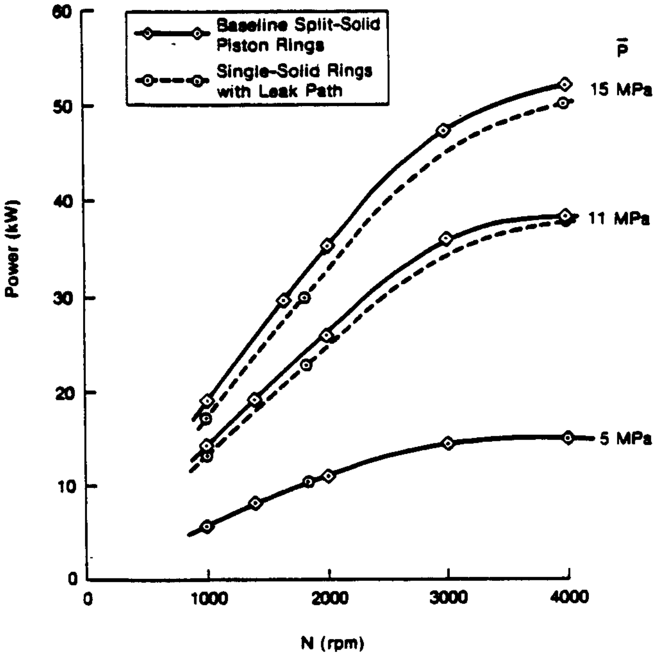

Figure 3-44 Single-Solid Piston Rings in Engine No. 5

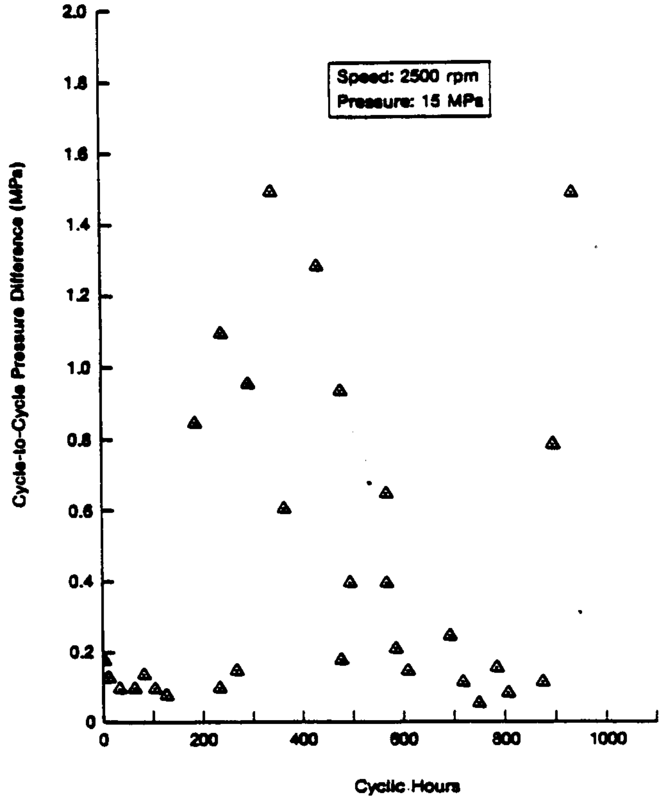

Figure 3-46 Single-Solid Rings in Engine No. 7

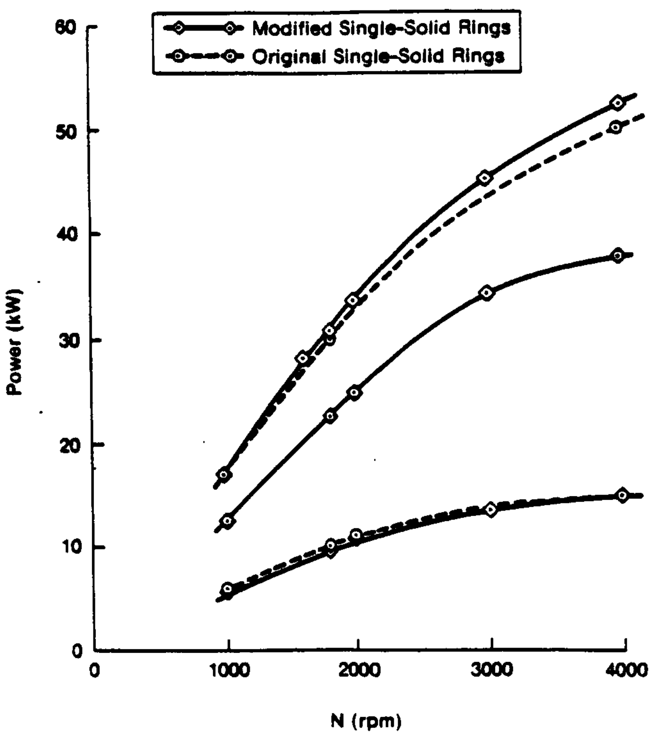

Figure 3-45 Modified Single-Solid Rings in Engine No. 5

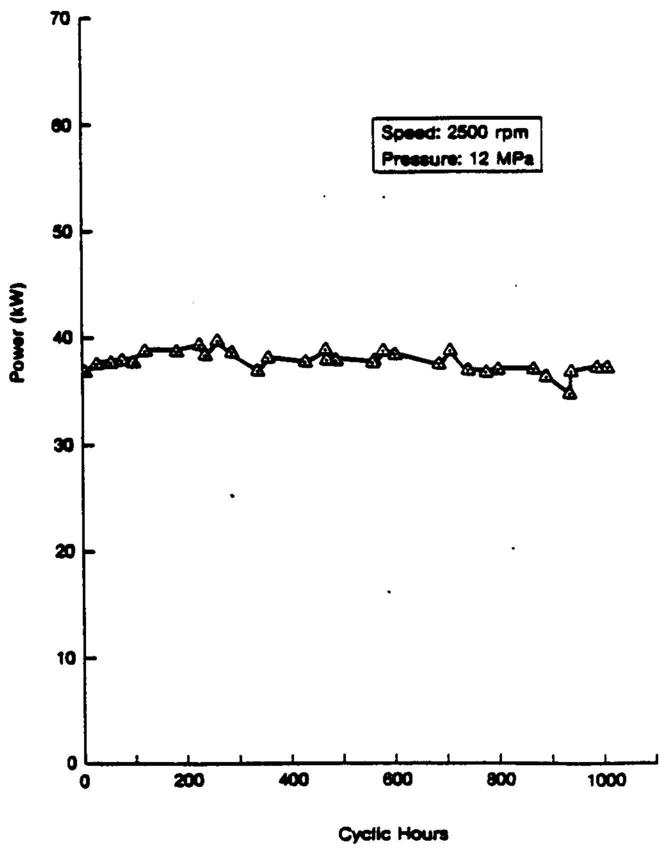

Figure 3-47 Single-Solid Rings in Engine No. 7

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Criteria	PS	MP	TBI	706	151	045	400	Twin 045 or 400
1. Configuration Insensitivity to Nozzle Shifts	P	P	P	P	P	P	P	P
2. Repeatability	F	P	P	P	P	P	P	P
3. Low End Resolution	—	P	P	P	P	P	P	P
4. Response	—	P	P	P	P	P	P	P
5. Expected Endurance	—	?	P	P	P	P	P	P
6. Full Flow Capability	—	—	F	F	F	P	P	P
7. Reasonable Fuel Pressure and Pulsations with Both Full and Low-Flow Capability	—	—	—	—	—	F	F	P

(Column heading: Component)

Key:
P = Pass; F = Fail
PS = Proportional solenoid valve
MP = Micropump
TBI = GM throttle body injection PWM valve
706, 151, 045, 400 = Bosch PWM valves

Figure 3-48 Mod II Fuel Flow Control Selection Matrix

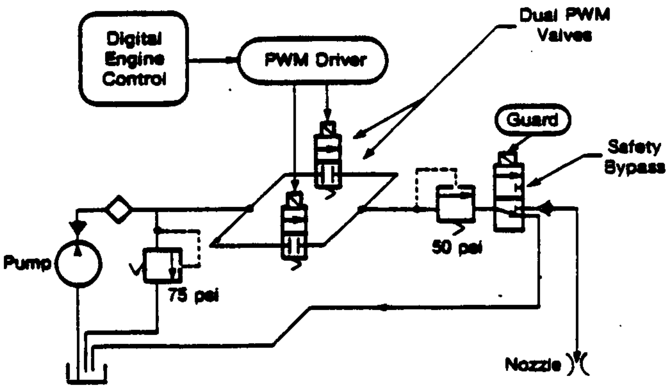

Figure 3-49 Mod II Fuel System Schematic

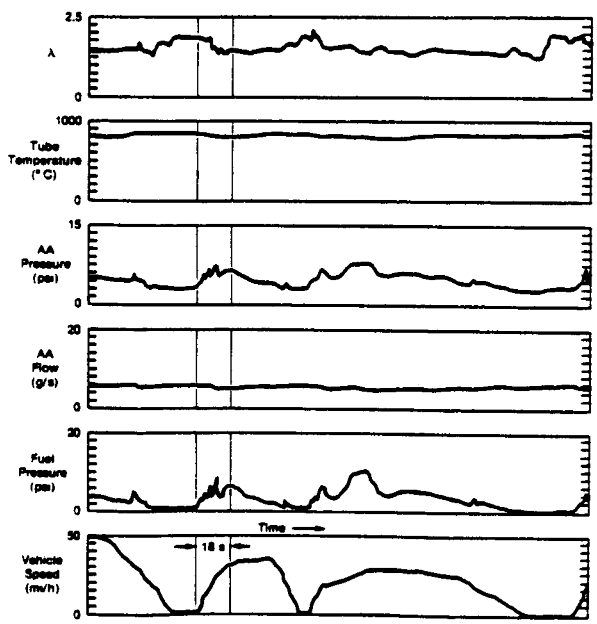

Figure 3-50 CVS Cycle Air and Fuel Flow Variation

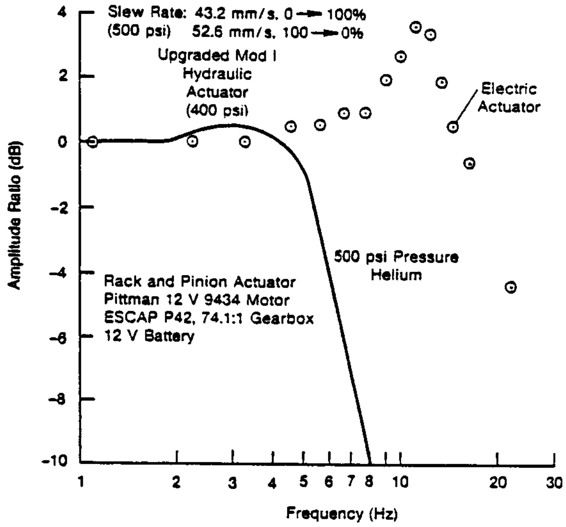

Figure 3-51 Electric Actuator - Small Signal Response

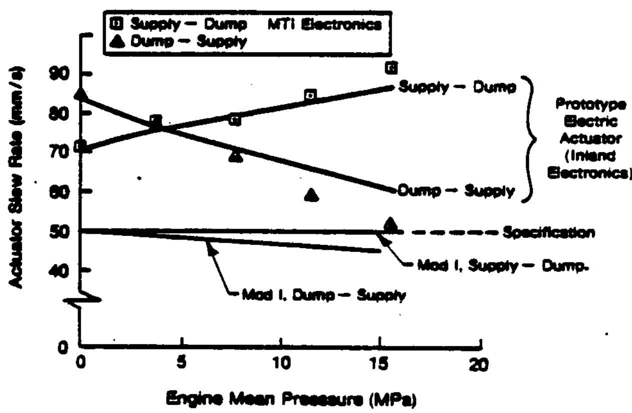

Figure 3-52 Prototype Actuator Slew Rates - Inland Drive Electronics

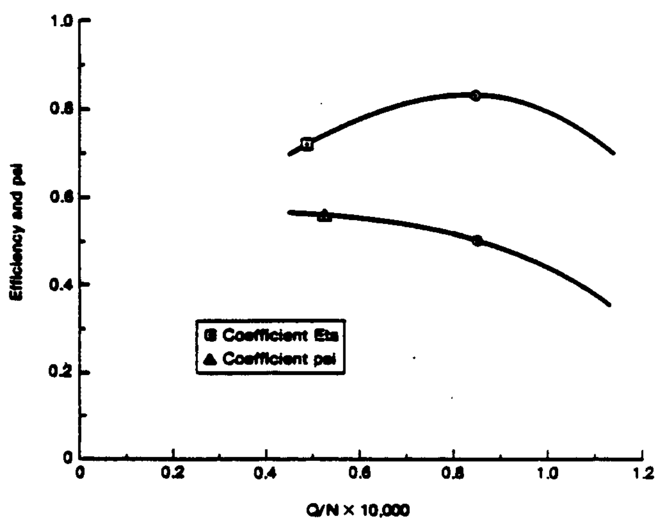

Figure 3-53 Mod II Blower Efficiency and Pressure Head Coefficients

IV. MOD I ENGINE DEVELOPMENT

Mod I Hardware Development

During the first half of 1985, the testing of the Mod I and Upgraded Mod I engines on the program was devoted to a single purpose: the development of technologies related strictly to the Mod II engine. As a result, development of the Mod I engine was curtailed if it was not pertinent to the new engine design. The major effort consisted of development tests concerning:

- CGR combustors
- Ethylene glycol coolant evaluations
- Mod II heater tube/fin evaluation
- Single-solid piston ring evaluation
- Cold start-up tests
- Manual transmission evaluations
- Several controls modifications
- Appendix gap tests.

<u>Mod I Engine Test Program</u>

At midyear 1985, a total of five Mod I engines were operational within the ASE Program. Two of these engines are in the original Mod I configuration and are located at USAB in Sweden. The remaining engines are Upgraded Mod I of which two are located in the United States at MTI and one at USAB. The specific purpose of each engine is listed below:

ASE <u>Engine</u>	
3	Mod I (USAB) EHS development
5	Upgraded Mod I (MTI) general dev.
6	Upgraded Mod I (USAB) Endurance Testing
7	Mod I (USAB) seals/ring dev.
8	Upgraded Mod I (MTI) Transient vehicle performance

The following is an itemized account of the activities of the program engines. During this report period, an additional 2800 operating hours were accumulated bringing the program total engine hours to over 11,000 hrs of Stirling engine operation.

<u>Mod I Engine No. 3</u> - Mod I engine No. 3 accumulated ~60 hrs during the first half of 1985 bringing its total hours to 1870 hrs. During this report period, it continued its designated task of EHS development. As a part of this task, the main effort was the development of the CGR combustor. This combustor being the one selected for use in the Mod II engine.

The specific results of this testing are covered in Section III (Component and Technology Development). Also under development on the engine were fuel nozzles and igniters. Testing on this engine is scheduled to continue through PY 1985. Then, the development effort will be shared on one engine with the seals development activity.

<u>Upgraded Mod I Engine No. 5</u> - Engine No. 5 is located at MTI and is used for general engine development purposes. During this report period it accumulated a total of ~514 hrs bringing its total operation time to 1364 hrs. Testing on the engine was limited to available test cell time.

Testing during this report period involved many different areas. Specifically, they included the following:

- Piston ring evaluations
- Mod II heater tube/fin configuration evaluation
- Ethylene glycol tests
- CGR combustor tests
- Dome/liner appendix gap tests.

Brief descriptions of the results of each test follows.

- Single-Solid Piston Ring Evaluation - Three sets of single-solid piston rings were evaluated initially. The first set ran for 180 hrs before being removed due to high cycle-to-cycle pressure imbalance caused by a dome rub which would have caused some ring damage. The second set did not function well from the start and were removed due to high cycle-to-cycle pressure imbalance. A third set ran well for 130 hrs before being removed to evaluate a set of single-solid rings with increased O.D. to improve cold start-ups. These rings were tested for 114.0 hrs and were removed to evaluate appendix gap effects on engine performance. Further details are available in Section III.

- Mod II Heater Tube/Fin Design Evaluations - An Upgraded Mod I heater head was made into a proposed Mod II tube/fin configuration and evaluated. Testing was conducted to evaluate its heat transfer characteristics. Power was down slightly with this head configuration. Analysis of the results can be found in Section III.

- Ethylene Glycol Coolant Evaluations - The use of ethylene glycol results in a denser cooling fluid with a lower specific heat when compared to water alone. To help understand its effects on engine performance, engine No. 5 was tested back-to-back with straight water and a 50% ethylene glycol mixture in its cooling system. The engine was operated normally with the water in the cooling system, and with ethylene glycol in the system. The same volumetric flow rate of coolant was maintained. By maintaining the same volumetric flow rate the engine's operation in a system in which it pumps its own coolant would be more closely simulated. The difference this causes in mass-flow rate due to density differences is shown in Figure 4-1.

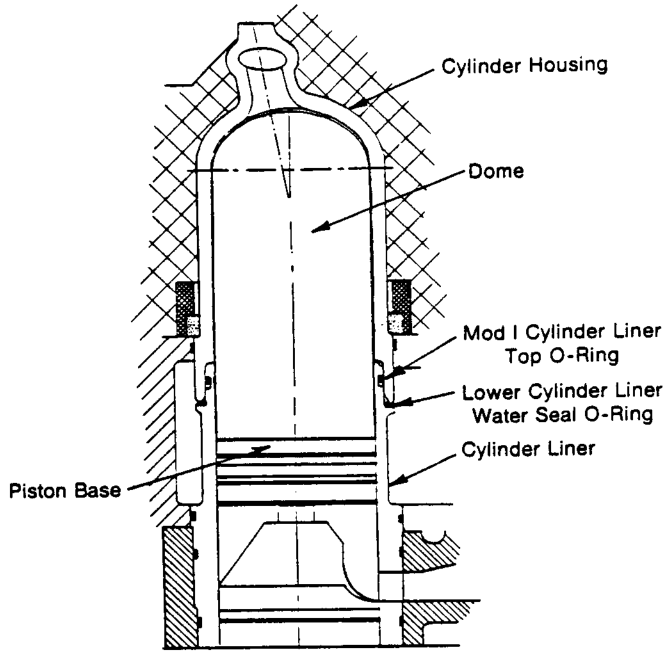

Figure 4-1 Mod I Cylinder Configuration

At all steady-state points with either coolant, the coolant inlet temperature to the engine was maintained at 50°C. The results of the tests are plotted in Figures 4-2* through 4-9. When the parameters power output and coolant heat injection are plotted against Qe with both fluids, it is apparent that virtually little or no difference in performance exists. This would be true as long as the inlet temperature was kept constant. The lower heat rejection to the cooling water is as expected, however, it should have been accompanied by a power reduction. Test results are under analysis.

- CGR Combustor Tests - The 12-tube CGR combustor was evaluated following test rig evaluations in MTI's combustion test rig. Emissions and general combustor performance was measured at two different atomizing

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*Figures can be found at the end of this section beginning on page 4-9.

air pressures. Initial performance of the combustor showed a good low ΔT (<10°C), however, with time the ΔT spread increased from 50° to 110°C. This was reduced significantly by 10 to 70°C, although the spread never returned to its original ΔT performance of 10°C or less by straightening the combustor to improve the ejector to tube alignment. The light-off performance of the combustor was only fair with ∿4 sec for ignition.

A summary of the completed test work is presented in Figures 4-10 through 4-13. Overall the combustor's performance is only fair while its durability is poor. The poor durability causes a decrease in reliability. The use of the bellows joint planned for Mod II or slip point for combustors on engine No. 6 will be examined to improve the combustor's distortion tendencies.

- Appendix Gap Tests - The engine was used for evaluation of the effects of appendix gap reductions on engine performance. The specific data obtained during the test included heater quadrant tube temperatures, regenerator/gas cooler interface temperatures, hot space working gas temperatures, cold space working gas temperatures, and cycle pressure wave versus crank angle. The purpose was also to determine the validity of prediction codes. The results of the test and analysis will be covered in Section V (Mod II Engine Performance Analysis).

Mod I Engine No. 7 - Engine No. 7 accumulated 1100 hrs during this report period for a total of 3403 engine hours. The engine is used exclusively for seals development and the results of the work are included in Section III.

Upgraded Mod I Engine No. 6 - Engine No. 6 accumulated a total of 900 hrs during this report period bringing its total engine hours to 1576. The engine during this report period was used as an endurance engine running at a set temperature of 820°C. The engine is run on a cycle simulating a CVS test cycle. As of the end of June, the engine had accumulated 960 hrs of endurance and will continue until 2000 hrs is reached approximately in November of 1985 at which time it will be torn down for full inspection. The main problems thus far are in the hydrogen compressor and combustor areas.

Upgraded Mod I Engine No. 8 - Engine No. 8 is the MTI-based Spirit vehicle engine used for vehicular transient testing. The engine accumulated ∿250 hrs during this report period and the engine total operating time was 954 hrs.

In preparation for the Mod II configuration selection, several concepts to improve performance from a fuel economy and an acceleration view were evaluated. Also cold start-up tests and CGR combustor evaluations were performed.

Initial performance improvement concepts evaluated were:

- DAFC
- Low idle speed
- Low idle pressure
- 720° versus 820°C operation
- Hot shutdowns between phases 2 and 3 of the CVS test cycle

All of the above were evaluated on a chassis dynamometer for performance characteristics.

The DAFC in its micropump configuration was eliminated early due to poor starts, poor repeatability, and unstable A/F ratio control. The low idle speed setting of 500 versus 750 rpm improved mileage. The hot shutdowns were found to improve mileage (stall the engine instead of normal coastdown). Low idle pressure (30 versus 52 bar) reduced mileage and did not work. Operating at 820°C improved highway mileage, although lowered urban cycle mileage compared to 720°C operation. Careful analysis of the data indicated that the lack of improvement with 30 bar idle and 820°C urban driving cycle fuel economy was due to the Upgraded Mod I compressor design. The Mod II hydrogen compressor and control system, based on the results of this testing, has been designed to provide performance to improve engine transient operation associated with lower pressures desired with low idle conditions and increased set temperature levels.

Late in this report period, additional evaluations were made on the effects of manual or automatic transmission at the 720° and 820°C tube temperatures. In general, it was found that the manual out performed the automatic on accelerations at both tube temperatures (Tables 4-1 and 4-2) due to the faster shifting of the manual transmission. Highway fuel economies were better with the manual transmission at both temperatures, while on the urban cycle the manual was slightly. better at 720°C, however not at 820°C (Tables 4-3 and 4-4).

Vehicle simulation analysis indicated that faster accelerations and improved fuel economy would be achieved with the manual transmission. Small or no difference in urban mileage with the manual transmission is most likely due to the fact that the urban fuel economy on the spirit is not very high (~22 mpg), so that differences are difficult to measure. The higher fuel economy for the Mod II vehicle (33 mpg) will make the automatic versus manual transmission differences more easily measurable.

Also evaluated during this report period was the engines cold start-up capabilities. The tests were conducted outdoors during January and February. Good starts were obtained above freezing (32°F), however, at temperatures lower than that several problems occurred with the vehicle systems. The two main problems which occurred were loss of hydraulics and electronic control malfunction. The automatic transmission fluid (ATF) used as hydraulic fluid was to thick to obtain good hydraulic pressure at the engines control valve. The engine electronics produced random guards shuting the engine down at the cold temperatures. With these two problems present, good engine evaluations could not be made. The problems can be corrected with low temperature hydraulic fluid and a commercial hydraulic pump and low temperature electronics.

CGR combustors were also evaluated in the Spirit vehicle. Details of this testing is found in other sections of this report.

Mod I Hardware Development Follow-Up Report

The following is a report on Mod I hardware development which updates earlier reports given in previous Semiannuals.

Mod I Heater Head Manufacture - The "high temperature" heater heads reported previously on all Upgraded Mod I engines remain in operation. Their operation includes that at 820°C as well as 720°C. Development time includes over 1000 hrs at 820°C on the endurance test engine (engine No. 6) in Sweden. There was no report of deterioration of the fin/tube assemblies on any engine.

Crankcase/Bedplate Cracking - No cracking of the aluminum cases of the reinforced design have been found. All Program engines now have the new design crankcase/bedplate. Over 5500 hrs have been accumulated on the castings since they have been redesigned, and no cracking has occurred.

Heater Tube Failures - No new failures of heater tubes occurred during this report period that could be attributed to normal engine operation. However, severe erosion of several heater tubes did occur, but in this particular instance a ceramic cement used to hold thermocouples in place on heater tubes caused the problem. The cement contained potasium and sulphur which caused "hot corrosion" to occur and severe tube erossion. No evidence of this corrosion has been found on any of present or past heater head tubes.

TABLE 4-1

0-60 MPH ACCELERATIONS WITH MANUAL TRANSMISSION

Shift rpm	Temp. (°C)	Tire Pressure (psi)	Average Time (sec)
2 K	720	45	23.56
3 K	720	45	22.48
4 K	720	45	22.28
2 K	820	45	18.90
3 K	820	45	17.59
4 K	720	45	16.71
3 K	720	30	16.73
4 K	820	30	16.205

TABLE 4-2

0-60 MPH ACCELERATIONS WITH AUTOMATIC TRANSMISSION

Shift rpm	Temp. (°C)	Tire Pressure (psi)	Average Time (sec)
4 K	720	45	30.36
4 K	720	45	28.735
4 K	720	30	28.335
4 K	720	30	28.765
4 K	820	45	22.785
4 K	820	30	22.21
4 K	820	30	20.65
4 K	820	30	19.275

TABLE 4-3
FUEL ECONOMY SUMMARY WITH MANUAL TRANSMISSION

Heater Head Set Temp.	Cold Start Urban $\phi 1$	Cold Start Urban $\phi 2$	Hot Start Urban $\phi 1$	Hot Start Urban $\phi 2$	No Start Urban $\phi 1$	No Start Urban $\phi 2$	Highway Fuel Economy Test
720	19.1	24.8	23.0	23.3	27.5	25.5	40.6
720	18.5	24.2	25.0	25.7	29.4	25.6	40.0
Avg.	18.8	24.5	24.0	24.5	28.45	25.55	40.3
820	15.2	23.8	26.3	24.8	29.7	25.1	42.6
820	17.5	24.3	26.2	24.4	30.2	24.6	41.2
Avg.	16.35	24.05	26.25	24.6	29.95	24.85	41.9

TABLE 4-4
FUEL ECONOMY SUMMARY WITH AUTOMATIC TRANSMISSION

Heater Head Set Temp.	Cold Start Urban $\phi 1$	Cold Start Urban $\phi 2$	Hot Start Urban $\phi 1$	Hot Start Urban $\phi 2$	No Start Urban $\phi 1$	No Start Urban $\phi 2$	Highway Fuel Economy Test
720	17.7	24.3	24.9	24.1	28.0	25.2	38.3
720	17.5	24.2	24.9	24.4	27.9	25.0	38.7
Avg.	17.6	24.25	24.9	24.25	27.95	25.1	38.5
	-1.2	-.25	+.9	-.25	-.5	-.45	-1.8
820~	17.4	24.1	25.6	24.9	28.7	25.0	39.7
820~	16.9	24.2	25.7	24.7	28.8	25.0	40.1
Avg.	17.15	24.15	25.65	24.8	28.75	25.0	39.9
	+.8	+.1	-.6	-1.2	-1.2	+.15	-2.0
770~	17.8	24.3	25.9	24.9	28.6	25.1	38.6
770~	17.5	24.4	26.0	25.5	28.9	25.4	39.8
Avg.	17.65	24.35	25.95	25.2	28.75	25.25	39.2

Cylinder Liner Top O-Ring Deterioration - As previously reported, tests are underway to determine the effects of removal of the lower cylinder liner water seal O-ring on the deterioration problem of the upper (top) O-ring. It was hoped the ring removal would give longer life by allowing water to reach the area of the O-ring and reduce its operating temperature. Tests of this configuration are underway on engine No. 6. It is still to early to tell if it was effective, and testing continues.

Hydrogen Compressor Connecting Rod Small End Bearing- Efforts to reduce the bushing wear in this area have been largely unsuccessful. The effort has involved adding oil feed holes to the area and also evaluating different bushing materials. The effort is continuing along these lines.

Mod I/Upgraded Mod I Engine Performance

Deere Data Analysis

A series of tests were performed at MTI to document several questions that remained open following initial test of this engine at the Deere facility and follow-up testing at MTI that were performed in the previous Semiannual. The testing addressed:

- Effect of a 50/50 glycol/water mixture on engine performance
- Effect of power setting techniques on engine performance
- Comparison of engine performance to MTI initial final acceptance tests and Deere test data
- "Best" engine performance.

It was reported in the previous Semiannual that this engine was tested at Deere as part of the ITEP. Performance levels measured at Deere were significantly degraded relative to MTI final acceptance levels. Follow-up tests at MTI restored engine performance to final acceptance levels via piston ring replacement. The reason for piston ring degradation during the Deere tests is not fully understood.

It is probable that facility-related incidents which occurred during the Deere tests (engine-to-dyno drive shaft imbalance with resulting high vibration and an engine coolant loop failure which resulted in a hot engine shutdown) contributed to the piston ring problem; however, this cannot be ascertained.

Results of the recent MTI tests show the following:

Glycol Mixture Effect - Tests were conducted at engine set temperatures of 720° and 820°C, with water as coolant and again with a 50/50 glycol/water mix. The results of these tests are shown on Figures 4-14 and 4-15. It can be seen that the glycol mixture has essentially no impact on performance at low engine speed and a 1 kW deficit in power at the 3000 rpm point. No signficant difference is seen in η or BSFC (refer to ethylene glycol tests on engine No. 5 previously discussed).

Power Setting Techniques - MTI final acceptance testing set the maximum power line using mean cycle pressure (Pmean) of 15 MPa and a 720°C set temperature reflecting the average of only the quadrant end tube thermocouples. This set temperature reflects an average of $\sim$704°C for all thermocouples. At Deere, power was set differently due to the engine control configuration. The control for the Deere engine sets maximum power with a maximum cycle pressure (Pmax) of 19.5 MPa and 720°C set temperature average of all thermocouples. The 19.5 MPa Pmax is equivalent to a Pmean of 15.4 MPa. Since, both cycle pressure and temperature at Deere were higher than at MTI it would follow that power and efficiency measured at the Deere facility should have been better than MTI final acceptance. Test results (Figure 4-16) show that power should have been higher by $\sim$2 kW with essentially the same efficiency level.

Comparison to Final Acceptance and Deere Testing - Results from the MTI retests are compared to original final acceptance tests on Figure 4-17. Final acceptance tests were run without an engine-driven atomizing air compressor (AAC) so this data was analytically adjusted to correct for its power requirement. Power setting techniques (720°C end tube, 15 MPa Pmean) and coolant media (water) were identical for both tests. It is apparent that the engine performance closely repeats the final acceptance level. Deere data showed a deficit of ~2 kW in power at 2600 rpm relative to final acceptance levels, whereas an improvement of ~2 kW was anticipated due to differences in power setting technique previously mentioned. A comparison of the recent MTI tests to the Deere tests, using identical power setting techniques and cooling media, is shown on Figures 4-18 and 4-19. It is apparent that the engine performance data recorded at Deere is an anomaly compared to recent MTI tests and original final acceptance data.

"Best" Engine Performance - A test was conducted to run a maximum power line through an engine speed range of 100-2600 rpm to document the best achievement power, η, and BSFC capability of the current Upgraded Mod I engine. This test is shown in Figures 4-20 and 4-21. A minimum BSFC level of ~220 g/kW-hr was achieved during this test.

Spirit Vehicle Testing - Upgraded Mod I Engine No. 8

Extensive testing was conducted with the Spirit vehicle during this report period. The testing focused primarily on improving vehicle performance based on short-comings identified as a result of GM evaluation of the vehicle in 1984. In addition, differences in performance at different set temperatures (720°/820°C) and automatic versus manual four-speed transmission were also documented. Results of this testing are summarized in the following sections:

Improved Vehicle Performance - GM testing of the Spirit in 1984 revealed problem areas for the vehicle that penalized its performance relative to a similar IC-powered vehicle. First, the vehicle as tested at its actual weight of 3250 lb had inadequate acceleration. To match industry accepted acceleration levels the vehicle had to be tested at a much lower weight (2500 lb). At this level, fuel economy was not as good as a comparable IC powered vehicle. Idle fuel economy and high cold start fuel consumption penalty (reflected as low urban mileage due to EPA test requirements of cold start for the urban driving cycle test) were cited as the primary reasons for the deficient fuel economy. Several steps were taken to correct these problem areas in this report period. Vehicle acceleration was improved by higher engine power, achieved by higher engine set temperature (820° versus 720°C used during GM tests) and by use of a manual four-speed transmission to take advantage of the Stirling engine's good low speed torque capability. Fuel economy improvements resulting from these changes were further enhanced by optimizing the Spirit final drive ratio and by incorporating modified hot shutdown procedures in the urban driving cycle. During an initial series of tests (January 1985), 0-60 mph acceleration times of ~20 sec were achieved with the improved configuration at a test weight of 3250 lb. Although this represents a significant improvement relative to the 25.2 sec recorded during GM testing, it was not as fast as had been predicted. The problem was traced to a slipping clutch (Figure 4-22). Combined cycle fuel economy during these tests at 3250 lb, was demonstrated to be 29.0 mpg. In June of 1985, these tests were repeated with a heavy-duty clutch/pressure plate assembly to eliminate the clutch slippage problems. Accelerations at the 3250 lb test weight improved to 16.2 sec and mileage remained approximately the same at 28.8 mpg. To illustrate the meaning of these improvements, it is necessary to compare the data on the basis of constant vehicle performance, i.e., same 0-60 mph acceleration.

This comparison is shown on Figure 4-23. This curve shows the 1984 U.S. fleet average combined fuel economy as a function of vheicle test weight, and the test results of the Stirling-powered Spirit. Using the GM testing of the Spirit as a benchmark, with a 0-60 acceleration time of 18.8 sec, the January and June test results were analytically adjusted from the 3250 test weight to those weights that would produce 18.8 sec acceleration times. The fuel economy is adjusted accordingly. The progress made in the 1985 test series is evident. From a fuel economy deficit of 13% relative to U.S. fleet average as tested at GM, an improvement to 21% better than the fleet average was demonstrated. Further improvement can be realized through control system refinement and reduction in idle fuel rate. These techniques will be incorporated in the Mod II engine/vehicle system.

Performance Results - 720° versus 820°C set temperature. Results obtained with this test series are shown on Table 4-5. The following is noteworthy: 1) highway fuel economy improves by √1 mpg as anticipated due to improve engine efficiency at the higher set temperature; 2) urban fuel consumption remains essentially the same, whereas an improvement on the order of 1 mpg was anticipated, with equal idle fuel consumption at the two set temperatures. However, the 820°C set temperature maintained the same idle pressure and speed which results in increased idle fuel rate relative to the 720°C setting. In addition, engine pressure reduction during down-power transients is slower for the 820°C testing, probably due to the lower engine operating pressure at 820°C; and, 3) acceleration rate - a significant reduction in 0-60 acceleration time is achieved at 820°C.

TABLE 4-5

PERFORMANCE IMPACT OF SET TEMPERATURE

		720 °C	820 °C
Manual Transmission			
CSFC	(g)	230	254
HSFC	(g)	60	51
CSP	(g)	133	138
Urban	(mpg)	22.7	22.9
Highway	(mpg)	40.3	41.9
Combined	(mpg)	28.3	28.8
0-60	(sec)	22.3	16.2

The net result of these three factors shows a substantial system benefit for 820°C operation.

Automatic versus Manual Transmission - Results of this testing is shown on Table 4-6. The better match of the manual transmission to the Stirling engine characteristics is evident. 0-60 accelerations are faster, due primarily to preferred gearing for the manual. Highway fuel economy is improved by √2 mpg as was anticipated. Urban fuel economy is relatively unchanged with the manual transmission. With the low Spirit urban fuel economy of √23 mpg, a small improvement would be anticipated, and probably falls within the error band of normal test data. As in the case of 820°C operation, the manual transmission shows a clear benefit for the Stirling engine installation.

TABLE 4-6

PERFORMANCE COMPARISON OF MANUAL VS. AUTOMATIC TRANSMISSION AT 820°C

Manual Transmission		
CSFC	(g)	254
HSFC	(g)	51
CSP	(g)	138
Urban	(mpg)	22.9
Highway	(mpg)	41.9
Combined	(mpg)	28.8
0-60	(sec)	16.2
Automatic Transmission		
CSFC	(g)	258
HSFC	(g)	46
CSP	(g)	137
Urban	(mpg)	22.8
Highway	(mpg)	39.9
Combined	(mpg)	28.2
0-60	(sec)	19.3

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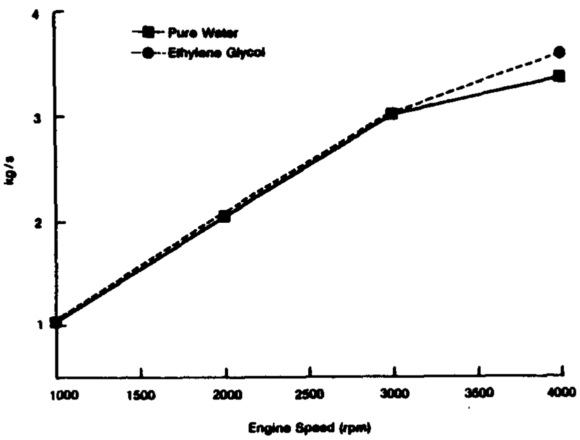

Figure 4-2 Coolant Flow Rates

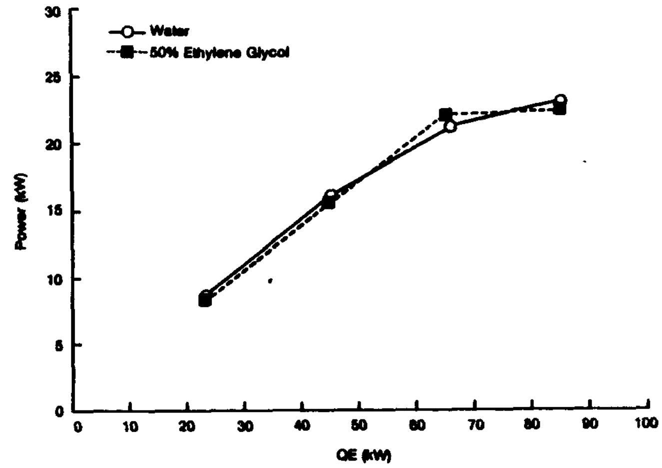

Figure 4-5 Engine Power Levels (7 MPa)
ASE Engine No. 5 (January/February 1985)

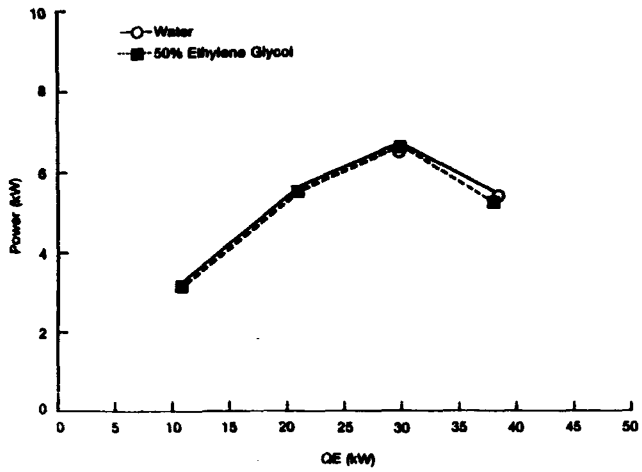

Figure 4-3 Engine Power Levels (3 MPa)

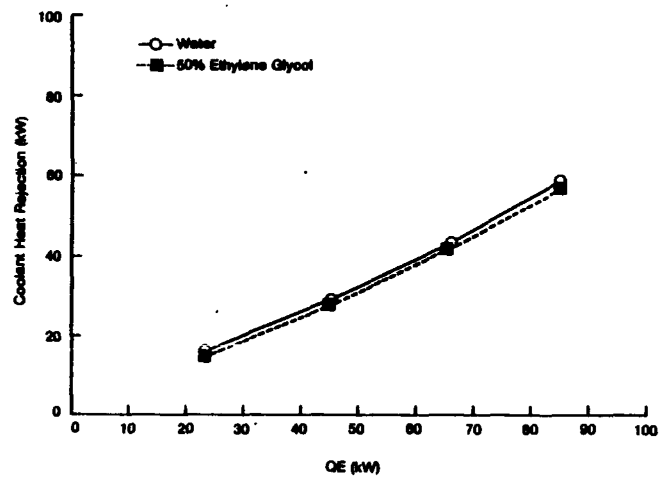

Figure 4-6 Cooling Water Heat Rejection (7 MPa)
ASE Engine No. 5 (January/February 1985)

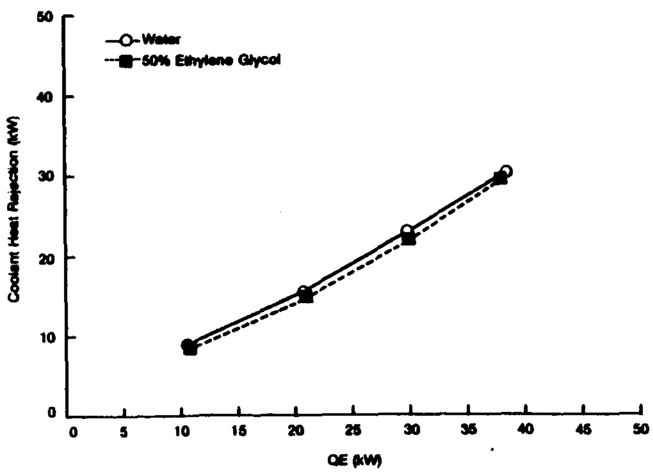

Figure 4-4 Cooling Water Heat Rejection (3 MPa)
ASE Engine No. 5 (January/February 1985)

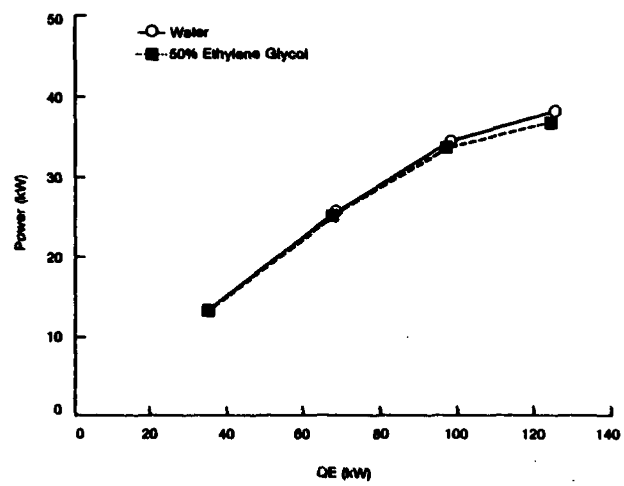

Figure 4-7 Engine Power Levels (11 MPa)
ASE Engine No. 5 (January/February 1985)

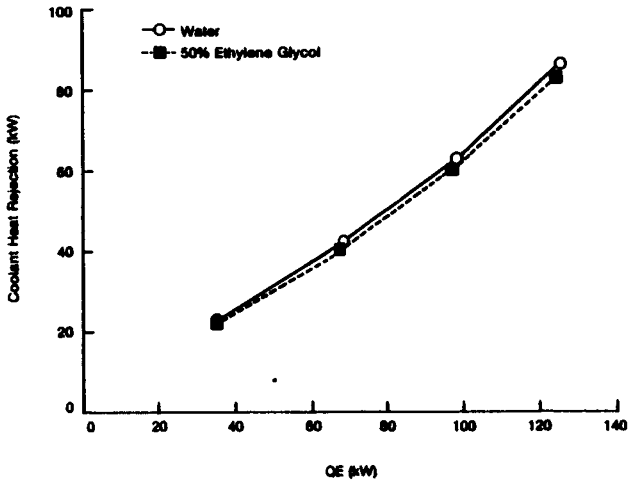

Figure 4-8 Cooling Water Heat Rejection (11 MPa)
ASE Engine No. 5 (January/February 1985)

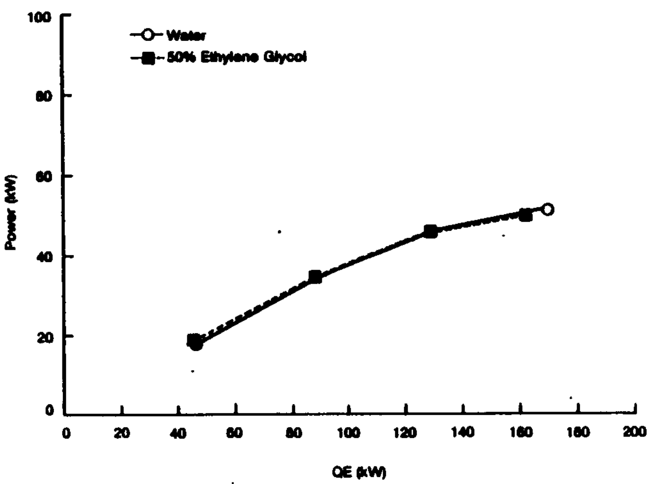

Figure 4-9 Engine Power Levels (15 MPa)
ASE Engine No. 6 (January/February 1985)

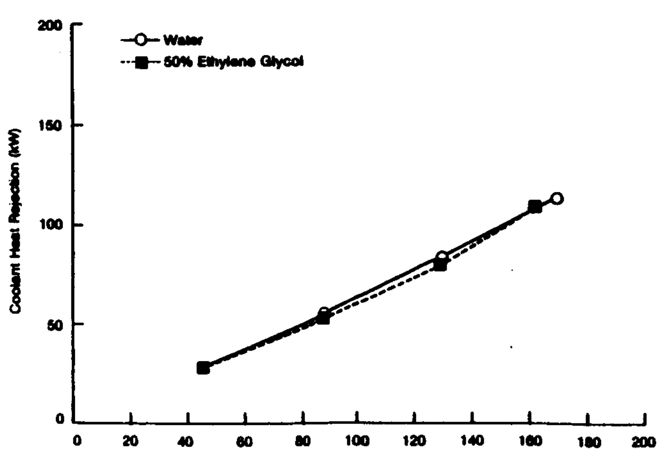

Figure 4-10 Cooling Water Heat Rejection (15 MPa)
ASE Engine No. 5 (January/February 1985)

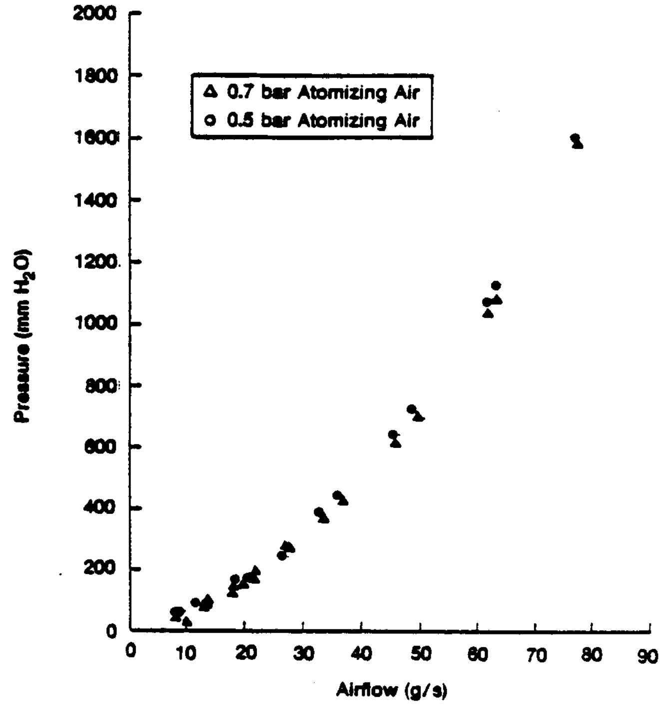

Figure 4-11 Pressure After Blower versus Airflow
(12-Tube CGR Combustor)

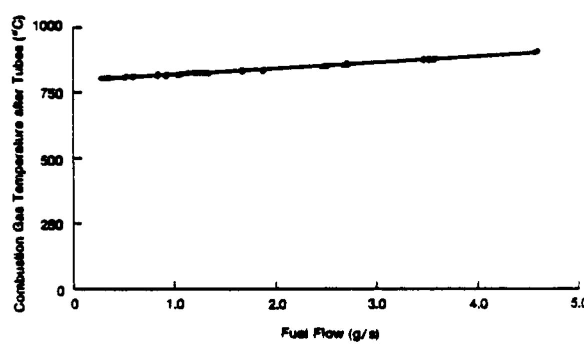

Figure 4-12 Combustion Gas Temperature After
Tube versus Fuel Flow (12-Tube CGR Combustor)

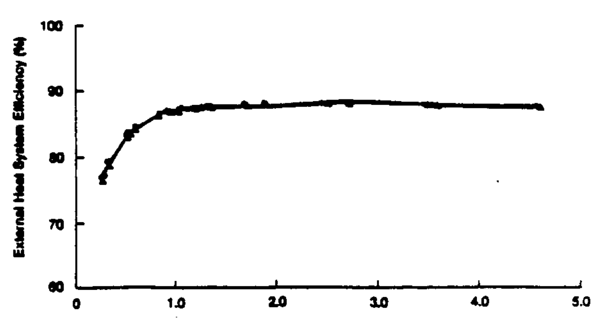

Figure 4-13 EHS Efficiency versus Fuel Flow
(12-Tube CGR Combustor)

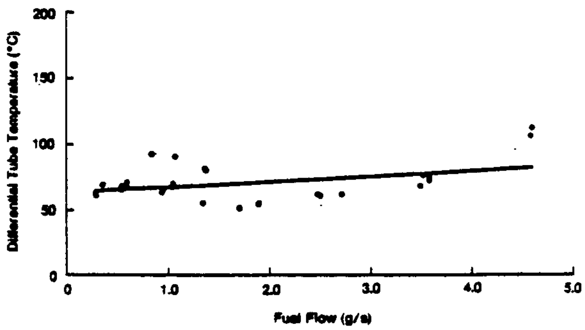

Figure 4-14 Final Differential Tube Temperature versus Fuel Flow (12-Tube CGR Combustor)

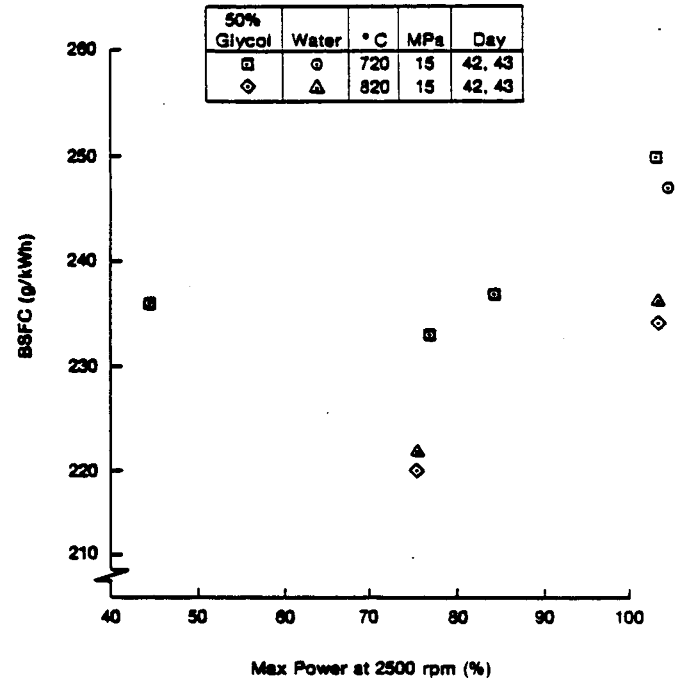

Figure 4-16 ASE Engine No. 9 Water-Glycol Comparison

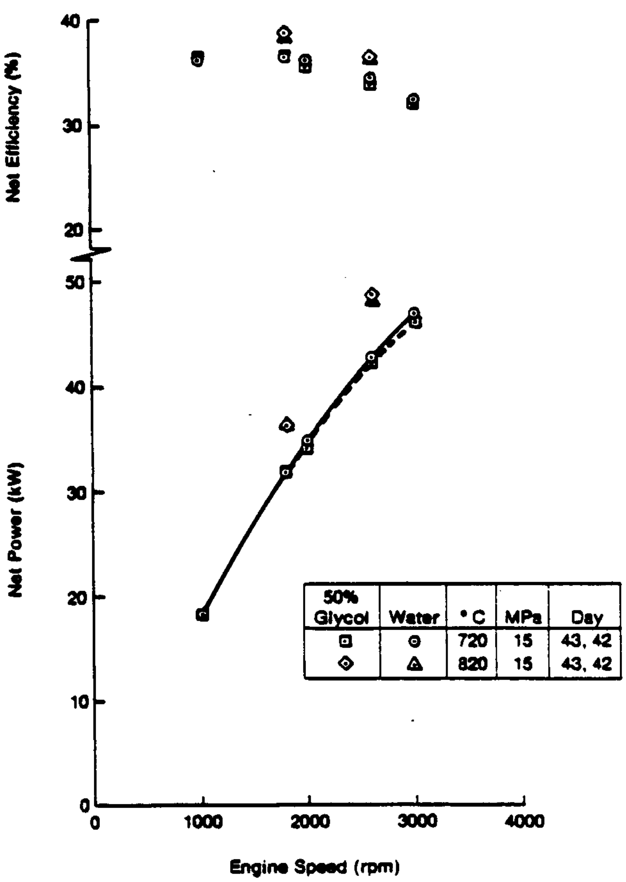

Figure 4-15 ASE Engine No. 9 Water-Glycol Comparison

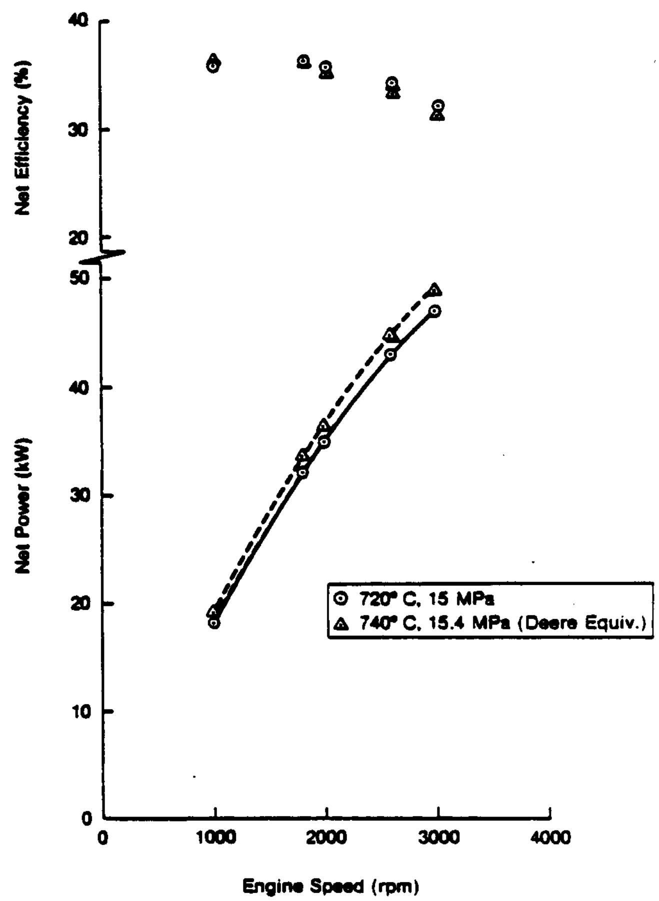

Figure 4-17 ASE Engine No. 9 - H_2O Coolant

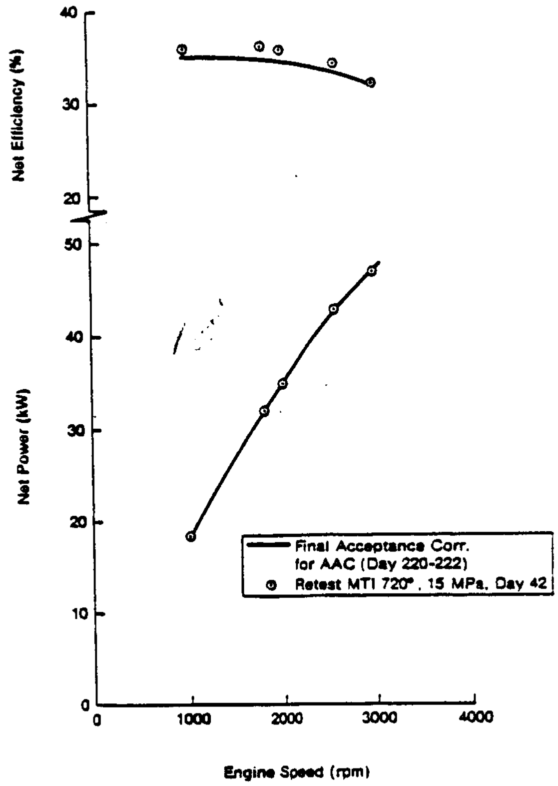

Figure 4-18 ASE Engine No. 9 - H_2O Coolant

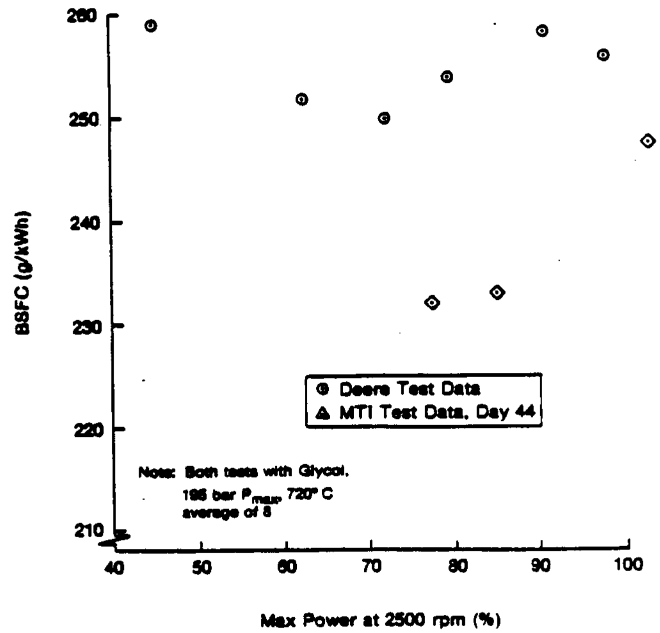

Figure 4-20 ASE Engine No. 9

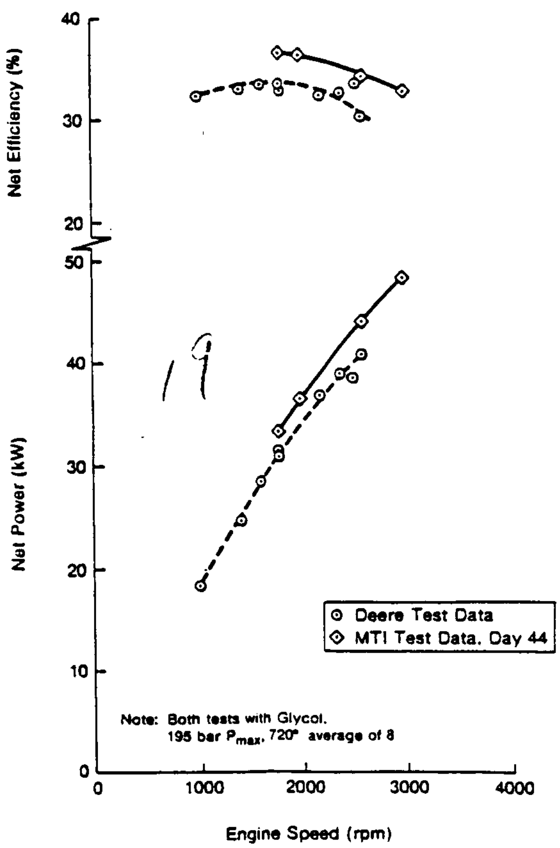

Figure 4-19 ASE Engine No. 9

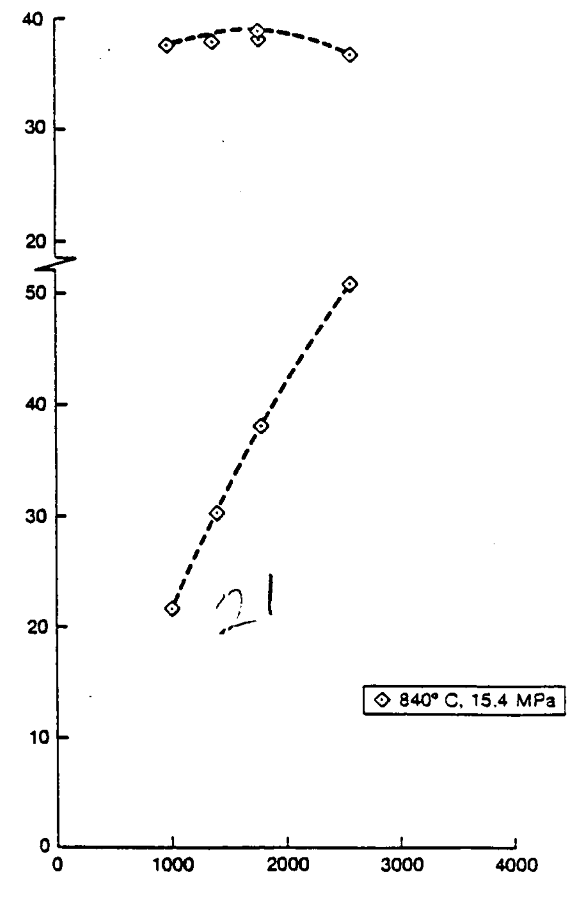

Figure 4-21 ASE Engine No. 9 - 50% Glycol Mixture Power Setting Comparison (Day 44/45)

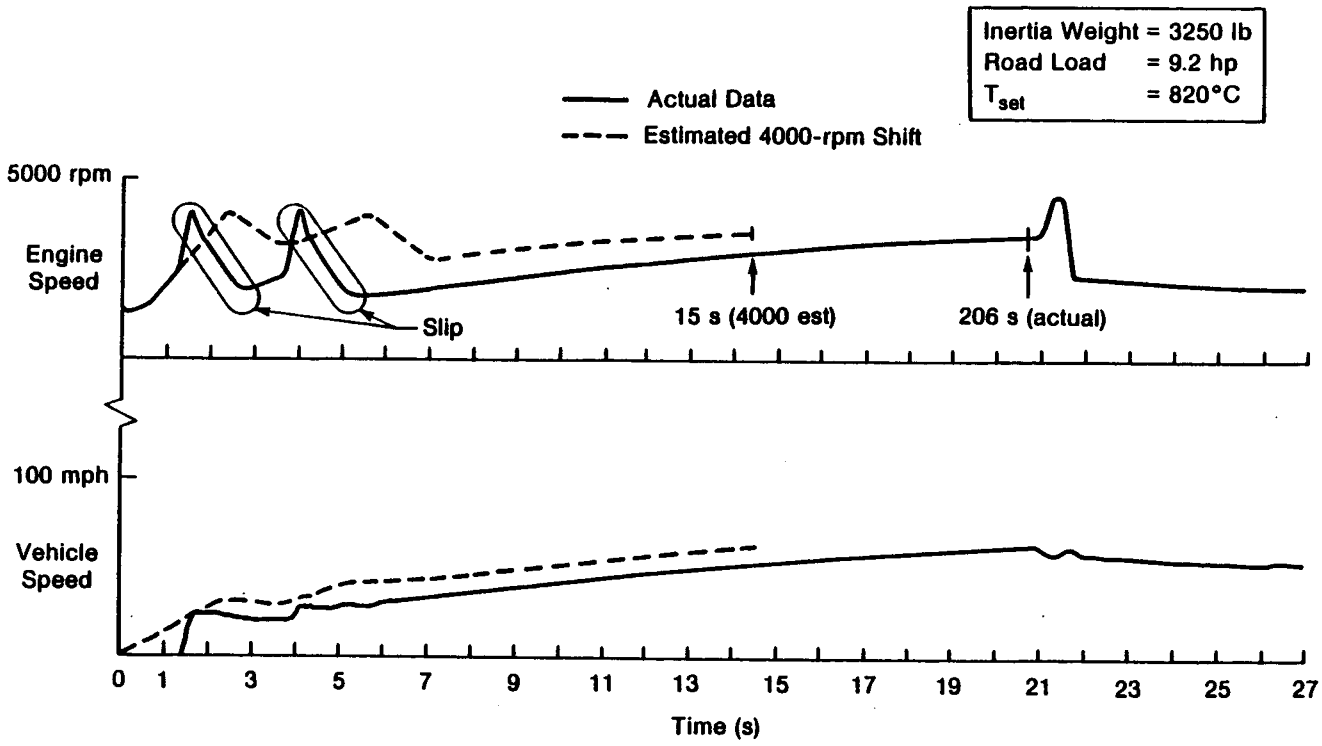

Figure 4-23 Upgraded Mod I Spirit Testing

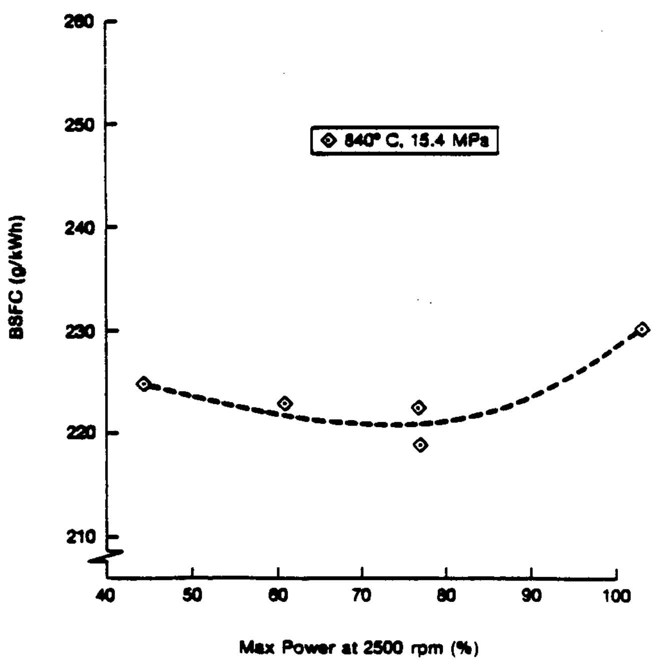

Figure 4-22 ASE Engine No. 9 - 50% Glycol Mixture Power Setting Comparison (Day 44/45)

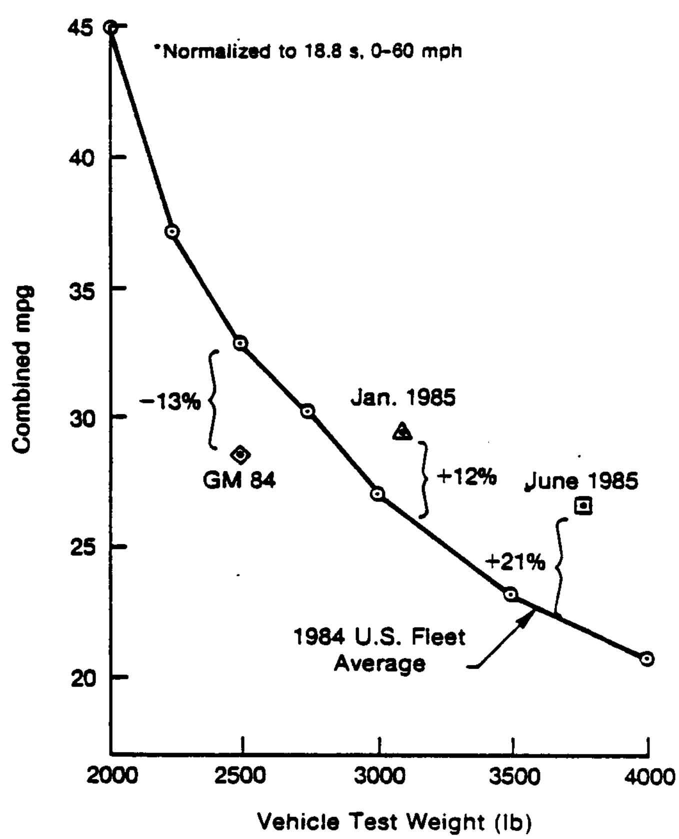

Figure 4-24 Fuel Economy Upgraded Mod I Spirit History

V. MOD II ENGINE DEVELOPMENT

Introduction

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

MTI has the prime responsibility for all phases of Mod II development. Although USAB has been an active participant in the analytical phase of Mod II design, MTI has taken the lead in establishing directions for engine optimization, and has done by far the major portion of detailed design work for the engine. This demonstrates to a substantial extent the achievement of one of the program goals, namely the transfer of Stirling engine technology to the United States. Furthermore, U.S. manufacturers and vendors will be used for most of the Mod II engine hardware, augmenting the technology transfer process.

The Mod II engine/vehicle system was optimized within the constraints of accepted design practice and annular V-block geom[illegible] to achieve maximum possible fuel [illegible] over the EPA driving cycle, while [illegible]ently providing acceptable vehicle acceleration and driveability. Optimized engine geometry was determined through use of MTI and USAB Stirling engine computer codes. However, practical engine design considerations did require some compromises of the optimum geometry, which are reflected in the final detail design drawings.

Performance predictions for the Mod II engine and for the auxiliaries comprising the SES, together with the characteristics of the designated Mod II automobile, were used to predict engine/vehicle performance by use of MTI's Vehicle Simulation Code. Vehicle drive-train characteristics were optimized within the constraints of available production hardware to achieve best possible fuel economy and still meet vehicle acceleration/driveability targets.

A complete set of manufacturing drawings were prepared for the BSE and for most of the control and auxiliary components compromising the SES during this report period. A sequence of evolutionary development of the engine was also defined, encompassing two different heater head designs. The initial build and tests of the engine are to be executed using a two-manifold design with a relatively low-risk procurement cycle. Final achievement of the projected performance goals will be accomplished using a single-manifold heater head for which manufacturing drawings were not completed, and for which the manufacturing techniques were not as well developed.

Mod II Engine Analysis

During this report period, analysis efforts centered on definition and optimization of Mod II engine performance to set the engine design. Basic engine thermodynamic modeling was refined by development of improved analysis techniques addressing the heater head heat transfer and loss mechanisms associated with the appendix gap (space between the side of the piston dome and the cylinder wall). Engine system performance analysis techniques were also improved by more careful modeling of the dynamics associated with transient engine operation in a vehicle installation. A key factor in vehicle fuel consumption is the ability of the engine to pump itself down rapidly as vehicle power demand is decreased, therefore the hydrogen control system was modeled in detail to more accurately represent transient performance. The vehicle simulation modeling was improved to reflect more closely the actual operation of the vehicle by a real driver. The improved engine representation thus achieved was mated with the chosen Mod II vehicle to optimize engine power level and component configuration, and thereby define final system projected performance.

Mod II Vehicle Performance Projections

The Mod II engine will be installed in a 1985 Celebrity vehicle with a manual four-speed transmission. The transmission of choice for the Mod II would be an automatically shifted manual four-speed transmission with no torque converter. Since such a transmission is not currently available, the manual transmission is used to demonstrate performance that would be obtained with the automatically shifted manual.

The MTI vehicle simulation (VESIM) code was used to predict the fuel economy, acceleration, and gradeability performance of the Mod II vehicle system. The predicted Mod II Celebrity performance (summarized in Table 5-1) meets or exceeds all of the applicable program goals.

TABLE 5-1
MOD II VEHICLE FUEL ECONOMY PERFORMANCE

	Spark Ignition	Mod II Prediction	Mod II Goal
Urban without CSP, mpg	-	41.1	-
CSP, g	-	125	-
Urban, mpg	26.3	32.9	-
Highway, mpg	39.6	58.2	-
Combined mileage, mpg	31.0	40.9	40.3
Combined mileage diesel fuel, mpg	-	47.0	-

The complete EPA urban driving schedule is composed of two identical cycles run with different starting conditions. The first cycle begins only after the engine and vehicle were allowed to come to thermal equilibrium at room temperature (20°C) 12 hrs prior to the test. With a cold start like this, a certain amount of fuel energy is required to heat all the hot parts of the engine to their respective operating temperatures. This extra fuel consumption is known as the cold-start fuel consumption (CSFC), and is directly related to the analytically determined stored heat of the engine. The Mod II engine requires 10.9 MJ fuel energy to heat the hot parts. This translates into a Mod II CSFC of 254 g of fuel.

The second cycle follows the first after a 10-min "hot-soak" phase. During this period, the Mod II engine, unlike previous ASE practice, is stopped immediately. A small amount of cooling water is circulated by the water pump to help keep the piston rings and O-rings cool. The only energy lost/used during this time is the conduction through the hot parts into the cooling water, the energy lost due to the temperature equalization in the preheater and the small amount of energy drawn from the battery to power the water pump. Upon start-up, this energy must be repalced into the hot parts, preheater, and battery resulting in a hot-start fuel consumption (HSFC). The Mod II HSFC is calculated to be 28 g.

The EPA test procedures for the urban driving schedule weight the cold-start and hot-start fuel consumptions to determine a single, additional fuel consumption known as the CSP. The EPA assumes that 43% of all vehicle starts will be cold starts and that the remaining 57% will consist of hot starts. This sets the definition of the CSP as follows:

$$\text{CSP} = 0.43\ (\text{CSFC}) + 0.57\ (\text{HSFC})$$

or for Mod II,

$$\text{CSP} = 0.43\ (254) + 0.57\ (28) = 125\ \text{g}$$

Since the fuel economy calculated by VESIM does not take into account any of this transient start-up behavior, the urban fuel economy numbers predicted by VESIM are adjusted for the additional CSP fuel consumption. Because the entire EPA highway schedule is run with a fully stabilized engine, no start-up fuel consumption penalty is needed.

The Mod II engine is projected to have a combined fuel economy of 40.9 mpg. Table 5-1 presents both the uncorrected urban mileage (urban without CSP) and the CSP-adjusted mileage for the final build engine. Also, include for comparison are the fuel economies of the spark-ignition Celebrity with a manual four-speed transaxle. The Mod II goal is 30% above the spark-ignition Celebrity.

The performance of the Mod II vehicle system in the torque-dependent operations of acceleration and gradeability are presented in Table 5-2, compared to the spark-ignition baseline vehicle and the Mod II program goals. The Mod II projections indicate that performance will be better than the baseline and goals.

TABLE 5-2
MOD II VEHICLE ACCELERATION PERFORMANCE

	Spark Ignition	Mod II Prediction	Mod II Goal
0-60 mph, sec	15.0	12.4	15.0
0-30 mph, sec	-	3.8	-
30-50 mph, sec	-	5.0	-
50-70 mph, sec	-	8.6	10.5
0-100 ft, sec	-	4.1	4.5
Gradeability, %	30%	30%	30%

The baseline vehicle represents a considerable challenge to the ASE Program. Using one of GM's most efficient engines, the 3000-lb test weight Celebrity with a manual four-speed transaxle achieves considerably higher fuel economy than the other members of the GM A-body family. With a combined fuel economy (per EPA certification) of 31 mpg, this vehicle is significantly above the fleet average fuel economy for its weight class (see Figure 5-1*). Achieving 30% over this level will place the Stirling Celebrity nearly 50% above the average U.S. vehicle in the 3125-lb inertia test weight class.

Mod II BSE Performance

The Mod II BSE performance was developed as part of an iterative process stemming from engine sizing to meet the Mod II vehicle goals. Once size is selected, the BSE is optimized to provide maximum fuel economy for the automotive installation. Key features optimized for the BSE are described below:

EHS

The EHS consists of the combustor, preheater, and insulation cover. Although not directly a part of the EHS, heat extraction by the heater head becomes an iteractive part of the EHS analysis. Given the characteristics of each individual component of the system, the EHS is then optimized to provide maximum fuel economy in the driving cycle. This is done by optimizing to a figure of merit which combines EHS efficiency (ηB) and CSP. This optimization produced the following results:

The relative calculated mileage based on the EHS figure of merit for the specified vehicle, as a function of the EHS geometry, is shown in Figure 5-2 against the quantity NWL for the different preheaters, where:

- N = Number of preheater plates
- W = Width of the preheater plates
- L = Length of the beaded part of the preheater.

From Figure 5-2, it is seen that the longer preheater gives a higher mileage independent of the plate width.

It is also seen that, for the same heat transfer area, NWL, and, therefore, mass, the longer preheater is better. This is due to the higher longitudinal heat conduction in a shorter preheater. The re-

*Figures can be found at the end of this section beginning on page 5-26.

duction in longitudinal heat conduction when going from 60- to 80-mm beaded lengths more than compensates for the loss in mileage due to the higher pressure drop over the longer and narrower preheater plates. Figure 5-2 also shows that the optimal value of NWL is two for the 80-mm long beaded part. This is equivalent to a preheater width of 25 mm. A preheater with a width of 25 mm will give a high pressure drop in the EHS. Therefore, a slightly wider preheater width of 30 mm is chosen for the Mod II. This gives a small decrease in mileage of $\sim$0.03 mpg.

With the selected EHS geometry, it was determined that full load EHS pressure drop would be rather high ($\sim$ 28 KPa) if an excess A/F ratio (λ) of 1.25 was maintained at the full load point. The 1.25 λ is used to maintain acceptable emissions levels. Since full load is never utilized in the urban or highway driving cycle, it was decided to reduce λ to 1.15 at full load in order to reduce EHS pressure drop to 25 KPa with a resultant decrease in combustion air blower power.

The resultant EHS characteristics are presented on Figures 5-3 through 5-6.

HES

The HES is the heart of the Stirling engine, in that the critical heat transfer occurs in the components of this system, the heater head and the regenerator. The heater head is optimized to provide the desired heat transfer into the engine cycle while minimizing heater tube length (hence engine height) and dead volume, which is detrimental to engine performance. Early Stirling cycle modeling assumed heat transfer into the cycle in a global sense, that is that it occurred through a uniform boundary. The preliminary design of the Mod II and its resultant initial build (Configuration No. 1) incorporates a heater head that has heater tubes on the front row (towards the combustion chamber) connected to the regenerator to minimize the dead volume over the regenerator, shown to be a critical parameter. During Mod II detailed analysis, the heater was modeled as a two-row heat transfer device and then coupled to the Stirling cycle analysis. This modeling showed that, for rear-row set temperatures of 820°C, the front-row temperatures approached 1000°C at low load conditions (idle) with the front row connected to the regenerator. This is not an acceptable operation for the engine. Based on the results of this study a revised heater design was executed whereby all front-row tubes were connected to the cylinder side to eliminate this problem. To minimize dead volume over the regenerator, a one manifold design was adopted, with the regenerator side (rear row) tube connecting directly into the regenerator housing.

The regenerator length and width and piston dome height are also selected in the optimization process. The regenerator sizing is basically a trade-off of pressure loss at high rpm versus axial conduction loss at average operating conditions. The piston dome length is selected to optimize appendix gap losses with minimal engine height while protecting the piston ring from high temperature working gas. For the Mod II analysis, a detailed appendix gap loss model was formulated to analyze losses in the appendix gap. This modeling is described below.

The appendix gap, shown in Figure 5-7, is the annular clearance space between the cylinder and the piston. As the piston reciprocates along the temperature gradient imposed on the cylinder wall, the appendix gap becomes an energy carrier between the temperature extremes in the engine. This results in enhanced irreversible heat transfer from the high-temperature region to the low-temperature region, as explained below. Previous experience indicates that the associated loss is detrimental to the engine performance.

To analyze the problem, a traditional approach usually assumes that gas in the appendix gap takes the instances average temperature of the piston and the cylinder. Under this simplification, the gas dynamics in the appendix gap is omitted in the analysis. With such an approach, the loss mechanism is described as follows. The wall temperatures are at quasi-steady-state and they coincide at the piston midstroke position. At the top position, the piston faces the high-temperature region and receives heat radially from the cylinder. The gas in between acts as a condution medium. At the bottom position of the piston, the process reverses and the heat is transferred radially from the piston back to the cooler cylinder wall. Thus, the radial heat transfer between the piston and the cylinder results in a net axial heat loss (shuttle heat conduction) due to the shuttle motion of the piston.

In general, the shuttle heat loss follows the basic relation:

$$Q \propto (kg/\delta)\ \pi(DS/2)\ (\Delta TS/4L)$$

where:

- Q = Heat transfer rate
- kg = Gas conductivity
- δ = Appendix gap radial dimension
- D = Cylinder bore
- S = Stroke
- ΔT = Temperature difference
- L = Piston length.

The above terms are grouped so that the relation takes the form of Newton's law of cooling, i.e., heat transfer rate = (effect heat transfer coefficient) x (effective heat transfer area) x (effective temperature difference). The proportionality constant varies among researchers depending on the method used and on the type of piston motion. The above relation suggests reasonably that reducing the gap dimension to zero is not practical.

Increasing the gap dimension in previous engine tests, however, did not result in appendix gap loss as is indicated in the above shuttle loss relation. It is now believed that the omitted gas dynamics may be the cause of the discrepancy. This is intuitively clear as the gas can move in and out of the appendix gap with larger gap dimension. The introduced complication to the loss mechanism is described below.

When the piston moves from the hot cylinder region to the cooled cylinder region, the density of the gas in the appendix gap increses due to the heat transfer to the cooler region. This induces additional hot gas flow from the expansion space into the appendix gap. In the return stroke, the process reverses and the gas of lower temperature in the appendix gap is pushed back into the expansion space. Without leakage through the piston ring, the net mass flow through the opening of the appendix gap is zero. As a result of the irreversible heat transfer within the appendix gap, there is a net enthalpy flow (pumping loss) through the top of the appendix gap. The mixing of gases of different temperatures in the expansion space will give rise to additional losses. The problem is further complicated when the pressure and temperature waves in the expansion space are also taken into account.

When the appendix gap dimension increases, the pumping loss may increase accordingly. In addition, as the amount of gas in the gap increases, the thermal capacity of the gas increases and the gas temperature in the appendix gap will not instantaneously take the average of the immediate wall temperatures. Instead, it will depend on the rate of heat transfer with the walls. The heat transfer mechanism may also change from pure conduction mode to convection mode, either laminar or turbulent. When the gas-wall heat transfer is coupled, the cylinder and the piston will have nonlinear temperture distributions. As the leakage through the piston ring is generally present, it will introduce additional losses.

In recognition of the importance of gas dynamics in the present problem, a numerical model was established to estimate the appendix gap loss. The following factors are included:

- Gas enthalpy flow loss (pumping loss)
- Thermal coupling of cylinder, piston, and gas
- Pressure and temperature wave
- Piston speed and form of piston motion
- Nonuniform radial gap dimension
- Leakage through the seal ring.

Figure 5-8 shows the heat flow involved in the calculation of the total appendix gap heat loss.

In the code, each cycle is divided into equal time steps and trial temperatures are used initially. Iterations are then applied for each time step and finally for the cycle to improve the initial guess according to the governing equations described below.

For the gas, the appendix gap length for each time step is divided into many small control volumes. It is assumed that the pressure in the appendix gap is uniform and equals the pressure wave in the expansion space. Under one-dimensional axial flow condition, the equation of state for ideal gas, the continuity equation, and the energy equation are combined into an implicit difference equation for the temperature in each control volume. The convective heat transfer with the cylinder and the piston is recorded and is then used for updating the wall temperatures in the next cyclic iteration. The mass flow is determined as the by-product of the found temperature in the control volume. Thus, the direction of the flow at the appendix gap top may be out of phase with the motion of the piston. The leakage mass flow, however, is determined through the usual orifice slit flow formula.

For the cylinder and the piston, the problem is mainly one-dimensional axial conduction with radial convection on the surface. In most engine designs and operating speeds, the quasi-steady-state assumption that the periodic heat transfer on the wall surface has relatively short wavelength and penetration depth into the wall is valid and is used in the code. As a numerical solution, the code permits position-dependent properties and surface boundary conditions. Because of the thermal coupling, the wall temperature is recalculated after each cyclic iteration.

The usual cyclic-steady test is used to terminate the iteration. Each heat flow component in Figure 5-8 is printed in the output so that a check of conservation of mass and energy can be performed as required. With a reasonably good initial guess by the user, the code usually converges quickly within 15 cyclic iterations. To ensure a sufficient number of control volumes, a user can test the sensitivity and increase the number of control volumes accordingly. Experience shows that 50 control volumes are usually sufficient.

Results from the developed code were implemented into the engine performance prediction code for the Mod II design optimization. The code allows the sensitivity studies performed on such parameters as gap geometry, piston motion, cylinder wall boundary conditions, and leakage effect.

The resultant HES geometry is shown in Table 5-3 for both the first build Mod II and the final optimized build developed with the expanded modeling techniques. It should be noted that significant changes to the Mod II geometry were defined as a result of the improved modeling.

TABLE 5-3
OPTIMIZED HES PARAMETERS

Quantity (Hot Dimensions)	Final Optimum	First Build
Number of heater tubes	25	30
Heater tube diameters, mm	3.04/4.56	2.64/4.16
Heater tube length, mm	361.8	250.5
Regenerator OD, mm	95.2	97.0
Regenerator porosity (%)	65.0	68.0
Regenerator wire diameter, microns	74.0	56.0
Regenerator length, mm	70.1	70.0
Dome gap length, mm	90.0	73.5
Number of hot manifolds	1	2
Cylinder material	HS-31	XF-818

Cold Engine and Drive System - The primary modeling accomplished for this part of the engine was to model the drive unit friction, including rolling element bearings, crosshead losses, oil pump power consumption, rod seal, and piston ring losses. The modeling assumed that split-solid dual piston rings would be utilized in the Mod II, as is the current design. A change to single-solid piston rings, if proven desireable by Mod I/Upgraded Mod I tests, would reduce the drive unit losses. The losses calculated for the Mod II are shown in Figure 5-9.

Mod II Stirling Engine System Performance

The SES is comprised of the BSE plus those items required for the engine to operate as a stand-alone unit. This consists of the following items:

- Combustion air blower
- Working gas compressor
- Alternator
- Coolant pump.

The losses for these components are shown in Figures 5-10 through 5-13. With the characteristics of all engine components defined, the engine system performance was defined for the final Mod II configuration. A table of performance characteristics for that final build is presented on Table 5-4 for selected operating points and an overall performance map is shown on Figure 5-14.

TABLE 5-4

PERFORMANCE CHARACTERISTICS OF FINAL BUILD MOD II ENGINE

PERFORMANCE BREAKDOWN IN FOUR OPERATING POINTS

Full Load Point (p = 15 MPa, n = 4000 rpm)		Part Load Point (p = 12 kW, n = 2000 rpm)	
Indicated Power	78.6 kW	Indicated Power	15.7 kW
Friction	9.9 kW	Friction	2.0 kW
Auxiliaries	6.4 kW	Auxiliaries	1.6 kW
Net Power	62.4 kW	Net Power	12.0 kW
EHS Efficiency	88.9 %	EHS Efficiency	90.4 %
Net Efficiency	18.2 %	Net Efficiency	33.2 %

Maximum Efficiency Point (p = 15 MPa, n = 1200 rpm)		Low Load Point (p = 12 kW, n = 1000 rpm)	
Indicated Power	30.4 kW	Indicated Power	9.1 kW
Friction	2.5 kW	Friction	0.9 kW
Auxiliaries	1.3 kW	Auxiliaries	1.0 kW
Net Power	26.7 kW	Net Power	7.1 kW
EHS Efficiency	91.0 %	EHS Efficiency	88.8 %
Net Efficiency	38.5 %	Net Efficiency	32.3 %

Mod II Vehicle System Performance

One of the key factors required to achieve fully optimized vehicle characteristics is to carefully integrate the engine, drive train, and vehicle as a total package and optimize this package for the actual driving conditions. Engine power level and drive train optimization, for example, addresses the conflicting requirements of acceleration, gradeability, and fuel economy. Additionally, the vehicle reflected should represent a reasonable segment of the automotive market. Finally, vehicle and engine transient operation must be accurately understood to ensure that vehicle performance projections are accurate. The analysis conducted to support the system optimization is presented in the following sections.

Selection of Engine Power Level - Engine power level was selected to give acceleration rates comparable to that of the IC-powered baseline vehicle while maintaining acceptable gradeability and optimum fuel economy. To accomplish this, an optimization matrix of engine power level and final drive ratios was developed.

Complete engine maps were generated for similar engines with maximum power ratings of 65, 60, and 55 kW. Each engine was optimized from the same base geometry to give the desired power with essentially the highest part-load efficiency. The MTI Vehicle Simulation Code then used these maps to predict combined fuel economy (with CSP), 0 to 60 mph acceleration time, and gradeability performance for each of the three engines for a series of axle ratios (4.10, 3.65, 2.84, and 2.66). The results were then plotted as acceleration time versus combined fuel economy for each engine over the range of axle ratios. A normalized version of this graph is shown in Figure 5-15. The minimum axle ratio necessary to satisfy the gradeability requirement for each engine is indicated on each curve. Numerically smaller axle ratios would not meet the gradeability requirement.

Optimization Results - As expected, the higher axle ratios resulted in faster accelerations, but lower fuel economies, and the smaller engines gave higher fuel economies, but slower accelerations. In fact, the benefits of changing the axle ratio gets smaller and smaller as you approach either extreme. At high axle ratios, the engine's maximum speed is reached sooner, forcing the transaxle to be up-shifted, which, in turn, results in a lower vehicle acceleration rate. With a decrease in axle ratio, the torque delivered to the drive wheels is lowered. Gradeability and acceleration performance deteriorate accordingly and fuel economy reaches a limiting value as the engine is forced to run at maximum pressure in order to meet the acceleration demands of the driving cycles.

Axle Ratio Selection - Observing the optimization curves in the region of acceptable gradeability suggests that the best engine size-axle ratio combination for the Mod II vehicle is 60 kW with a 2.84 axle ratio. Dropping to 55 kW requires a minimum axle ratio of 3.05, and results in fuel economy only a fraction above that of the selected combination, but at a significant cost in acceleration performance. Choosing the 65-kW engine at its best fuel economy point (i.e., minimum gradeability) gives somewhat better acceleration performance, but significantly poorer fuel economy. This condition could also be achieved with the 60-kW engine at the appropriate axle ratio.

However, in an attempt to achieve the highest possible fuel economy at the least cost to any of the other performance requirements, the 2.66 axle ratio (with a 60 kW engine) was selected for the Mod II. The impact of this selection on gradeability is to reduce the slope at which acceptable performance is achieved from 30 to 28%. This reduction in gradeability, though, will be restored as the rest of the vehicle system is optimized for this Stirling engine application. The final Mod II predicted power level of ∽62 kW will also assist in the gradeability restoration.

Mod II Vehicle and System Definition - The first consideration in the Mod II design process was vehicle selection.

Earlier Mod II proposed installations used a GM X-body car. For the current Mod II installation, the vehicle selection was reviewed in light of the fact that the X-body car will soon be out of production. The current Mod II engine power level of ∽62 kW indicates that the 3100-3300 lb test weight class vehicle is the proper size for the Mod II installation. The GM A-body vehicle, specifically the Chevrolet Celebrity, was therefore selected for the Mod II installation. The engine compartment is unchanged from the X-body car. Since the Mod II packaging was targeted for the X-body car, this allows the actual Mod II installation aspects to remain unchanged from previous efforts. Further, it was intended that the vehicle selected should represent a reasonable portion of the U.S. automotive market. The Celebrity does, in fact, fit this criteria. First, it is a front wheel drive car, as are the majority of cars sold in the United States. Second, the GM A-body family accounted for 20% of all GM sales in 1984.

In addition to providing a 30% fuel economy improvement over the baseline vehicle, acceleration rates must match those of the baseline vehicle. It is necessary to match acceleration rates in order to ensure that the fuel economy comparison is meaningful.

Artificially high fuel economy can be achieved by installing a low-power engine in any vehicle at the expense of poorer performance.

It should be noted that the engine power level of the Stirling selected to provide acceleration rates comparable to the IC engine is somewhat less than that of the IC (62.3 versus 68.6 kW). The reason for the lower required maximum power is shown on Figure 5-16. Although the IC engine has slightly higher power (and higher torque) at high engine speeds, the Mod II enjoys a sizeable advantage at the lower speeds. The resulting vehicle accelerations are then comparable.

Mod II Engine Transient Optimization - System modeling improvements were accomplished during the Mod II analysis process to enhance the accuracy of predictive techniques. This included a change in the driving cycle model to reflect the actions of a real driver as he/she would anticipate traffic situations and "smooth" the vehicle and engine operation. The computer model existing when the Mod II study began resulted in a very "jerky" engine/vehicle response to exactly match the prescribed EPA velocity trace. The code was modified to smooth the vehicle velocity trace while remaining within the ±2 mph velocity deviation permitted by EPA regulations. This more nearly represents the action that is followed by the driver when actually performing the EPA driving cycle. The second change in modeling was to represent the actual engine performance achieved with a model of the hydrogen control system integrated into the vehicle simulation/engine response model. Previous modeling assumed that engine power (determined by working gas cycle pressure level) would be reduced instantaneously as demanded by the vehicle. To model the actual response, two codes were assembled. The first models actual hydrogen compressor pumping power and performance. The second models this performance mated to the vehicle operation and the total hydrogen system, including the storage system. The modeling resulted in the following analysis.

The Mod II requirements of low engine idle pressure, 820°C operating temperature and superior fuel economy indicate the need for a compressor system capable of rapid engine pump-down rates while consuming a limited amount of power near the idle condition. Rapid pump-down rates allow the engine pressure to be kept closer to the level needed to supply the desired level of power at a given engine speed. If the engine pressure cannot be reduced as quickly as the engine power demand is decreased, excess fuel is consumed and fuel economy suffers.

With the 2.0-MPa, 400 rpm idle condition, the extra shaft power available to drive the hydrogen compressor is decreased dramatically from the Mod I level. Thus, as the compressor works to reduce the engine pressure toward idle, the power available to run it decreases as well. If the compressor consumes too much power, a low-pressure idle condition is prevented. It was for these reasons that several working gas compressor systems were identified and analyzed toward the end of selecting a system which would allow the Mod II to meet its program objectives.

The compressor system, for this analysis, is defined as the engine working gas volume, hydrogen compressor, storage bottle(s) (and their associated plumbing), valves, and filters. Each compressor system was modeled by its characteristic compressor performance (mass flow and power requirement), system volume, and operating logic. These models were incorporated into the VESIM and were then evaluated over the urban and highway driving cycles. The selected system was then chosen on the basis of best combination of performance and risk.

The compressor system simulation was based on the fact that at any given engine speed, the power demand defines the engine pressure. Knowing this, VESIM then uses the compressor system model to check for the next achievable engine pressure. In this manner, the engine pressure in the vehicle code is controlled by the performance of the modeled compressor system.

When an increase in engine pressure is desired, the compressor system model uses the current engine and storage bottle(s) pressures, volumes, and temperatures to determine the maximum engine pressure that can be achieved without any compressor work. The engine pressure is thus limited such that the minimum engine cycle pressure is less than or equal to the bottle pressure.

Under pump-down conditions (i.e., when a decrease in engine pressure is requested), the supplied compressor characteristics are used to determine the next achievable engine pressure. Compressor mass flow and power requirement are represented as functions of compressor speed and displacement, engine pressure, and storage bottle pressure. The power available to run the compressor at any point is defined as the current engine power, less the road load, accessory and acceleration power requirements. If the calculated compressor power requirement is less than the available power, then the next achievable engine power is calculated and used in the vehicle code. If the required compressor power is greater than the available power, then the compressor cannot be used and the engine pressure cannot be decreased.

The system options identified for this investigation were the type of compressor (e.g., single- or double-stage, number of active volumes), number of storage bottles, and type of compressor drive (e.g., slaved to or independent of engine speed).

The existing Mod I compressor system consists of a single-stage, double-acting compressor or fixed displacement that is slaved to engine speed, and a single 7 ℓ sotrage bottle. It was designed for the operating conditions of 720°C engine operating temperatures, and 4.1 MPa mean engine idle pressure. In this situation, the pressure ratio between the engine and the storage bottle (5:1) is acceptable for the single-stage compressor. The relatively high idle pressure does not place tight restrictions on the compressor power or pump-down rate. Figure 5-17 shows the compressor system demand pressure under these conditions, along with the Mod I system's response.

For operation under the Mod II conditions of 820°C and 2.0 MPa idle pressure, the Mod I system is not adequate. The pressure ratio at idle is nearer to 10:1, which is too great for the single-stage compressor. The low idle pressure results in both a demanding pump-down rate (see Figure 5-18) and limited available power near idle. The shaded areas represent excess engine power relative to vehicle demand, and hence, wasted fuel.

This high-pressure ratio resulting from using the single storage bottle with the low idle engine pressure seems to indicate the need for a two-stage compressor. When investigated, however, it was found that a two-stage compressor to meet the Mod II's needs would be subject to high, nonreversing rod loads and would still have a high power requirement near idle. The latter would result in limited capacity and also a limit on pump-down rate.

By going to two storage bottles, the maximum pressure ratio can be cut in half, allowing for a return to a single-stage compressor. Rod loads and power requirements are similarly reduced, while the total mass of working gas in the system can be maintained at approximately the same level as with the single bottle.

The principles of operation for the two storage bottle system are illustrated in Figure 5-19. One of the storage bottles is initially charged to a relatively high pressure (∼20 MPa), while the other bottle starts at about half that pressure. (The sensitivity of system performance to initial bottle pressures was found to be slight, so the actual initialization pressures will be determined through hardware testing.) When an increase in engine pressure is desired, the engine presusre is first supplied from the low-pressure bottle. This continues until the maximum cycle engine pressure equals the low bottle pressure. The remainder of the working gas is supplied from the higher pressure bottle.

Under the pump-down conditions, the high bottle is first pumped up to its full pressure. After that, the compressor pumps into the lower pressure bottle until the idle pressure is achieved.

Although the use of a single-stage compressor does provide the benefits of acceptable, reversing rod loads, and it is part of the current ASE Program technology base, there is still a limit on the available power near idle that tends to place restrictions on the compressor capacity, and therefore on the pump-down rate. The solution to this dilema is to use a variable-capacity compressor. This would allow for high pump-down rates when the power was available, yet it would also provide for achievable low idle pressures. To obtain a variable-capacity compressor, two approaches were identified; a compressor with fixed displacement, but independent (of engine speed) and variable speed control, or a compressor slaved to engine speed, but with variable dispacement.

A fixed displacement, variable-speed compressor would be operated such that compressor speed would be continually adjusted so that the compressor power requirement during pump-down equaled the available power. An electric drive was investigated for this purpose, but it was determined that a very large motor would be needed, and that the additional electric demand would be too great for the designed Mod II alternator. Upscaling the alternator size would adversely affect system efficiency.

The alternative drive system identified for the compressor system was a hydrostatic transmission. This option appeared attractive, until it was determined that the speed ratio needed for the Mod II application (5:1 speed-up) is outside current technology and applications, and that the maximum torque transmission would require a large, heavy (45-50 kg) hydrostatic pump-motor combination.

A direct drive (slaved to engine speed) compressor with a variable capacity would be operated by adjusting the displacement such that the compressor power requirement during pump-down was less than or equal to the available power. One way of obtaining variable capacity would be to use multiple fixed displacement compressors, and just control the number of compressors in use. This method, however, would introduce extra bearings and seals and their associated losses, added weight and packaging space, and increase round-pumping losses.

The selected option involves a single compressor that has three active volumes, as shown in Figure 5-20. Each volume can be independently short-circuited giving a total of seven discrete pumping modes.

The selection of the volumes for each of the three chambers was determined through an optimization procedure. The idle power availability serves to set the minimum volume. This volume is sized such that the compressor power at the idle point (2.0 MPa mean engine pressure, 400 rpm compressor speed, and ∽10 MPa low bottle pressure) is just below the available idle power. If the total capacity of the compressor (all volumes active) is varied, the intermediate volumes will fall out due to the geometric relationships among the three coaxial volumes. The sensitivity of fuel economy to the maximum compressor capacity is shown in Figure 5-21 for a three-volume compressor that uses only three (I, III, and VII) of the seven modes available. (The urban-505 driving cycle represents only the first phase of the two-phase urban driving cycle.) The benefit of using all seven modes is also shown in this figure. The final volume selection is shown in Table 5-5. Note that two pairs of modes (II-III, and IV-V) have essentially the same displacement. This results in a Mod II compressor which has three active volumes and five discrete capacities. Figure 5-22 shows how this system responds to the demanded engine pressure during the first two minutes of the urban driving cycle. Note the considerable reduction in wasted fuel achieved with this system relative to the Mod I system shown on Figure 5-18. This represents an improvement in fuel economy of ~4 mpg that was achieved for the Mod II by optimizing the hydrogen control system.

TABLE 5-5
MOD II COMPRESSOR

Three Volumes

1) VA = 12.0 cc
2) VB = 12.6 cc
3) VC = 4.0 cc

Seven Modes (Five Usable Modes)

Mode	I	II	III	IV	V	VI	VII
Displacement (cc)	4.0	12.6	12.0	16.6	16.0	24.6	28.6

Initial Engine Design

Initial design of the BSE was completed during this report period and concluded with the BSE Design Review presentation to NASA on April 2-3, 1985. The design of the SES and of the second heater head was also virtually completed during this report period, concluding with an SES Design Review presentation to NASA early in August 1985. The SES design is presented in this Semiannual report to provide an overall description of the entire Mod II engine design in a single document. A complete and detailed description of the BSE design is given in MTI 85TR47, entitled, "Mod II Basic Stirling Engine Design Review Report and Presentation". A detailed description of the SES design is given in MTI 85P65, entitled, "Mod II Stirling Engine System Design Review Report and presentation".

EHS Design - The EHS consists of the combustor, fuel nozzle, igniter, air preheater with inlet air and exhaust manifolds, the insulation cover, and the flamestone. The complete assembly of the EHS is shown in Figure 5-23.

Combustor - The combustor is a CGR type, as opposed to the EGR type used in Mod I engines. WIth CGR, a portion of the combustion gases is extracted from the flow stream immediately downstream from the finned heater head tubes and mixed with incoming combustion air. In an EGR combustor, the recirculated gas is taken from the exhaust pipe, after the flow has gone through the preheater and given up heat to the incoming flow.

The 12-tube, TTE configuration was selected for the Mod II engine. Figure 5-24 illustrates the radial/tangential orientation of the ejectors and mixing tubes of the combustor. It is apparent in Figure 5-24 that the tangential tube orientation at the inner circle of the combustor will impart a high swirl velocity to air entering the combustion region near the fuel nozzle. At the OD of the combustor, all of the incoming primary air from the preheater must pass through the 12 ejector nozzles. These nozzles impose the major pressure drop in the air/exhaust flow circuit, ~9.4 KPa at maximum power.

The mixing tubes are welded at their outer ends to a liner ring, and are supported at their inner ends by an inner support ring, through which the tubes are free to slide to accommodate thermal expansion. The combustor outer wall or liner is made of 0.4 mm thick Inconel 600, embossed with an all-over random pattern to produce an effective thickness of 0.78 mm. In addition, the top face is formed to a spherical surface, with 24 radial stiffening ribs stamped in the sheet metal. These design features are included to provide the greatest possible resistance to buckling, while keeping weight to a minimum. The inner tube support ring is secured to the top or outer liner with with 12 tabs inserted through slots in the liner, and bent over. The liner ring at the outer ends of the tube has an upper flange which is spot-welded to the upper liner. The ejector nozzles are inserted through support clips and holes in the outer liner and seal-welded to the liner wall. The welded assemblies are made in support fixtures that hold the separate pieces in correct alignment as they are welded. All components except the outer liner are made of 0.6 thick 310 stainless steel.

A flange on the bottom of the outer liner is clamped and sealed to a mating support flange on the preheater. An insulated neck at the center of the outer liner extends up through the insulation cover and carries a flange to support the fuel nozzle. The joint between the combustor liner and insulation cover is sealed with a stainless steel bellows, which allows relative motion between the two parts due to thermal expansion. The support ring at the outer ends of the combustor mixing tubes extends downward and is supported in a seal rail at the top of the finned, rear-row, heater head tubes. This forms a seal preventing the combustion gases from bypassing over the top of the finned tubes.

Fuel Nozzle and Igniter - The BOM nozzle design from the Mod I engine was selected for the Mod II engine. This design mixes fuel and atomizing air inside the nozzle body, and has 12 outlet spray orifices. The Mod II version of the design, shown in Figure 5-25, incorporates some significant changes compared to the Mod I version. The Mod I nozzle assembly incorporated two radial press fits between concentric components which must seal tightly. These fits are disassembled when the nozzle is cleaned. When re-assembled, fuel leakage sometimes occurs at these metal-to-metal interfaces, which causes coke buildup on the nozzle tip. In the Mod II nozzle, one press fit was eliminated by combining the spray-nozzle tip with the nozzle body in a single piece. The other radial press fit was replaced with a conical, axially compressed fit between the swirler sleeve and the inner bottom face of the spray-nozzle tip. Axial clamping pressure on this conical fit is imposed by three spring washers and a threaded retainer cap, which clamps all the nozzle components into the body. A lock-tab washer is included to prevent loosening of the retainer cap. The integrated body/spray-nozzle-tip and the swirler piece are made of 310 stainless steel. The insert that carries the O-ring seals and the four fuel pipes is made of 304 stainless steel. The threaded retainer cap and lock-tab washer are carbon steel with a zinc-chromate, corrosion-resistant coating.

The internal features and geometry of the nozzle which accomplish the mixing of fuel and atomizing air were retained, unchanged from the Mod I configuration. The body of the nozzle contains internally threaded holes for connection of fuel and atomizing air supply tubes. The body also incorporates a two-bolt flange for securing the nozzle to the flange on top of the combustor liner.

The nozzle is also designed to receive an igniter assembly, installed in a center hole through the nozzle. This igniter location was selected in preference to a separate, side igniter to avoid the need for a second, bellows-sealed penetration of the insulation cover and combustor liner. The igniter consists of a Kanthal center electrode with nickel connection-terminal, epoxied into an aluminum-oxide ceramic insulation sleeve. This sleeve is, in turn, epoxied into a Torlon-plastic, threaded mounting nut. This nut assembles into tapped hole in the retainer cap of the fuel nozzle, to hold the center-electrode assembly in place, electrically insulated from the nozzle body. The center sleeve of the fuel nozzle swirler is extended to form a concentric, outer electrode for the igniter.

Preheater and Air Manifold Assembly - The Mod II air preheater is a metallic plate-type, counter-flow heat exchanger similar to that used on Mod I and Upgraded Mod I engines. The plate design was optimized to achieve highest possible fuel mileage for the Mod II in automotive application. The plate configuration is shown in Figure 5-26.

As compared to the Upgraded Mod I, plate thickness is the same (0.1 mm) but the plates are longer and narrower. This decreases axial heat conduction loss and hot mass, at the expense of higher pressure drop. Control of inter-plate spacing is improved by closer spacing of dimples, and by using deeper dimples that seat against the flat surface of the adjacent plate, rather than on another dimple. The corrugation contours were modified to improve formability. Mod II plate material is alloy 253 MA, which has better oxidation resistance than the 310 stainless steel used in the Mod I preheater.

The annular matrix of plates is made in the same manner as that of the Mod I engine. The matrix is encased between inner and outer sheet metal sleeves or walls that are seal welded to the matrix. An annular inlet air manifold is welded to the bottom of the matrix. Immediately above this is the annular exhaust gas manifold. Each manifold has two gas-flow connection ports 180° apart, to provide circumferentially uniform flow distribution through the matrix. The inlet and exhaust connections are separated 90° from one another to facilitate connections of pipes and hoses.

A conical ring welded to the inner wall of the preheater reaches inward and down to seat in the lower seal rail at the bottom of the finned, rear-row heater-head tubes. This seal prevents combustion gases from bypassing the rear-row tubes by flowing underneath the tube manifolds to the preheater exhaust inlet. A second seal ring at the top of the inner wall of the preheater supports the combustor outer liner. The combustor and preheater flanges are held together by a ring-clamp, with a braided-packing seal. The sheet metal parts of the preheater/manifold assembly are 310 stainless steel where higher temperatures are encountered, and 304 stainless steel where temperatures are lower.

The bottom of the assembly mounts on an aluminum sheet-metal transition piece. The round preheater/manifold assembly clamps to the transition piece with a tapered-ring clamp, which envelops a large O-ring seal. The lower part of the transition piece adapts to the upper surface of the engine block. Another tapered-ring clamp and O-ring seal are employed at the top of the manifold assembly, to hold the insulation cover in place.

Insulation Cover - The insulation cover forms a gas-tight shell over the preheater/combustor assembly with thermal insulation to minimize heat loss. The cover is made up of an outer, 1.0 mm thick aluminum casing and an inner 0.4 mm thick casing of 310 stainless steel. Held between the two casings is a 40 mm thick layer of Kaowool ceramic-fiber, needled blanket insulation. This material was chosen in preference to Microtherm insulation due to its better durability and lower density. The higher density of Microtherm would increase CSP and reduce fuel mileage.

Both the inner and outer casings are formed to spherical contours on the top surface. In addition, radial stiffening ribs are stamped into the spherical top surfaces to increase the stiffness and resistance to buckling.

At the center of the cover a flanged sleeve extends through the aluminum casing and insulation, to be welded to the stainless inner casing. This seals the insulation in a closed chamber. Inside and concentric to this sleeve, a stainless steel bellows extends from a welded connection at the inner casing, upward to a flange surface at the level of the outer casing. The flange surface of the bellows is clamped between the top of the combustor neck and the fuel nozzle. This bellows forms a gas-tight seal between the cover and combustor, which can accommodate some vertical motion of the combustor relative to the insulation cover. This arrangement preserves the location of the fuel nozzle with respect to the combustor, despite relative motion between combustor and insulation cover.

Flamestone - The flamestone, shown in Figure 5-27, is the bottom wall of the combustion chamber, and an insulation barrier to protect the engine block from combustor radiant heat. In addition, the flamestone forms a barrier to combustion gas flow and prevents or minimizes bypass gas leakage past the front-row heater tubes. The design selected for the Mod II consists of a contoured disk of Kaowool ceramic-fiber insulation board, supported by a thin "pie-plate" of 310 stainless steel. The support plate is fastened to a metal post with radial gussets, which extends downward to an attachment point at the center of the engine block. The attachment incorporates a spring clip and conical gripping surface which can maintain a downward pull over 3 or 4 mm of travel.

The Kaowool board is protected from erosion by a 0.3 mm thick covering of Nextel 312 ceramic-fiber fabric. The fabric is stretched over the Kaowool disk and clamped in place between the side wall of the pie-plate and sheet-metal clamping pieces. These clamping sectors have an outer rim which seats on the inner manifold of the heater head housing. This provides a locating support surface and seal, forcing combustion gas to flow through the front-row tubes.

HES Design - The HES includes the heater head tube array, the heater head housing or pressure vessel, and the regenerator. Two different configurations of the HES were designed for the Mod II engine, and both will be built and tested. Configuration No. 1 incorporates a two-manifold heater head housing, investment cast of XF-818 alloy, with CG-27 tubes (Inconel 625 will be be used for development purposes). The other design (designated configuration No. 3) uses a one-manifold heater head housing cast in Haynes Stellite 31, with Inconel 625 tubes.

Configuration No. 1 Heater Head - Configuration No. 1 HES design is illustrated in Figure 5-28, which shows one of the four quadrants comprising the entire HES. The cast pressure-vessel/housing envelops both the cylinder or expansion space, and the annular regenerator which wraps around the periphery of the cylinder. A nonpressure-bearing partition wall defines the cylinder and separates it from the annular regenerator. Two manifolds, one connected to the expansion space and one to the regenerator, are attached to the main body of the housing by a concentric neck. The two manifolds are arranged as inner and outer concentric quadrant areas, so that four housings assembled on the engine block produce two complete concentric manifold rings. This effect is illustrated in Figure 5-29. Two different castings are required. One type for cylinders No. 1 and 3, a second for cylinders No. 2 and 4.

Thirty hairpin tubes interconnect the manifolds of each quadrant. Tubes are 4.1 mm OD with 0.75 wall thickness. The rear tube leg is straight and has 85 fins brazed in a vertical array. The tube/fin arrangement is shown in Figure 5-30. Fins are 0.6 mm thick, made of Inconel 600, and spaced 0.5 mm apart by tabs and dimples formed in the fin. The fins encompass only one tube, unlike Mod I fins, which extend over three tubes.

The braze joints between fins and tubes are done with Nicrobraz LC alloy, deposited by slurry-dip process. Brazing of tubes into the manifolds is done with Nicrobraz 150 alloy. Both brazing operations are done in a single heat in a vacuum brazing furnace. Four quadrants are assembled in a fixture which holds the heater head housings in proper orientation and clamps all the tubes in proper location to form the circular cage of tubes. The complete assembly and fixture is placed in the furnace for brazing.

As shown in Figure 5-30, a top rail is mounted at the top of the finned length of the rear-row tubes. This rail is slotted for each tube, and maintains tube spacing at the top. The rail interfaces with the combustor inner liner to form a seal and block combustion gases from leaking over the top of the finned tubes. A seal rail is also installed at the bottom of the finned tubes, tack welded and brazed to the outer manifold of the housing. This rail interfaces with the seal ring extending from the preheater.

Thorough stress analyses were made of the tubes and housing casting, considering fatigue, creep, primary and secondary stresses. Figure 5-31 is a representative finite-element model of the heater head housing employed for analysis. Table 5-6 presents a summary of the required safety factors (FR) and the safety factors determined by analysis (Fo). A comparison of these two values indicates overstrength values ranging from 10 to 26% for four of the stress criteria. The remaining three criteria meet the required safety factors.

Configuration No. 1 Regenerator - The annular regenerator is made of wire mesh screens stamped in a doughnut shape, stacked and compressed in a fixture, and vacuum-sintered into an annular biscuit. A course screen is included at each end of the matrix stack to provide strength and support. The regenerator has a radial width of 11.4 mm and a height of 69.1 mm. The matrix material is 304 stainless steel, and has a nominal porosity of 68%.

The partition-wall sleeve is plated on its outer surface with nickel-phos brazing alloy and then inserted into the regenerator matrix. A small seal ring is also mounted on the bottom OD of the matrix. A stuffer piece is mounted on top of the regenerator, piloting on the upper end of the partition wall. This entire assembly is vacuum brazed together in a single brazing operation. The inner surfaces of the partition wall and stuffer are then machined to final dimension. The finish-machined assembly is pressed into the bore of the heater head housing. The regenerator assembly is shown in Figure 5-32.

The stuffer piece serves two purposes. It fills the space above the regenerator, reducing dead volume in the cycle, and permitting simpler machining of the heater head housing. Secondly, it forms the upper dome or wall of the expansion space and an extension of the partition wall, separating the regenerator flow channel from the expansion space. The flow channel from the housing neck to the top of the regenerator consists of 80 radial slots machined on the top of the stuffer, connecting to 80 vertical holes drilled through the outer ring of the stuffer. These holes communicate to a shallow distributing space just above the top of the regenerator matrix. The stuffer, partition wall, and bottom seal ring are all made of 316 stainless steel.

Clamping Plates - The heater head housings fit into round recesses machined in the engine block, immediately above the bore which houses the annular gas cooler. A shoulder or flange on the lower end of the heater head housing is gripped and restrained under a split clamping plate, which is held to the engine block with seven bolts per half. Two half-plates are employed for the two heads on each bank of the V-block. The clamping plate is made of AISI 4140 steel heat-treated to a hardness of R_C25-30.

Configuration No. 3 Heater Head - This design offers improved performance compared to configuration No. 1, due to reduced dead volume and reduced hot mass and conduction heat loss. Configuration No. 3 uses only one manifold above the expansion space. There is no manifold above the regenerator; individual tubes penetrate directly into the heater head housing above the regenerator. This eliminates the dead volume and hot mass associated with this manifold. The heater head housing is cast from Haynes Stellite 31, which has similar properties to NASA UT4GAT and which has greater strength than XF-818. This allows a reduction in wall thickness throughout the cylindrical, dome and single-manifold parts of the housing. This also reduces hot mass, and presents a thinner conduction path for heat flow down the cylinder wall from the hot region toward the cooler.

This heater head is interchangeable with configuration No. 1. It will fit the same engine block and gas cooler. It also incorporates the same finned tube section and vertical location with respect to the preheater. This heater will also mate with the same combustion gas seals at the combustor and preheater. Because the heater head housing is smaller in diameter, a different clamping plate is required. Also, because there is no inner manifold ring and the diameter of the inner tube-row circle is smaller, a different, smaller flamestone is required.

A somewhat longer piston dome (90.0 versus 73.5 mm) is used, to optimize engine performance.

The design is illustrated in Figure 5-33. Each quadrant has 25 Inconel 625 tubes of 4.5 mm OD and 0.75 mm wall thickness. The tubes extend radially inward from the manifold, bend upward to form the front row, then undergo a hairpin bend downward to form the rear row, and then bend under the manifold to insertion points into the heater head housing above the regenerator. All tubes are identical from the manifold to the end of the rear row. However, each tube in a quadrant will have a different configuration from the end of the straight, rear-row section to its insertion point into the housing. With two different types of quadrants, this means that there will be 50 different configurations of tubes for the entire heater head.

The housing is shown in Figure 5-34. The lower end is virtually identical to that of configuration No. 1, to mate with the engine block. At the upper end, two different types of head are required, one for cylinder No. 1 and 3, a second for cylinders No. 2 and 4. The manifold and neck connection to the dome are oriented differently for the two types, due to the different angle between the cylinder axis and the manifold plane when mounted on the V-block. Individual bosses are cast on the outside of the dome to receive the tubes. Inside, just above the regenerator, the volume occupied by the stuffer piece on configuration No. 1 is cast integral with the housing in this design.

To avoid excessive thermal stress in the housing during engine start-up, thermal sleeves are used at the tube insertion points into the housing. The tubes are brazed to the housing to a depth of only 4.5 mm. The sleeves are inserted from below for a depth of 18.5 mm, and contact the housing only at four narrow, annular rings. The remainder of the sleeve is separated from the housing by an insulating, annular clearance space. This design will protect the housing from very rapid heating at the tube penetrations during start-up, when the bulk of the housing is cool, thereby minimizing thermal stress. Stress analysis of the entire configuration No. 3 heater head housing was also done. The analysis indicated the safety factors given in Table 5-7.

It should be noted that this design has all of the front-row tubes connected to the expansion space, with the rear-row tubes connected to the regenerator. This is opposite to the connection scheme of configuration No. 1. The configuration No. 3 arrangement offers an advantage of lower front-row tube wall temperature at idle conditions. The configuration No. 1 arrangement was analyzed and found to produce a front-row tube temperature of 990°C at idle with a rear-row set temperature of 820°C. Configuration No. 3 arrangement is predicted to give a front-row temperature of only 870°C at the same conduction.

The rear-row fins for this design are similar to those of configuration No. 1, except for width and hole size. At the bottom of the fin stack, two flat fins extend outward a few millimeters to pick up the seal rail for the heater-to-preheater seal. In addition, a single flat fin is attached near the bottom of every other front-row tube. These form a ledge which supports the rim of the flamestone.

Regenerator for Configuration No. 3 - The regenerator for this heater head is also made of sintered, 304 stainless steel wire screens. The sintered annular regenerator biscuit has virtually the same dimensions as that for configuration No. 1 and is brazed to the OD of a cylindrical partition wall in the same manner. The regenerator design is shown in Figure 5-35. However, the partition wall is Inconel 718 alloy in this case. This was selected to provide a better match of thermal expansion with the HS-31 housing and a higher strength to resist radial compressive load imposed by the regenerator mesh. An additional sleeve of Inconel 718 is welded to the top of the partition wall after the regenerator matrix is in place. This top sleeve seats in a machined pilot bore of the housing, and has a shoulder that bears against a mating shoulder of the housing just above the regenerator. These interfaces form the seal between the expansion space and the gas space above the regenerator.

CEDS Design - The CES, consisting of water jacket, cylinder liner, crosshead guides, pistons and rings, piston rods, seals and cold connecting duct plates was considered a separate entity from the EDS in the Mod I engine. The EDS consisted of the crankshafts, connecting rods, gears, bearings, and casings for these parts. In the Mod I, virtually all these components were separate parts. However, in the Mod II engine, several of these components were integrated together and the two systems are more conveniently discussed as an integrated whole.

Cast V-Block - The unified cast iron block in V-4 configuration establishes basic geometry of the engine and incorporates the functions of waterjacket, cold duct plates, crosshead liners, and crankcase. It also incorporates several of the gas-control lines as drilled passages, which were external tubing runs in the Mod I engine. The block contains two banks of two cylinders, the banks separated by a V-angle of 40°. The block configuration is illustrated in Figure 5-36. This figure also emphasizes another important aspect of the block design: the pilot bores for locating the cooler/cylinder liners, main seal housing and heater head housing are all machined in a single part (the block) at a single machining set-up. The longitudinal view of the engine block is shown in Figure 5-37, which is a section view of the engine assembly along the crankshaft axis.

The block casting has a flat machined bottom face to which a cast aluminum oil pan is bolted. Three transverse walls in the casting also terminate at the bottom plane, and support the three main bearing caps. The block and assembled caps are line-bored to generate the locating bores for the three main bearings. The front face of the block is machined to mount a cast-aluminum engine-front-cover. The front cover is bolted and doweled to the engine block, and supports the hydrogen compressor, water pump, and front oil-supply bushing. The oil supply bushing rides on the front extension of the crankshaft and serves to transfer lube oil from a drilled passage in the front cover to the oil passages in the crankshaft. The front cover also encloses the connecting rod and crank throw for the compressor, which are overhung outboard from the front main bearing. The front cover also houses the one-to-one gear set that drives the water pump off the crankshaft.

Cooling water flow passages in the cast block are illustrated in Figure 5-38. Two through-cored passages are divided into four independent passages by small flow blockers mounted on the gas coolers. Headers for the two inlets and two outlets are thin-wall, tubular steel weldments. This keeps casting weight and complexity to a minimum. The gas coolers for the four cycles are functionally in a parallel flow arrangement for the cooling water.

The cold connecting ducts to convey gas between adjacent cylinders are cast integral with the block. This requires that the block casting by capable of containing H_2 gas at maximum cycle pressure in these regions, and have fatigue strength to support cyclic pressure. The cold-ring manifolds are cast in the block as semi-torroidal cavities of tapering depth, to distribute and collect gas uniformly under the annular gas coolers. The cold-ring configuration is shown in Figure 5-39.

The engine block is to be a sand casting in nodular iron, 80-55-06 alloy. Nodular or ductile iron was chosen over gray iron because of its greater ductility, which is vital for safety and fatigue life of that portion containing high cyclic pressures. Also the lamellar graphite structure of gray iron creates long passages allowing hydrogen permeation and leakage. The spheroidal graphite structure of nodular iron does not suffer this fault.

Gas Cooler/Cylinder Liner - The annular gas cooler surrounds the piston, its inner wall forming the cylinder liner. It fits into a pilot bore in the cast iron block, and lies directly beneath the annular regenerator in the heater head housing. The inner wall or cylinder liner must contain the cycle pressure, and must provide a hard wear surface for the piston rings to slide on. End plates or tube sheets are machined integral with the cylinder wall. The outer shell is formed by two circumferential segments brazed between the end plates, leaving entry and exit windows to allow water to flow in and out of the shell.

The cooler body/cylinder liner is made from 329 stainless steel, which has sufficient yield strength (550 MPa) to support the stresses imposed by cycle pressure in a relatively thin wall. Each cooler has 348 tubes of 321 stainless, 1.5 mm OD and 0.25 mm wall thickness, arranged in four concentric rings. Six tubes are omitted at the entrance and exit windows to provide inlet and outlet plenums for the circumferential water flow. The water flow path through the cooler consists of two semi-circuler paths in parallel, each four tube-rows wide and 42 tube-rows long. The cooler configuration is shown in Figure 5-40 as a brazement prior to final machining. The tubes and outer-wall segments are vacuum brazed in place using Nicrobraz 51. This brazement is finish machined, after which the cylinder liner bore surface is ion-nitrided to a hardness of R_c66-70, and then honed to a surface finish of 0.5 μm.

Main Seal Assembly - The main rod seal for the Mod II engine will be a PL seal similar to that used on the Upgraded Mod I engine. The PL seal element itself is reduced in length from 20 to 15 mm, compared to the Mod I seal, as part of the effort to reduce overall engine height. The seal will be made from Rulon J material. The fit of the seal throat on the piston rod ranges from 0.01 to 0.20 mm tight. Also, the dimensioning and tolerances on the conical bottom of the seal and the metal seat with which it mates were changed to ensure either line-on-line contact, or contact concentrated at the bottom tip of the seal.

The seal installation assembly is shown in Figure 5-41. A further change from the Mod I seal installation is the addition of a close-fitting guide cylinder for the spring carrier. This guide is made as an extension of the seal seat. This feature was added to ensure that the spring carrier is held in proper alignment to the seal axis, and spring pressure is uniformly applied around the seal circumference. Both the seat/guide and the spring follower are brass.

The seal seat fits loosely in the main seal housing, allowing the seat and seal to slide laterally to a position of proper alignment with the piston rod. An O-ring on the bottom face of the seat effects a gas/oil seal on the sliding interface.

The seal housing pilots into a cylindrical bore in the engine block, concentric to the crosshead guide bore, just below the concentric bore where the gas cooler fits. The OD of the seal housing is a clearance fit through the cylinder bore of the gas cooler, allowing the seal housing to be assembled into the engine block after the cooler is in place. The bottom of the seal housing has an annular groove on the OD which forms an oil gallery around the bottom of the seal housing. This annular gallery of each of the four engine cycles intersects an oil-supply passage drilled from front to back at the center of the engine block. This arrangement distributes pressurized oil to the two lube oil jets press-fitted into the bottom of the seal housing of each cycle. These jets are extended to within ∽4 mm of the piston rod where it penetrates the seal to ensure proper oil impingement even at very low speed, when oil supply pressure is low. The oil gallery is sealed at the top by an O-ring around the seal housing, and at the bottom by the smooth metal interface between the bottom of the seal housing and the engine block or crankcase.

The spring follower pilots into a bore on the top of the seal housing, and houses the spring, cap seal and timed-injection bushing. A cover piece fits on top of the entire assembly, clamping the timed injection bushing in place and compressing the spring to the proper length. The cover is secured to the housing with three screws. The top surface of the cover is accurately contoured to provide a close fit to the bottom of the piston dome, and forms the bottom surface of the cycle compression space. It should also be noted that a portion of the cold gas duct interconnecting adjacent cycles is machined in the seal housing. This machined passage allows cooled gas from the bottom of the cooler of the adjacent cycle to flow up through the seal housing into the compression space immediately above the seal housing.

The cap seal is virtually identical to that of the Mod I engine, is made of Rulon LD, and has a fit on the piston rod ranging from 0.01 to 0.05 mm tight. The timed injection bushing, made of Vespel SP-1, has four radial gas-feed holes which carry gas at Psupply to the OD of the piston rod just below the cycle compression space. Through most of the piston-rod travel, the clearance between rod and bore of the injection bushing is small, and gas feed is obstructed. However, the piston rod is machined to a smaller diameter in an annulus just below the bottom of the piston. At the bottom of the piston stroke (peak of the compression cycle), this smaller annulus enters the injection bushing, creating a larger clearance and allowing momentary gas flow into the compression space.

Two gas flow passages cross the cylindrical interface between the OD of the seal housing and the bore ID in the engine block. One is the timed-injection gas flow at Psupply; the other is a vent and oil drain passage. The injection gas flows from a check-valve block bolted to the engine block, through a drilled passage in the engine block, to the drilled passages in the seal housing shown in Figure 5-41, and up to the injection bushing. To maintain a leak-tight flow path across the interface between engine block and seal housing, a "port seal" assembly is used. This assembly is shown in Figure 5-42. It consists of a central stainless steel tube with Buna N elastomer sleeves at each end of a spacer, a Delrin washer and an internally-threaded nut or sleeve. The assembly is inserted into the drilled passage in the engine block and extends into the bottom-drilled port in the seal housing (bottom right side of Figure 5-41). When in place, the threaded nut is tightened, which compresses the Buna-N sleeves, forcing them radially outward to seal against the wall of the passage. One sleeve seals in the seal housing, the other in the engine block. The installation is shown in Figure 5-43. A similar port seal is used for the seal vent/oil drain passage, which is also shown in Figure 5-43.

Piston/Crosshead/Connecting Rod - The Mod II assembly of piston, piston rod, crosshead, wristpin and connecting rod is shown in Figure 5-44. The piston consists of an Inconel 618 dome electron-beam welded to an AISI 4130 steel base. A radiation shield is interposed above the base, and its rim is included in the electron-beam weld. Two guide rings of Rulon LD are cemented into annular grooves in the base, above and below the two piston rings. The piston rings are the same split-solid configuration used on the Mod I engine. The space between the two ring grooves is vented into the dome cavity, which, in turn, is vented to the Pmin space below the cap seal by a drilling in the piston rod.

The piston rod is made of Nitralloy 135-M. The material is first through-hardened to R_c38-40 to obtain the required strength. After rough machining, the portion of rod that slides in the main seal is nitride hardnened to R_c68-70, and is then ground and polished to a surface finish of 0.06 µm. The lower end of the rod is necked down to pass through a hole in the wristpin, and terminates in a thread that screws into the crosshead. This lower portion of the rod is induction hardened to R_c50-52 to obtain the higher strength required by the smaller diameters in the attachment to the crosshead. Vertical position and axial alignment of the rod with respect to the crosshead is established by an accurately machined shoulder on the rod, which seats on the square top surface of the crosshead.

The piston rod is attached to the piston base by a heavy shrink fit. The detailed configuration around this interface was established by finite-element stress analysis and fatigue testing of samples of several candidate designs. The detailed configuration is shown in Figure 5-45.

The connecting rod is a split-fork design with needle-roller bearings in the lower end, and plain journal bearings at the upper, wristpin end. The connecting rods are made of AISI 8620 steel, carburized to obtain a case hardness of Rc60 and depth of 1.5 mm on the big-end bore after rough machining. This is then fractured to break off the lower or cap end. The caps are bolted in place and the big-end bore is then finish ground to a surface finish Of 0.1 µm. This bore surface serves as the outer race for the needle-roller bearing. The connecting rod/wristpin bearing has the same hardness and surface finish, with an oil-film bearing clearance ranging from 0.010 to 0.032 mm. The connecting rod forks were configured to provide the maximum possible wristpin bearing area.

The connecting rod for the hydrogen compressor is similar to that for the power pistons, except that it does not need to be fractured with a separate bolt-on cap. A one-piece design can slip over the front end of the crankshaft.

The crosshead provides an oil-lubricated, cylindrical bearing surface to support lateral forces exerted by the connecting rod, and allow only vertical, reciprocating motion of the piston rod. The Mod II crosshead is made of AISI 4340 steel hardened to R_c40-45. It has a center web that fits between the forks of the connecting rod and through which the wristpin fits. The cylindrical, sliding surface is machined to a 0.4 µm surface finish. The top of the crosshead is configured to guide oil flow running down the piston rod to the crosshead sliding surfaces. Grooves on the face of the connecting rod forks next to the crosshead center web guide oil into the wristpin-bearing oil film.

The wristpin is also made of AISI 8620 and is carburized and ground to the same hardness and surface finish as the connecting rod bores. The wristpin is a tight fit through the crosshead, unlike the Mod I, which was a full-floating pin. A hole is cross-drilled through the pin to accommodate the shank of the piston rod where it threads into the crosshead.

Crankshaft - The Mod II employs a single crankshaft which carries all four pistons and the hydrogen compressor on three main bearings. The power pistons are paired, two between each main bearing, and the compressor is overhung. The short stroke (30 mm) of the engine allows substantial overlap between adjacent crank throws and main journals, which provides a stiff crankshaft. All journals were increased in diameter to 38.4 mm from a previous value of 35 mm, to provide additional resistance to bending deflection. Thorough stress and deflection analyses of the crankshaft were performed using finite-element methods. The crankshaft is illustrated in Figure 5-46.

The crankshaft journals act as inner races for the needle-roller bearings at all crankpin and two main bearing locations. Particular care was given to selection of crankshaft material to ensure high quality and freedom from defects and inclusions that might jeopardize roller-bearing life. The shaft is made from AISI 9310 and all journal surfaces are carburized to a hardness of Rc58-60 and case depth of 0.8 mm. The journals are ground to a surface finish of 0.2 µm.

The imbalance forces from the pistons, connecting rods and compressor are totalled, and two-plane balancing is applied using rotating eccentric masses at the extreme ends of the shaft. One eccentric is incorporated in the splined drive hub to which the flywheel is bolted and the other is incorporated on the shaft-mounted gear that drives the water pump, at the opposite end of the shaft. Counter-rotating imbalance forces are handled by small balance weights on the gear-driven water-pump shaft at one end of the engine, and on the gear-driven oil-pump shaft at the opposite end of the engine.

Lubrication System - The Mod II engine lubrication system consists of a gerotor oil pump, gear-driven by the crankshaft, a pressure-regulating relief valve, an automotive oil filter, a main oil gallery passage running the length of the engine block, and separate oil feed passages and jets to the several elements requiring lube oil. The oil pump is mounted in an extended rear-main bearing cap, shown in Figure 5-47. The relief valve is installed to tap directly into the pump discharge, and limits the oil supply pressure to 350 KPa (51 psig). The entire oil-supply flow system is shown in Figure 5-48. Spent oil from the bearings, main seals, crossheads, etc. drains down into the oil pan or sump, which is bolted to the bottom of the engine block. The relationship between oil pan and oil pump is shown in Figure 5-37.

At the front end of the engine, oil is supplied to the crankshaft oil-supply bushing. This is a nonrotating bushing that floats with some radial freedom in the engine front cover. The front end of the crankshaft extends through the bushing and has a cross-drilled passage to interconnect the axial oil drillings and bearing supply orifices of the crankshaft with the oil supplied through the bushing. The three main bearings are supplied oil through drillings in the engine block which connect with the main oil gallery passage in the block. The piston-rod seals, crossheads, and wristpins are supplied with oil by jets built into the main seal housings. The annular oil grooves supplying these jets are machined on the bottom of the seal housing of each cycle, and intersect the main oil gallery of the engine block.

Roller Bearing - The Mod II crankshaft rotates in three main bearings. The front and center mains (M2 and M3) are split-cage needle-roller bearings wherein the crankshaft journal is the inner race. Fracture-split outer races are retained in the crankcase and bearing-cap bore by a slightly tight fit and anti-rotation dowel. The rear main (M4) is a stock SKF combination roller/ball bearing assembly that can be fitted onto the crankshaft without being split. The ball bearing portion of the rear main provides positive axial positioning of the shaft in the crankcase, and supports the axial load imposed through the throw-out bearing when the clutch is disengaged.

The crankpin bearings are split-cage needle-roller bearings wherein the crankshaft journal serves as inner race, and the connecting rod bore serves as outer race. The rollers and cage of crankpin and two main bearings are identical, with the geometry given below:

Journal diameter, mm	38.4
Bearing OD, mm	53.0
Roller diameter, mm	7.3
Number of rollers	14
Roller length	24.8
Dynamic capacity, N	45,060

Using life-predicting analysis procedures recommended by SKF Industries, it was demonstrated that the bearings would provide the following life under maximum power operating conditions:

	Hours
Main M2	1533
Main M3	3067
Main M4	524
Crankpin	551
Seven-bearing system	128

The 128 hr life for the seven-bearing system (three mains and four crankpins) exceeds the goal of 100 hrs at maximum power. Further analysis demonstrated a bearing-system life of 4169 hrs under EPA combined driving-cycle load conditions, which exceeds the goal of 3500 hrs.

Cooling Water Pump - The Mod II engine uses a gerotor, positive-displacement water pump because the annular coolers improse higher pressure drop than the Mod I cannister cooler, and because the gerotor offers higher efficiency than centrifugal pumps of comparable head/flow performance. The pump is constructed from standard-size gerotor elements in a specially designed, cast-aluminum housing. The inner gerotor is 400-series stainless steel, and the outer gerotor is Ultem 2700 plastic, to form a reasonably good wear couple against the stainless and aluminum parts under water-lubricated rubbing conditions. The aluminum housing is hard-anodized to provide a nongalling, good wearing surface where sliding contact occurs.

Carbon face seals are installed at each end of the gerotor shaft to isolate the pumped fluid from the rolling-element bearings and oil-lubricated drive gear and over-running clutch. A small pancake DC, electric motor is mounted on the outboard end of the shaft. This motor drives the pump at 400 rpm after engine shutdown to provide 0.4 kg/s water flow to prevent heat soak-back damage to O-rings. During normal engine operation, the pump is driven off the crankshaft by a one-to-one gear set, with a needle-roller, over-running clutch between the driven gear and the pump shaft. This clutch de-couples the pump from the engine crankshaft during after-cooling operation.

Vehicle Integration

Initial engine design during this period also encompassed the installation of the Mod II engine into the Celebrity A-body vehicle. Figure 5-49 shows the position of the engine in the Celebrity engine compartment. This illustrates a slight interference between the vehicle hood line and the insulation cover of the engine EHS. This will be accommodated by a slight flattening of the insulation cover. An analysis of the resulting configuration indicated an increase in heat loss of less than 15 W due to the local reduction in insulation thickness. To achieve this installation position, the transaxle will be rotated forward 12 degrees around the axis of the output drive shafts. This lowers the engine shaft centerline, without affecting vehicle handling characteristics. However, it does cause interference between the starter and the front cross member of the engine/transaxle subframe. This cross member will be relocated forward ~50 mm.

The vehicle installation design was addressed by use of a full-scale mock-up of the engine and auxiliaries, installed in an actual Celebrity vehicle front compartment. Figure 5-50 is a photo view of this installation, taken over the top of the radiator. Figure 5-51 is another view of the engine installation, taken from in front of the car at bumper level with the radiator removed. Installation of several of the SES and auxiliary components is shown in Figure 5-52.

The engine cooling system is illustrated in Figure 5-53. This shows the two radiator cooling fans, one behind and one in front of the radiator. Also shown are the engine cooling-water headers, water pump, and hoses. The cooling circuit for the hydrogen compressor and gas after-cooler is a separate, parallel branch circuit.

SES Components

Three main groups of SES components were also addressed during this period, these are:

1. Alternator and motor/blower for combustion air supply
2. PCV and control blocks for main pressure control
3. Hydrogen compressor and short-circuiting system.

Combustion Air Supply System - Power to supply combustion air will be transmitted to the air blower electrically for the Mod II engine. This will allow flexibility and control of blower speed, provide good transmission efficiency, and obviate the need for a variator and hydraulic power supply system. The alternator will be driven off the engine crankshaft by a two-stage belt-drive system with a four-to-one speed-increase ratio. The alternator design is illustrated in Figure 5-54. The permanent-magnet rotor is supported on ball bearings. The stator assembly has two windings, both three-phase types. A low-voltage winding supplies current for recharging the battery, and a second, high-voltage winding provides power to drive the air blower.

The air blower is a high-speed, centrifugal blower designed to operate at 43,000 rpm at engine maximum power conditions. The blower volute is a dual-outlet, spiral scrollcase with dual exit diffuser sections. This configuration is shown in Figure 5-55. The volute is made from two aluminum sand castings. One casting includes the blower inlet connection and the impeller shroud. The other casting includes the motor mounting flange and one bearing housing. Each casting includes half of the volute aerodynamic passage, which can thereby be formed without cores. Each casting can be made as a single cope-and-drag casting. In high production, the volute could be made from two aluminum die castings. The blower impeller has nine backward-leaning vanes, with a wrap angle of $\sim 190°$. The impeller will be machined from 7075 wrought aluminum alloy. Shroud clearance over the impeller will be between 0.15 and 0.21 mm.

The blower motor is a variable-voltage, permanent-magnet DC unit with two sets of windings. A 12 V winding serves to drive the blower up to $\sim$20,000 rpm during engine up-start. A second winding serves continuous operation, with voltage from 48 to 240 VDC. The assembly of motor and blower is shown in Figure 5-56.

PCV

The hydraulic servo-actuator of the Mod I PCV has been replaced with an electric motor, speed-reducing gearbox and rack-and-pinion drive for the Mod II engine. The design of the Mod II PCV is shown in Figure 5-57. Layout design of the valve and actuator were completed during this report period. Final issue of detail component drawings will be completed in the next report period.

Hydrogen Compressor - Layout design and preliminary component design were completed for a single-stage, three-volume hydrogen compressor for the Mod II engine. A system of external short-circuiting solenoid valves was also designed, which enables the selection of any single or combination of compressor volumes, to meet scheduled compressor flow requirements. The compressor configuration is shown in Figure 5-58.

Mod II Hardware Procurement

A schedule of hardware procurement is shown in Figures 5-59 and 5-60. Late delivery of drawings has put the schedule in jeopardy, however, it is still felt that a January 1986 run of the engine in a BSE configuration is still possible.

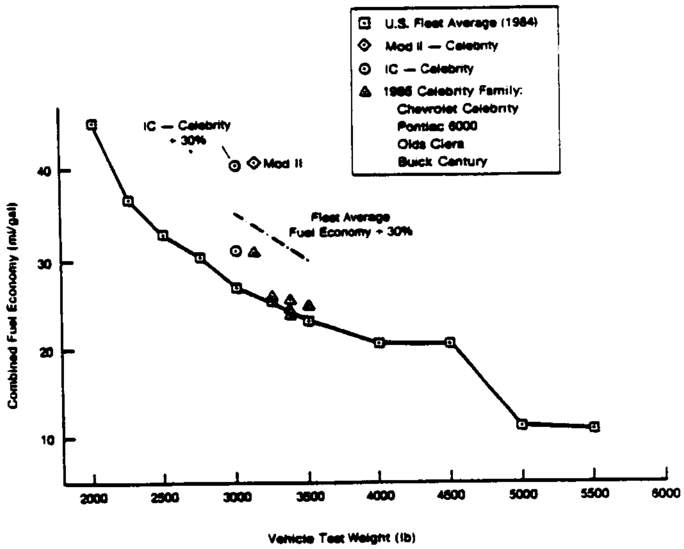

Figure 5-1 Comparison of Spark Ignition and Mod II Fuel Economies

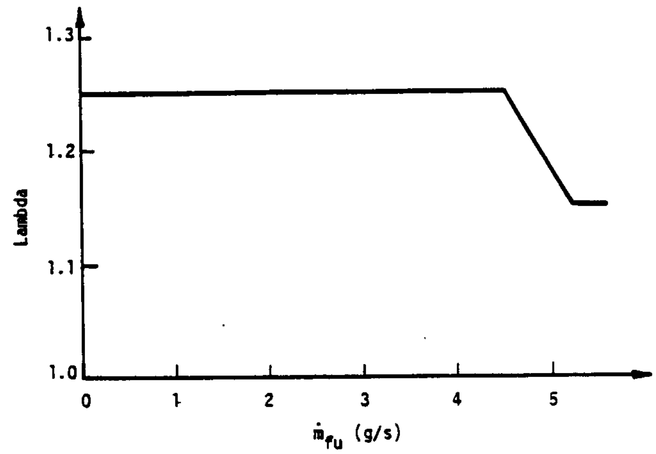

Figure 5-3 ASE Mod II Excess Ratio versus Fuel Flow

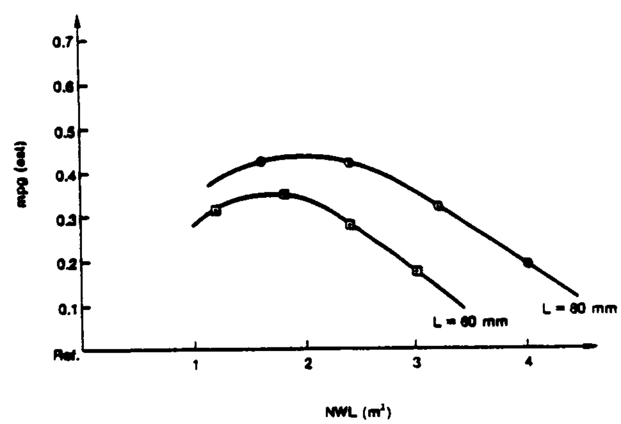

Figure 5-2 Mpg versus NWL for Different Preheaters

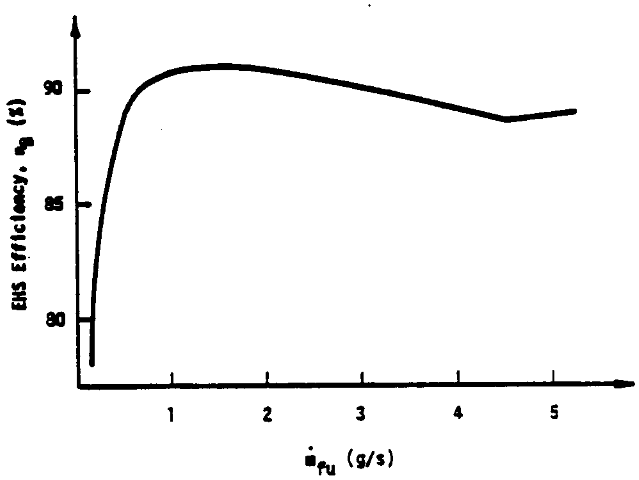

Figure 5-4 ASE Mod I EHS Efficiency versus Fuel Flow

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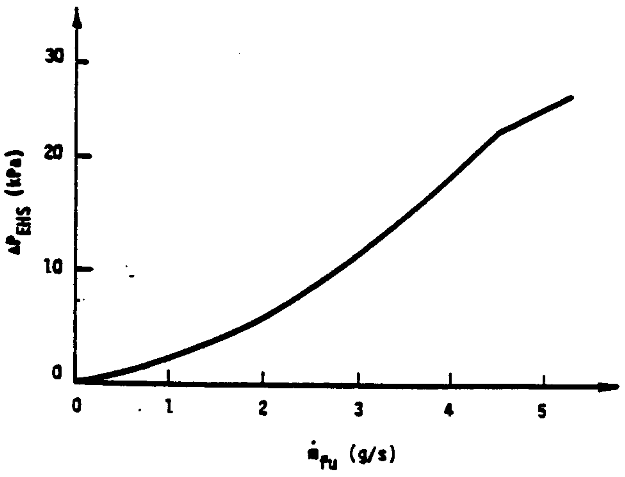

Figure 5-5 ASE Mod II EHS Pressure Drop versus Fuel Flow

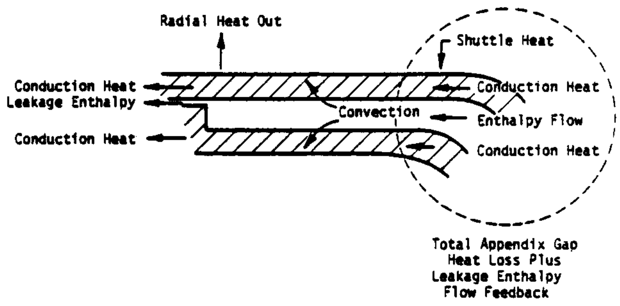

Figure 5-8 Heat Flow Involved

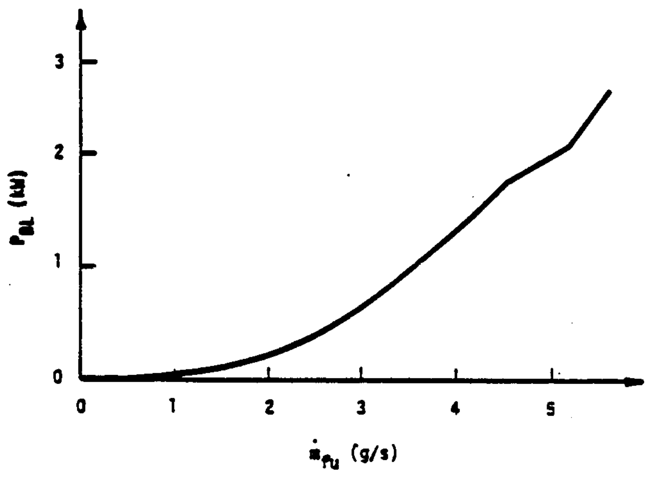

Figure 5-6 ASE Mod II Blower Impeller Power Demand versus Fuel Flow

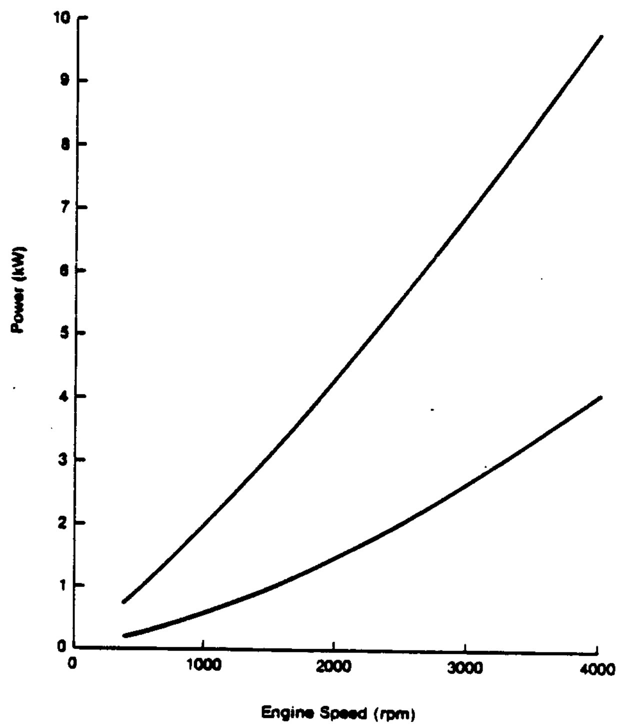

Figure 5-9 Mod II Total Engine Friction Power Load at 2 and 15 MPa versus Engine Speed

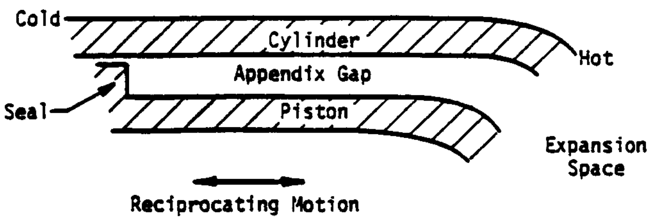

Figure 5-7 Definition of the Appendix Gap

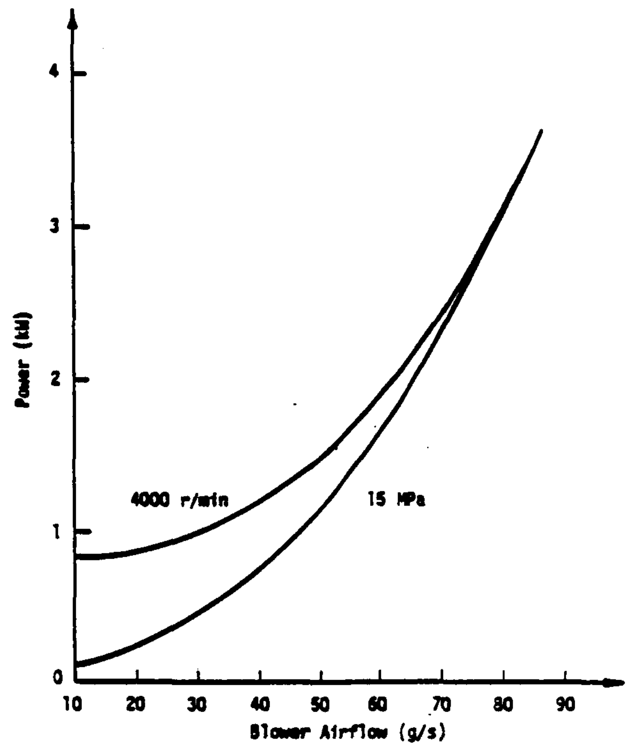

Figure 5-10 Mod II Burner Blower Demand from the Engine Shaft versus Combustion Airflow through the Blower

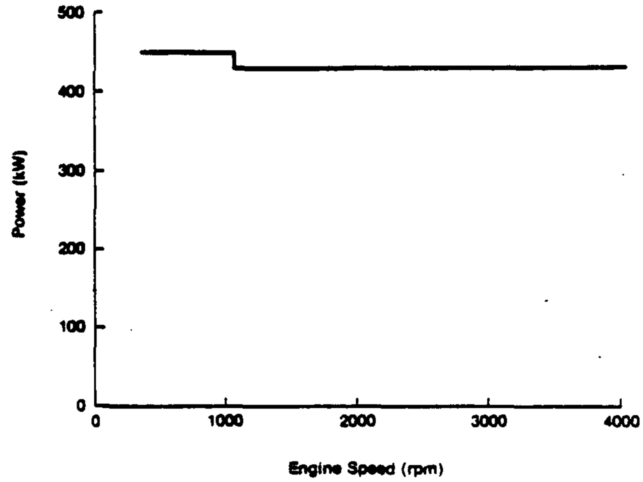

Figure 5-12 Mod II Alternator Poewr Demand, Excluding Blower, from the Engine Shaft versus Engine Speed

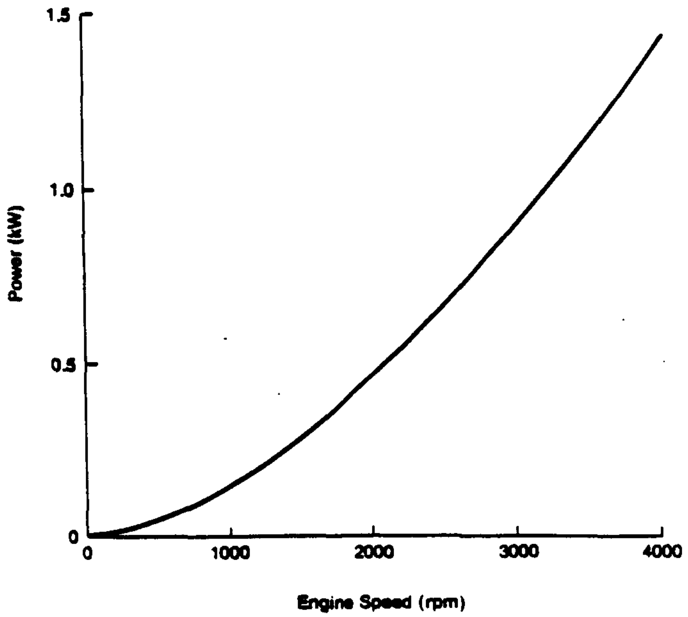

Figure 5-11 Mod II Working Gas Compressor Power Demand from the Engine Shaft versus Engine Speed

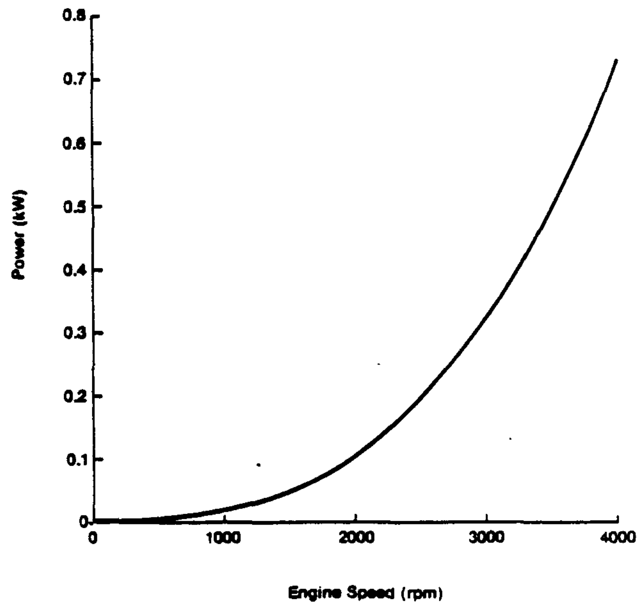

Figure 5-13 Mod II Coolant Pump Power Demand from the Engine Shaft versus Engine Speed

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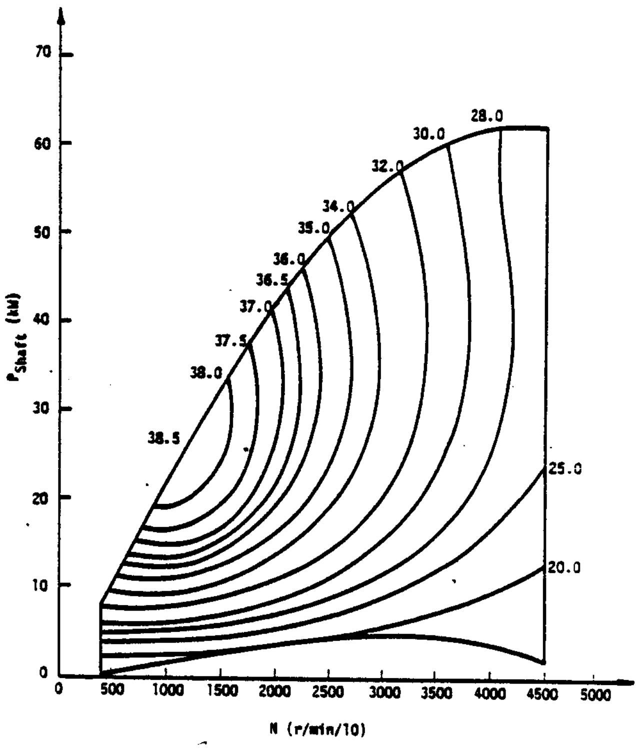


Figure 5-14 Mod II Final Build SES Performance Map

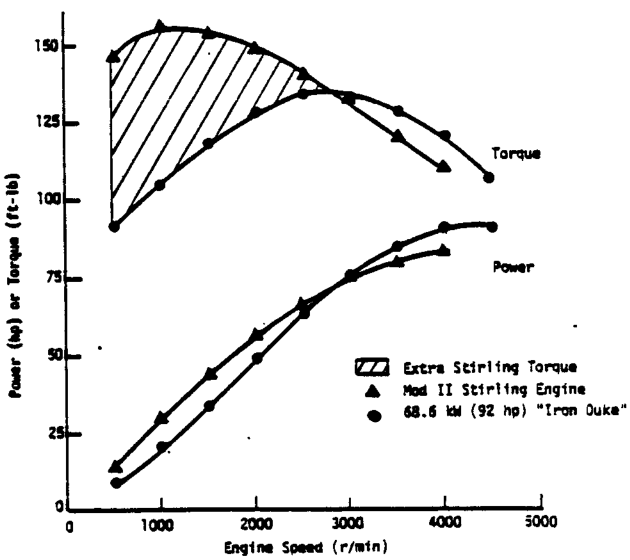


Figure 5-15 Comparison of IC and Stirling Engine Torque

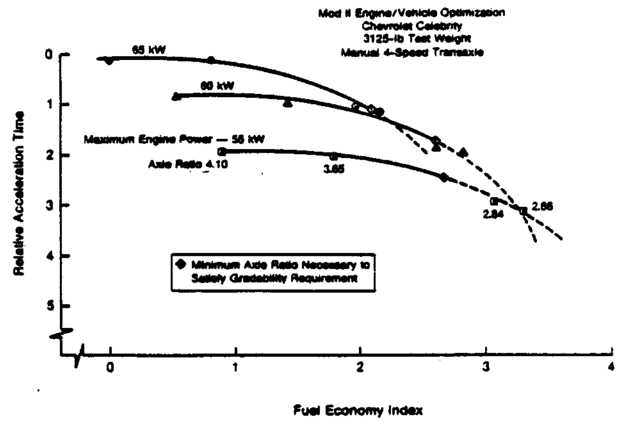


Figure 5-16 Comparison of Stirling and IC Performance

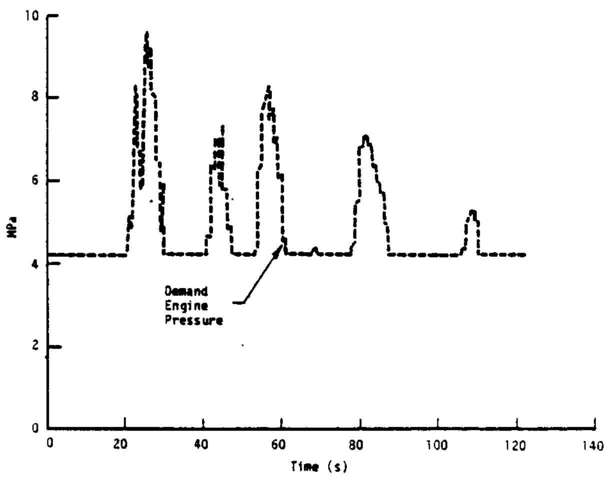


Figure 5-17 MPC System (720°C)

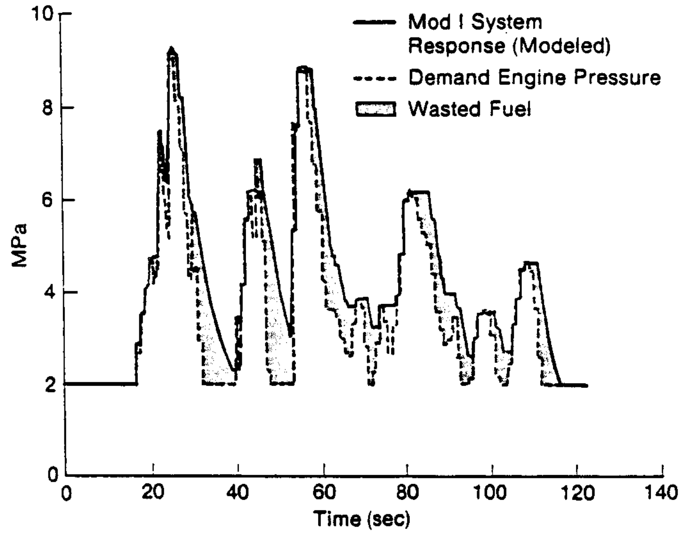


Figure 5-18 Mod I Compressor System (820°C)

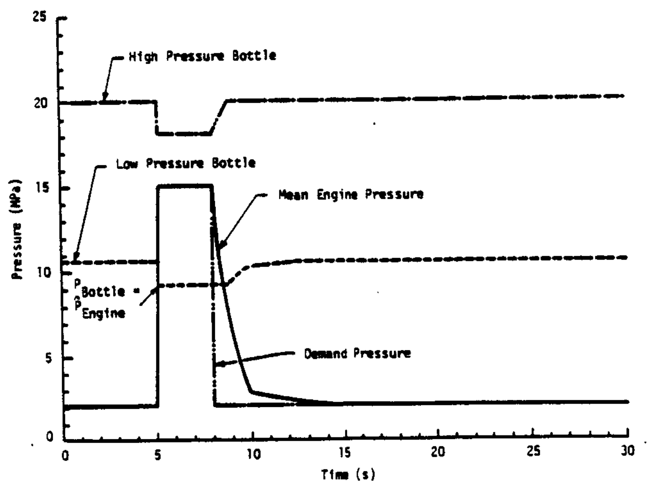
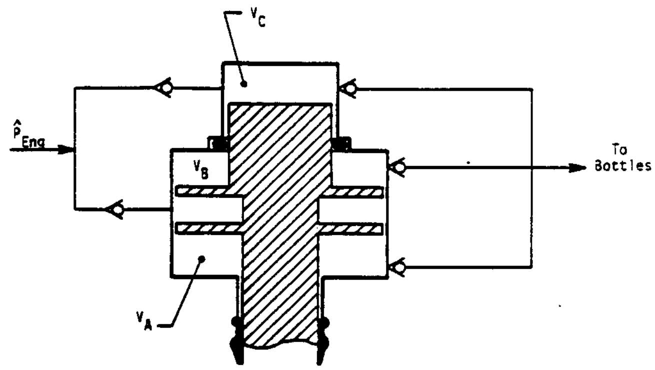


Figure 5-19 Two-Storage Bottle Operation



Mode	I	II	III	IV	V	VI	VII
Active Volumes	V_C	V_B	V_A	$V_B + V_C$	$V_A + V_C$	$V_A + V_B$	$V_A + V_B + V_C$

Figure 5-20 Mean Pressure Control System Three Volume/Seven Mode Compressor

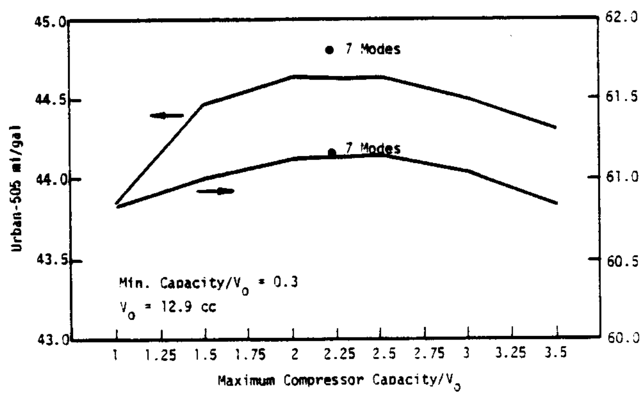


Figure 5-21 Mod II Fuel Economy Sensitivity to Maximum Capacity (Three-Volume/Three-Mode Working Gas Compressor)

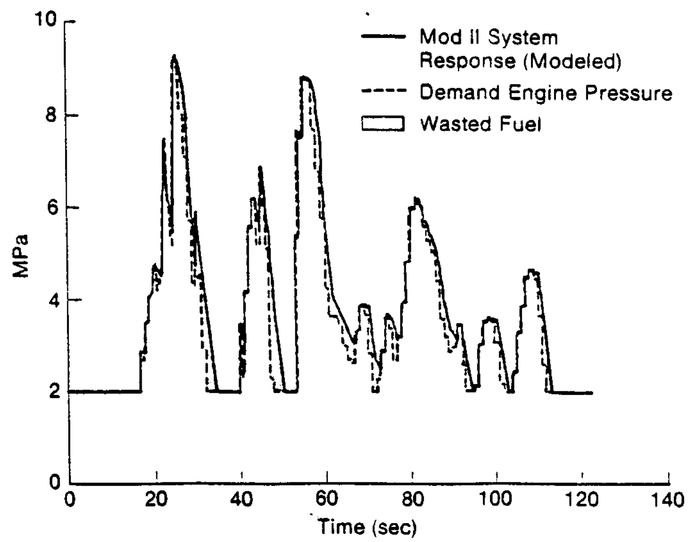


Figure 5-22 Mod II Compressor System (820°C)

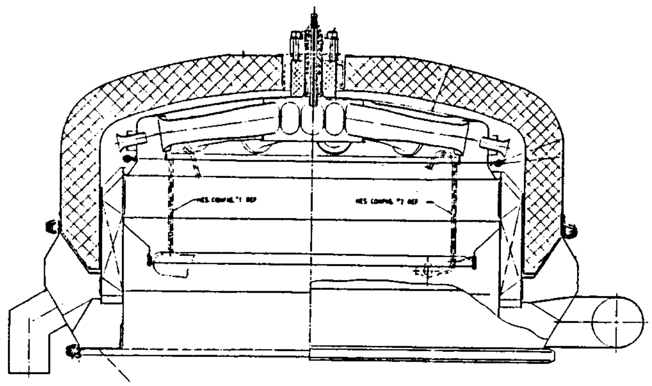


Figure 5-23 EHS Assembly

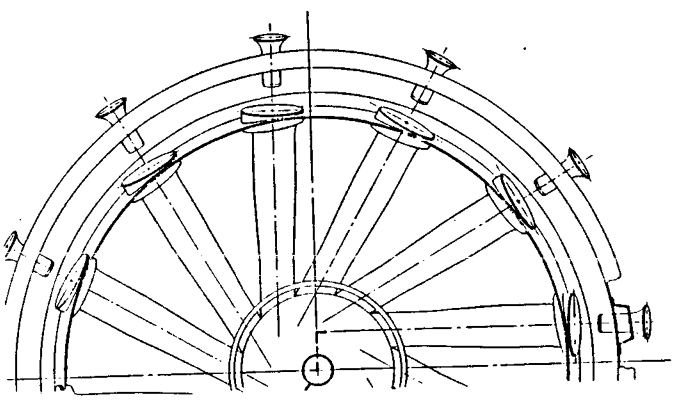


Figure 5-24 Orientation of Ejectors and Mixing Tubes in 12-Tube CGR Combustor

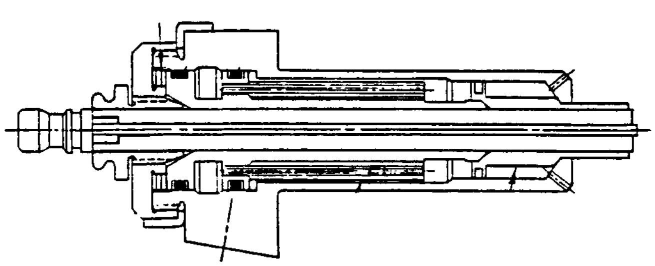


Figure 5-25 Mod II Fuel Nozzle Assembly

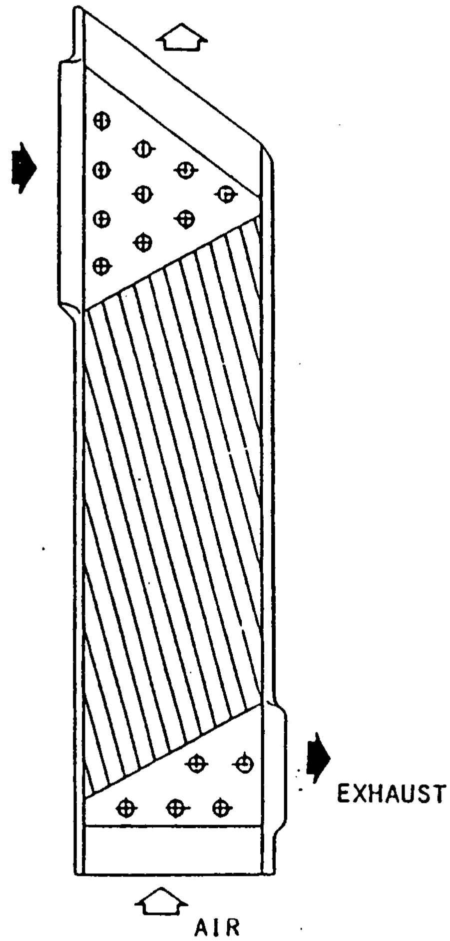


Figure 5-26 Preheater Plate Configuration

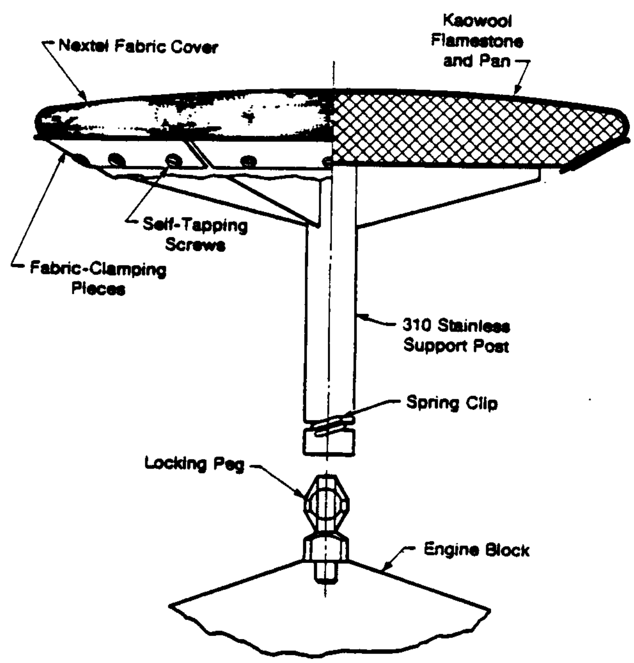


Figure 5-27 Flamestone Assembly

c-2

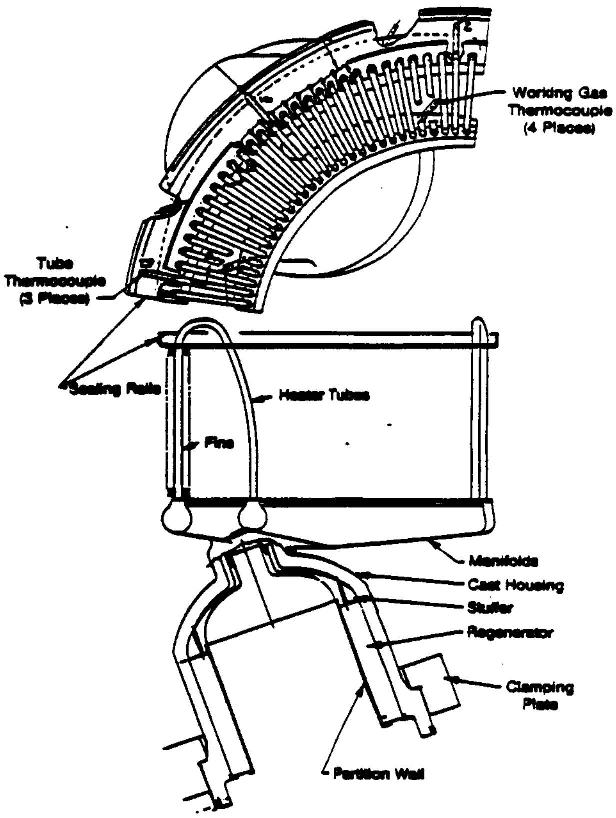


Figure 5-28 Configuration NO. 1 HES Design

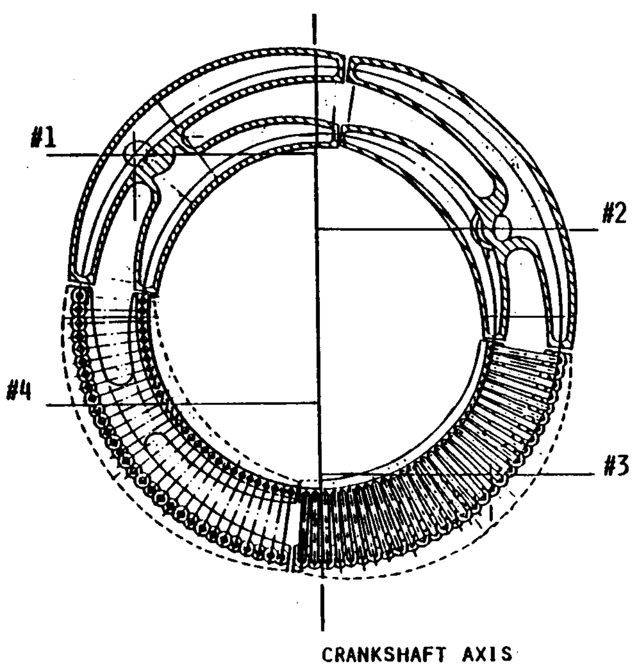


Figure 5-29 Heater Head Manifold Arrangement

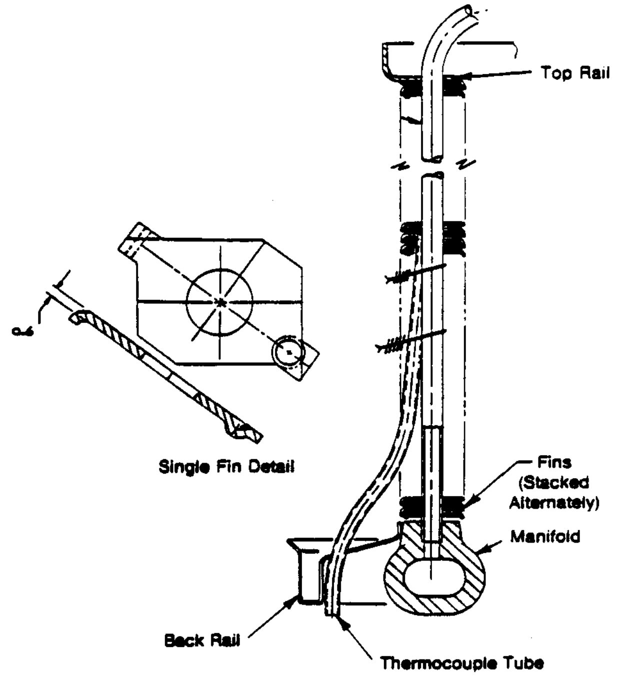


Figure 5-30 Tube-Fin Arrangement, Heater Head No. 1

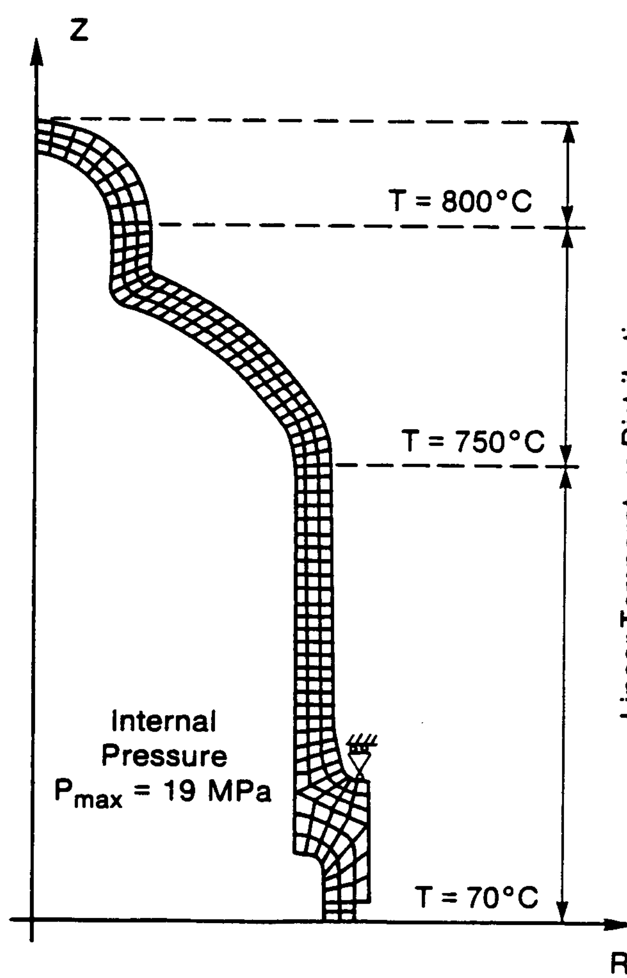


Figure 5-31 Finite-Element Model of Heater Head Housing for Stress Analysis

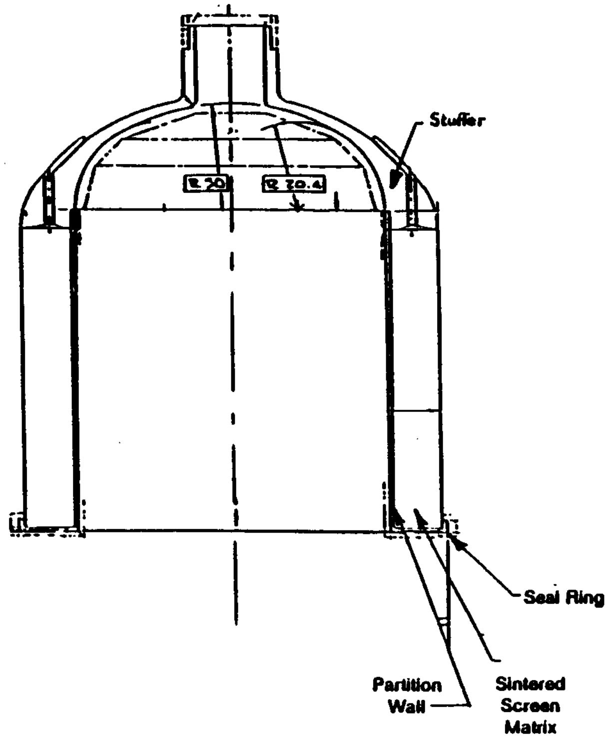


Figure 5-32 Regenerator Assembly,
Configuration No. 1

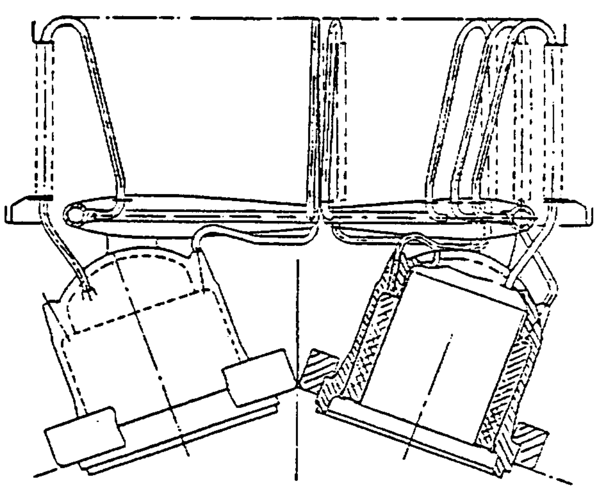


Figure 5-33 Configuration No. 3 Heater Head Design

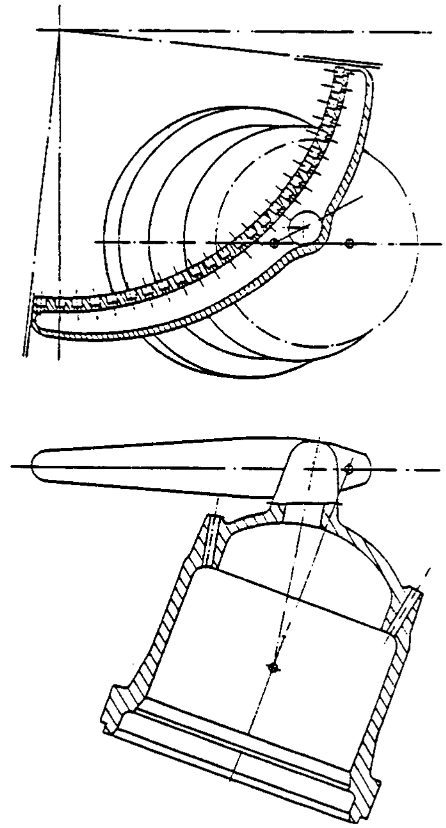


Figure 5-34 Configuration No. 3 Heater Head Housing

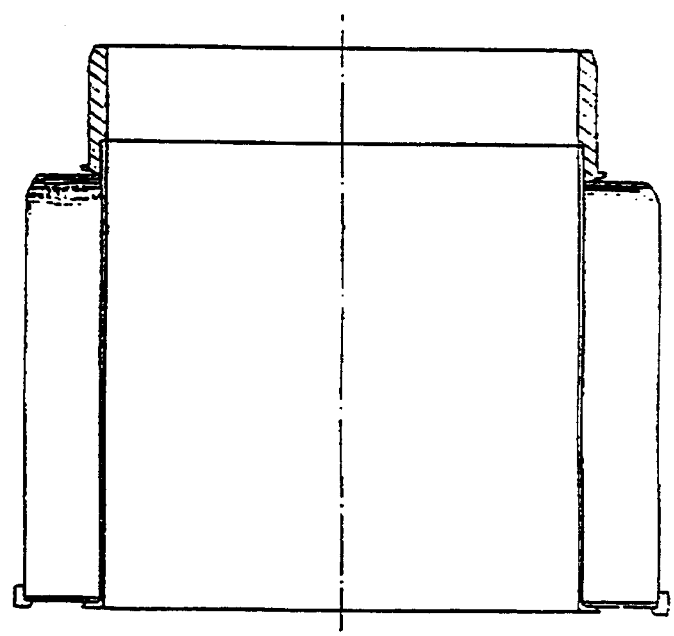


Figure 5-35 Regenerator for Configuration No. 3

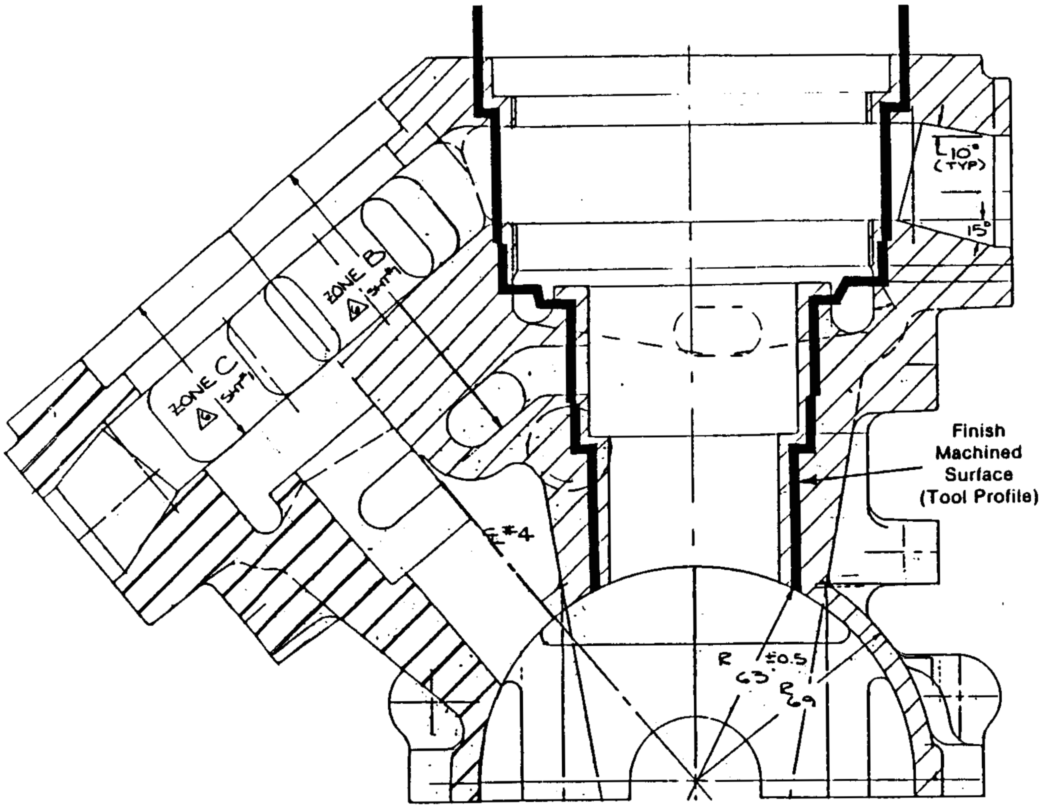


Figure 5-36 V-Block Configuration

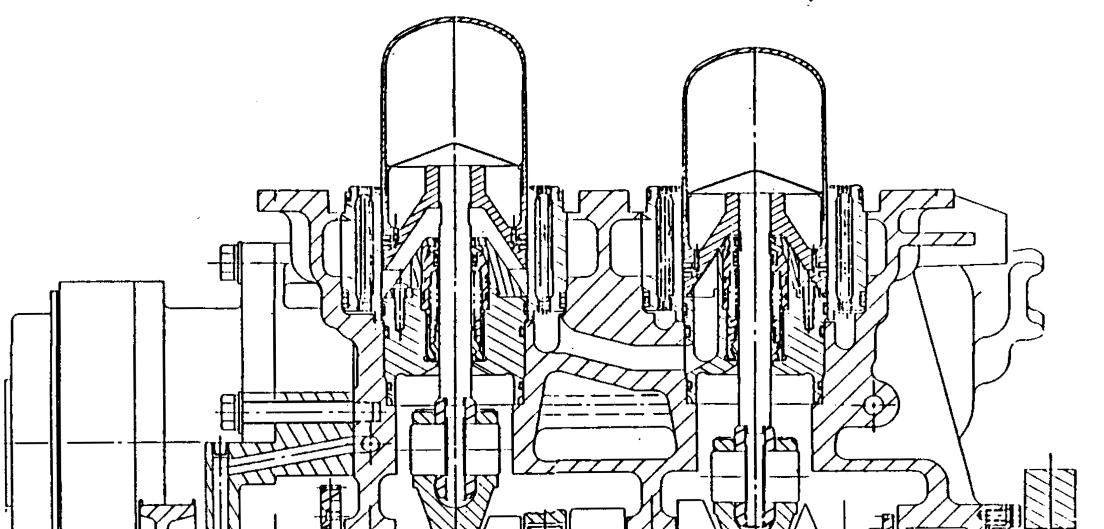


Figure 5-37 Longitudinal View of Mod II Engine

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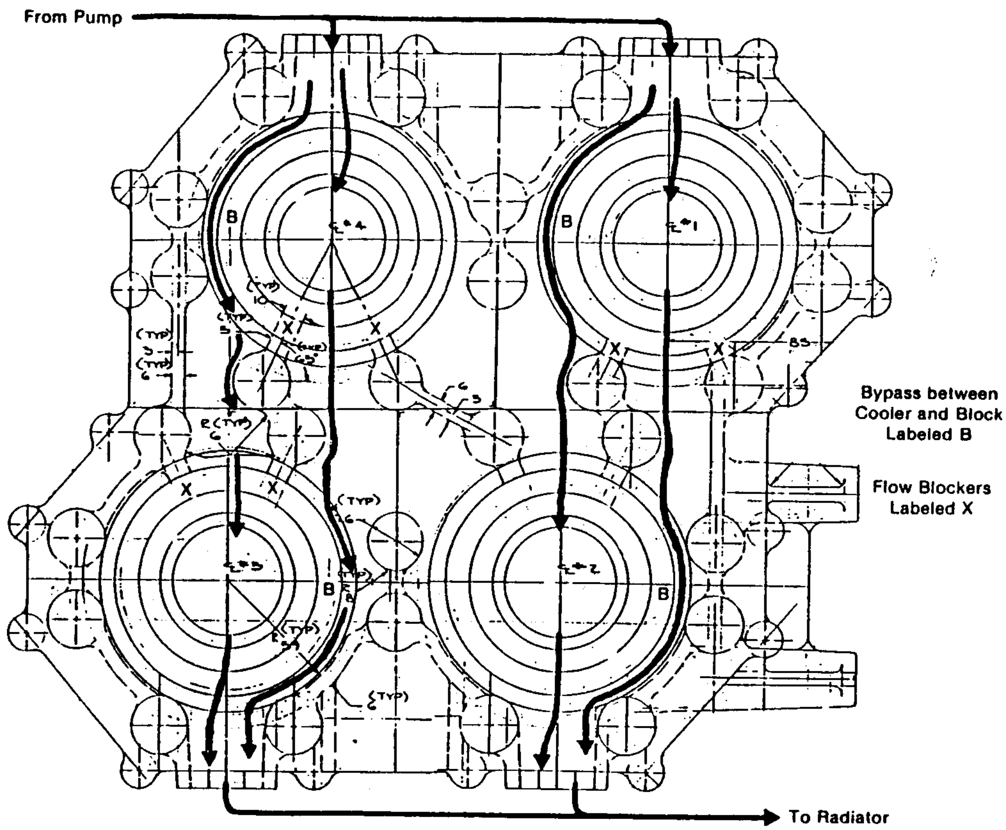

Figure 5-38 Cooling Water Flow in Mod II Cast Block

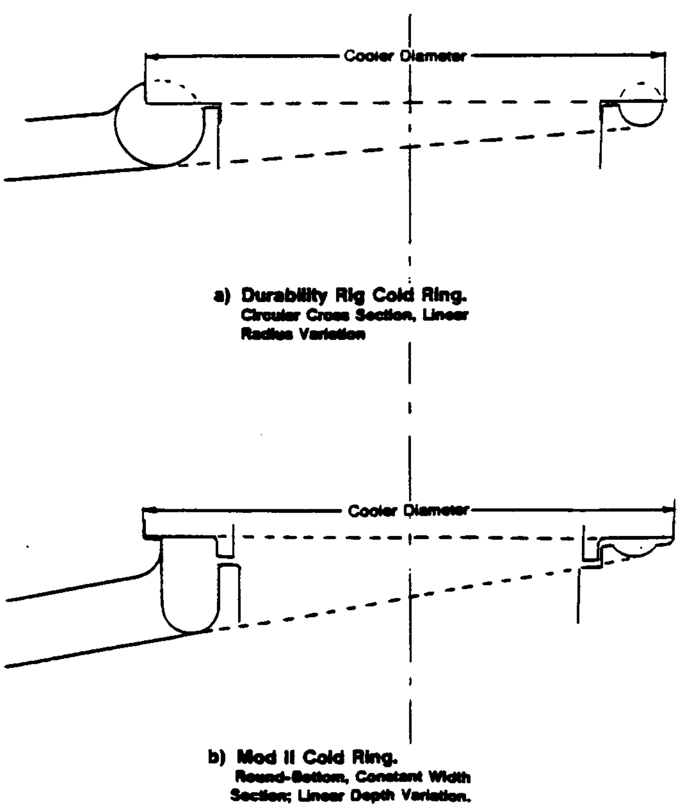

Figure 5-39 Mod II Cold Gas Ring Design

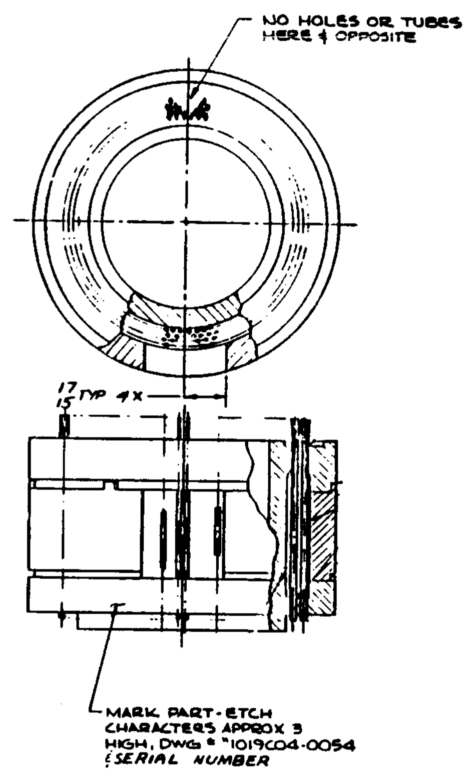

Figure 5-40 Mod II Gas Cooler Configuration

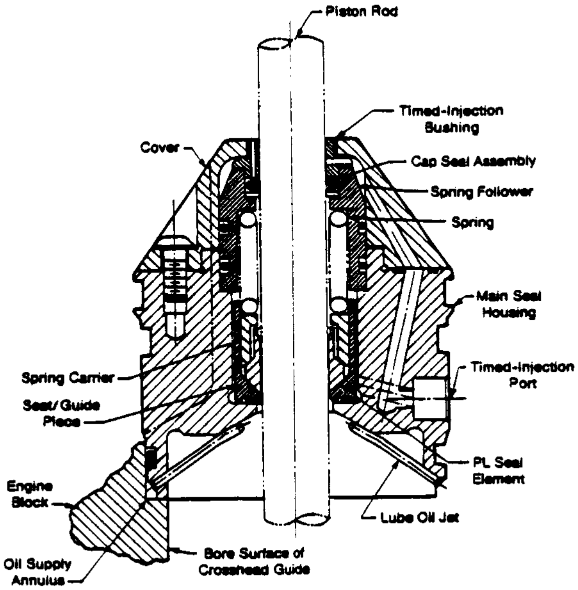

Figure 5-41 Main Seal Assembly

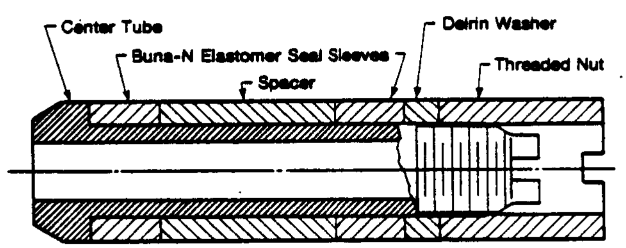

Figure 5-42 Port Seal Assembly

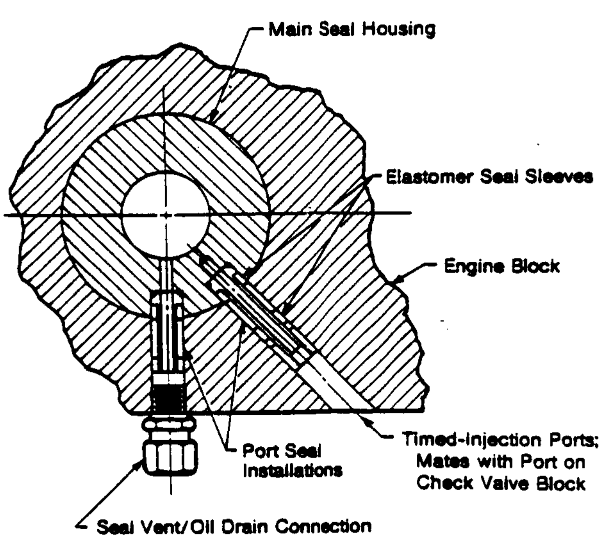

Figure 5-43 Installation of Port Seal Assemblies

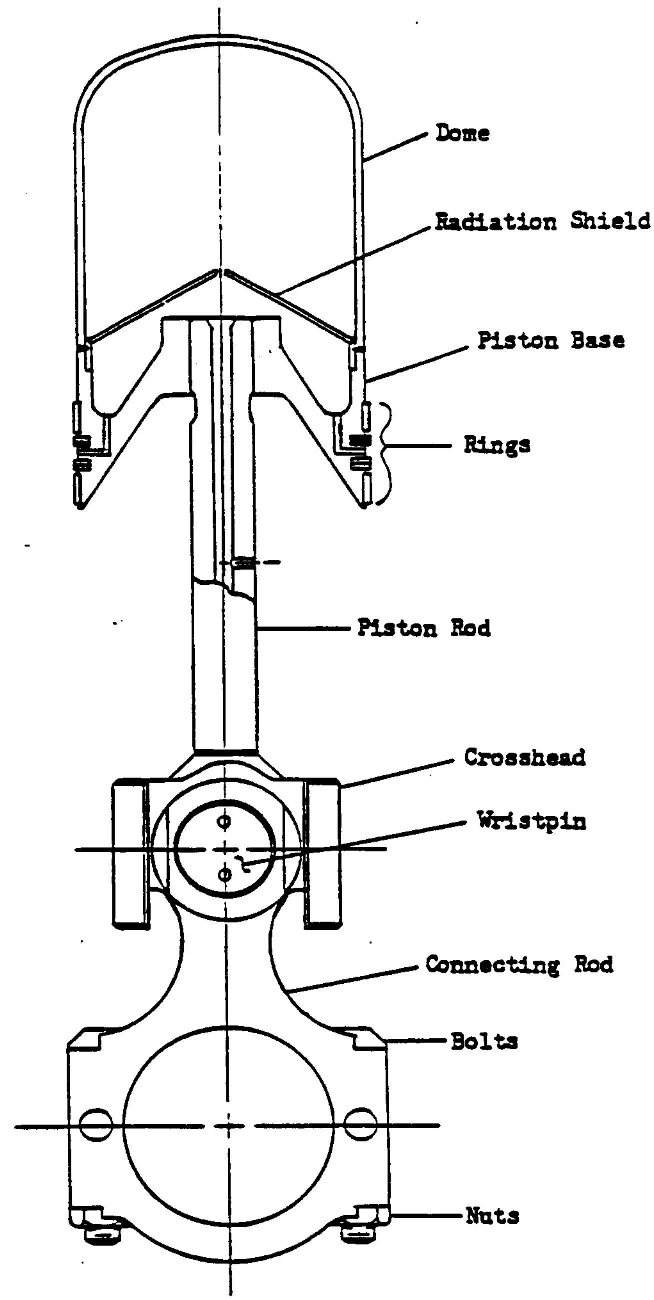

Figure 5-44 Mod II Piston/Crosshead/
Connecting Rod Assembly

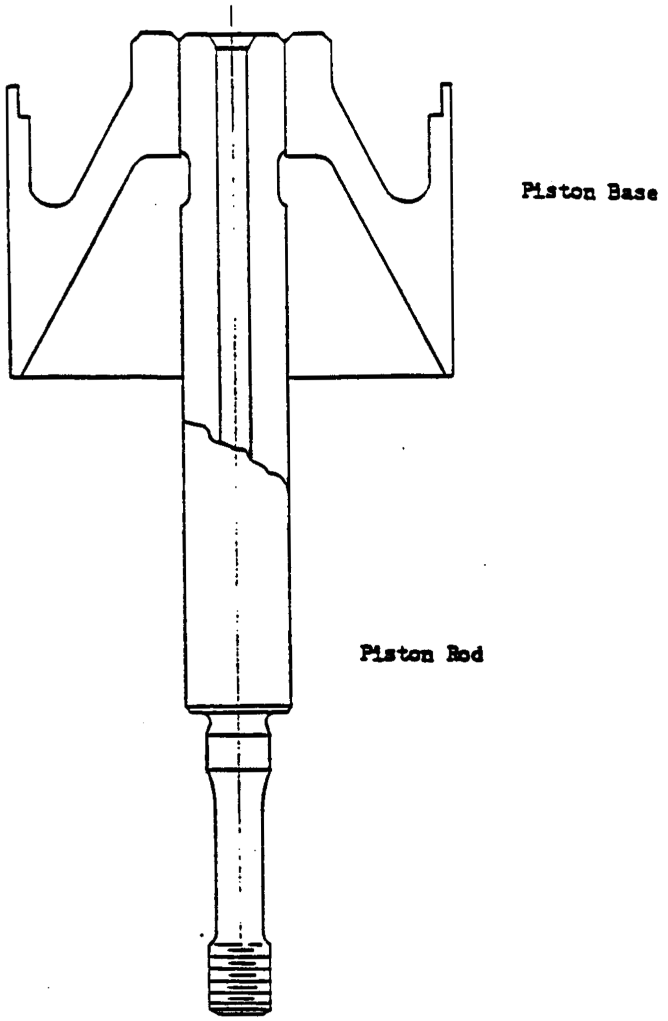

Figure 5-45 Details of Piston Rod/Base Design

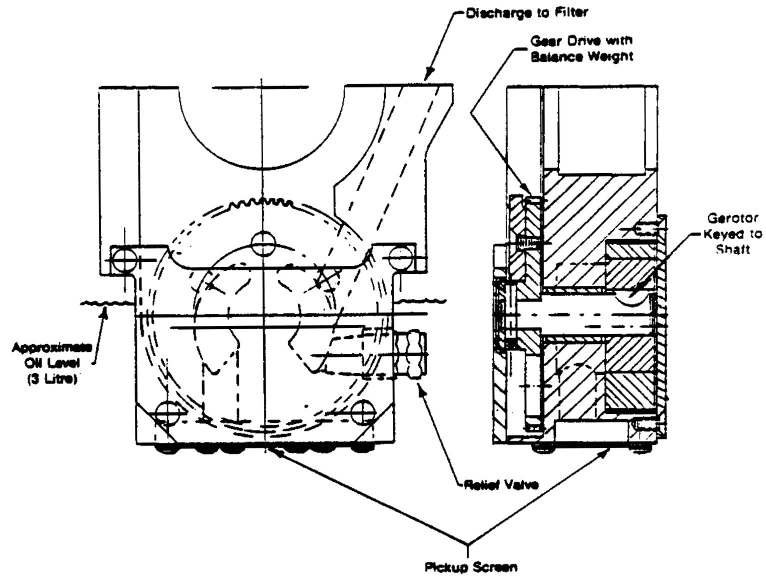

Figure 5-47 Mod II Oil Pump Assembly

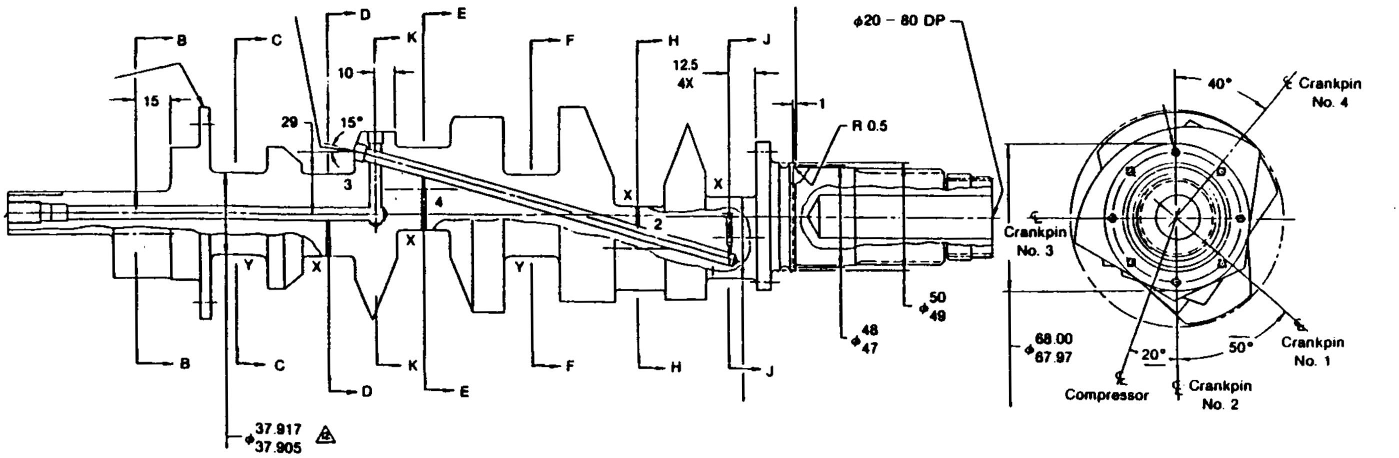

Figure 5-46 Mod II Crankshaft

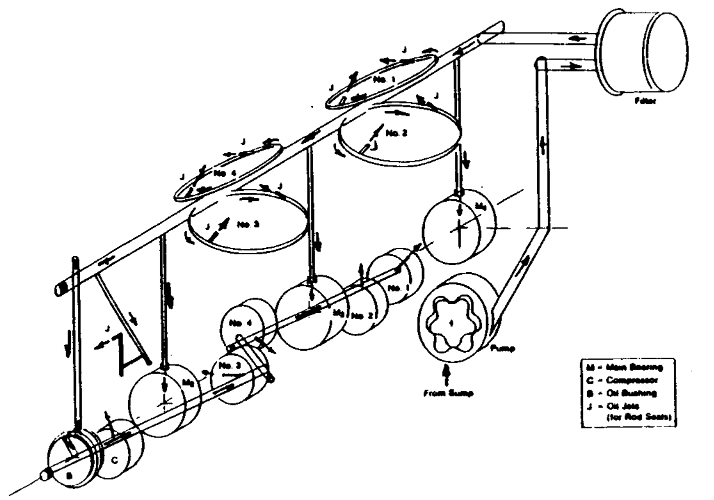

Figure 5-48 Mod II Oil Supply Flow System

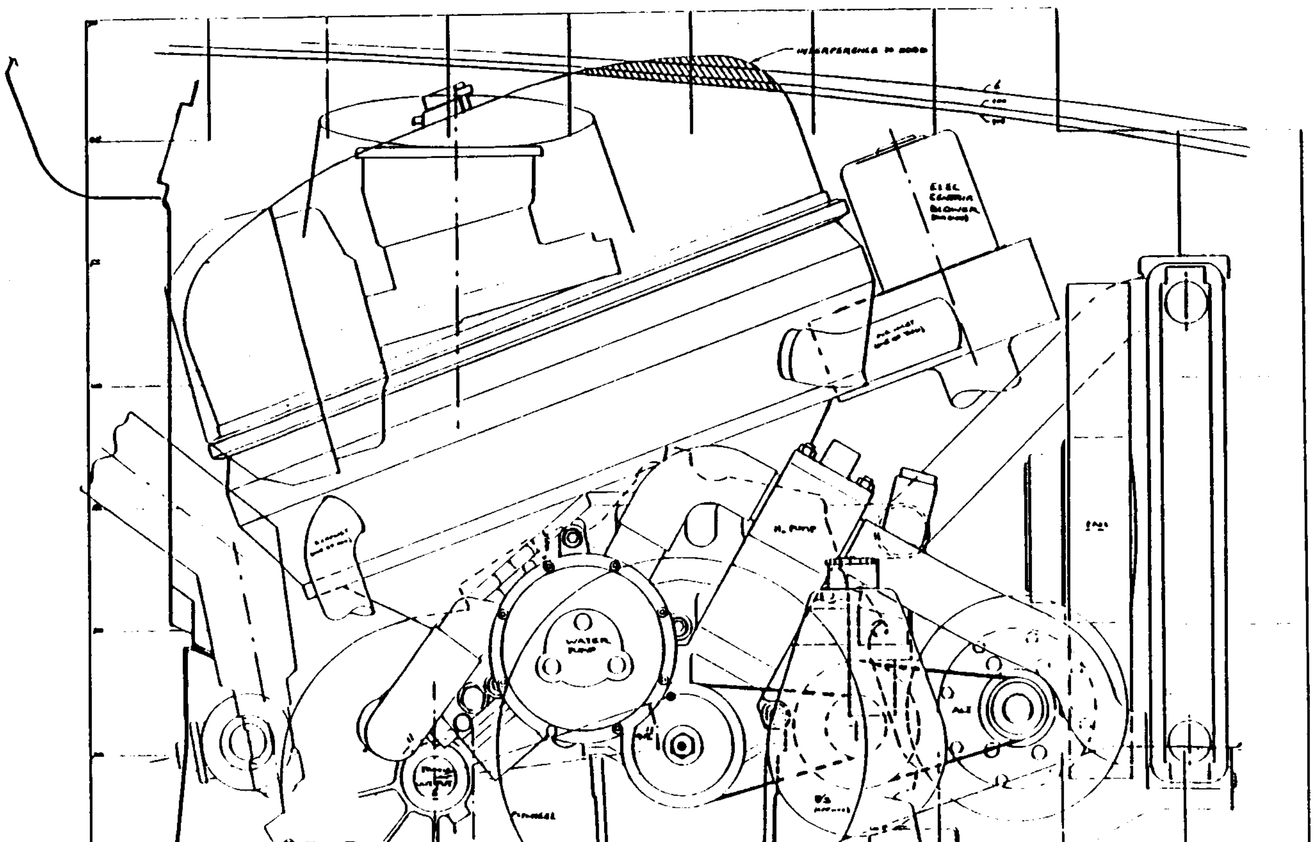

Figure 5-49 Mod II Positioning in Celebrity Engine Compartment

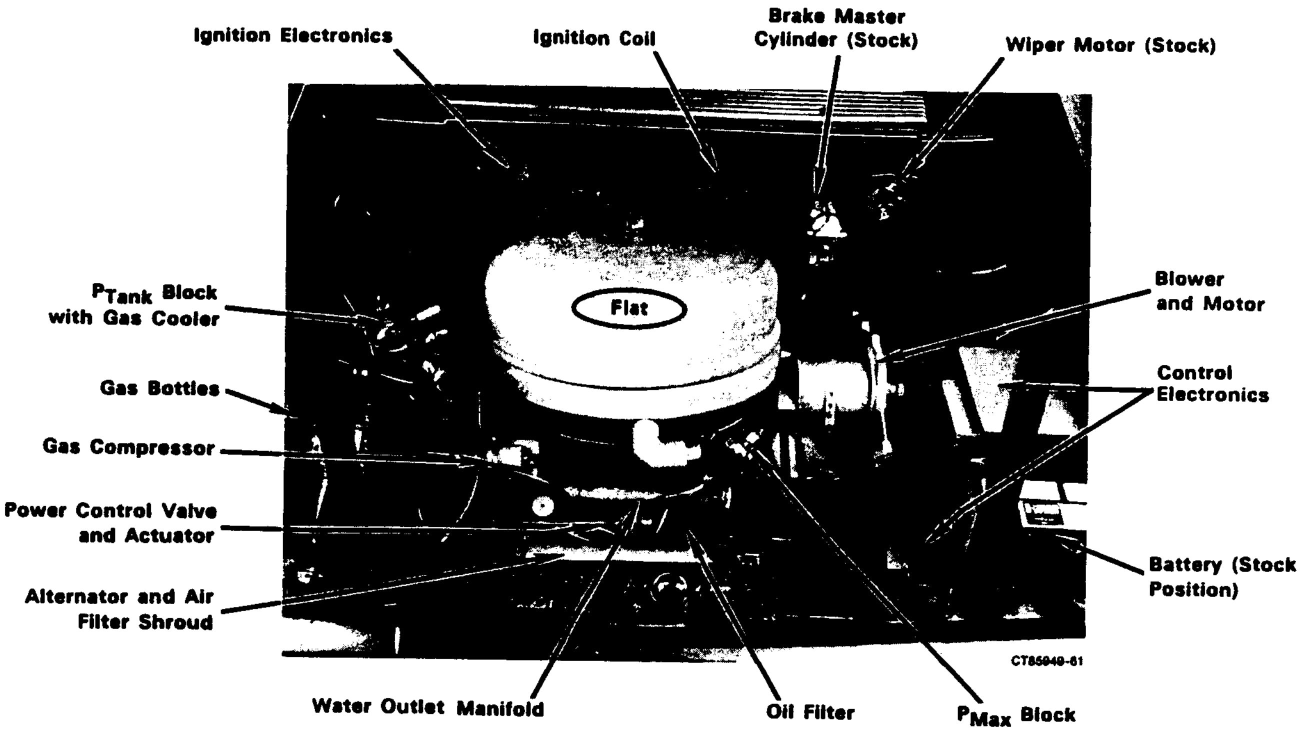

Figure 5-50 Top Front View of Mock-Up Installation in Celebrity

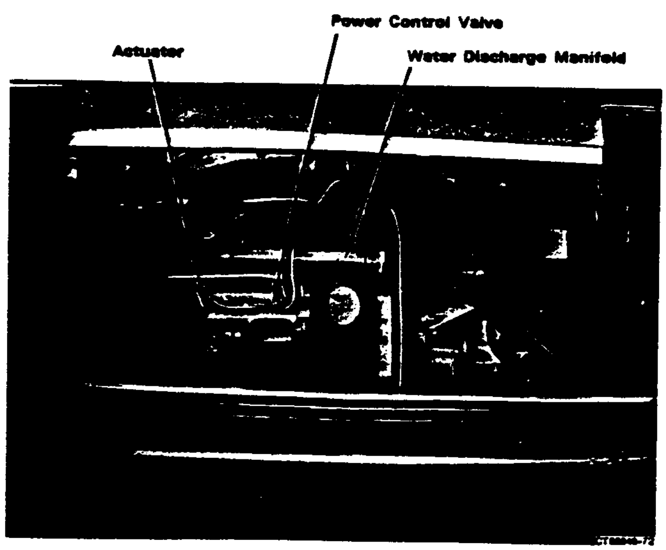

Figure 5-51 Lower Front View of Mod II SES Mock-Up Installation in Celebrity

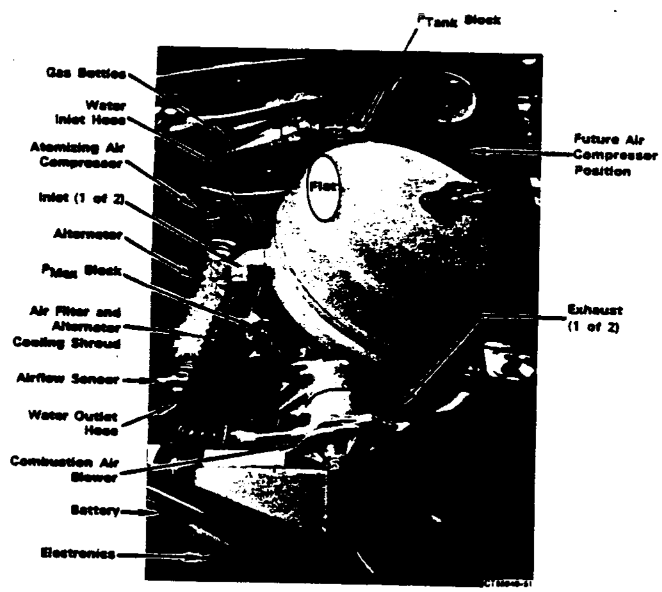

Figure 5-52 Driver-Side View of SES Mock-Up Installation in Celebrity

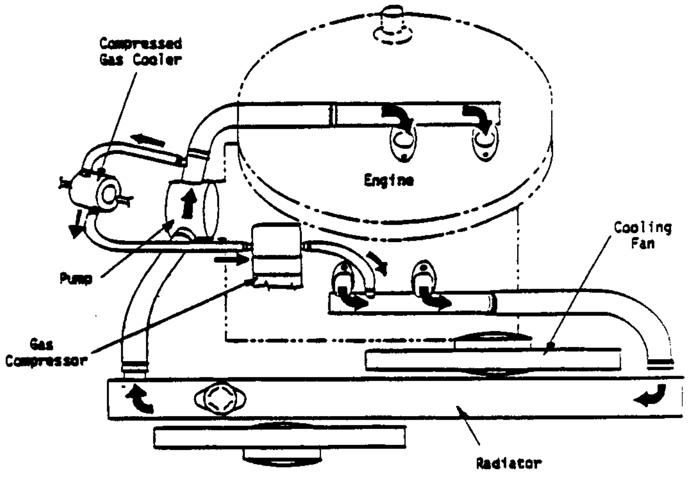

Figure 5-53 Engine Cooling System as Integrated in Celebrity

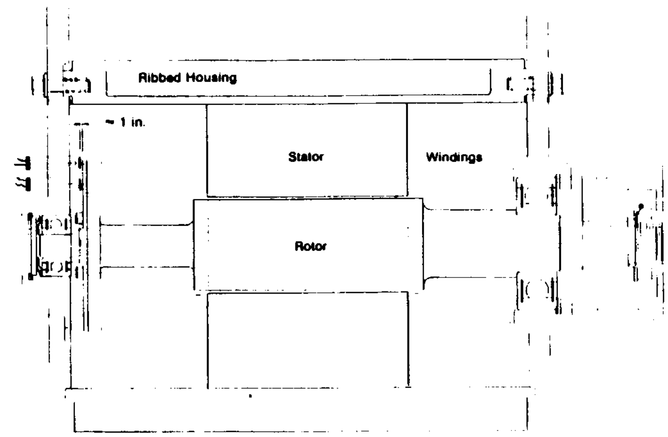

Figure 5-54 Mod II, 12,000 rpm Alternator Assembly

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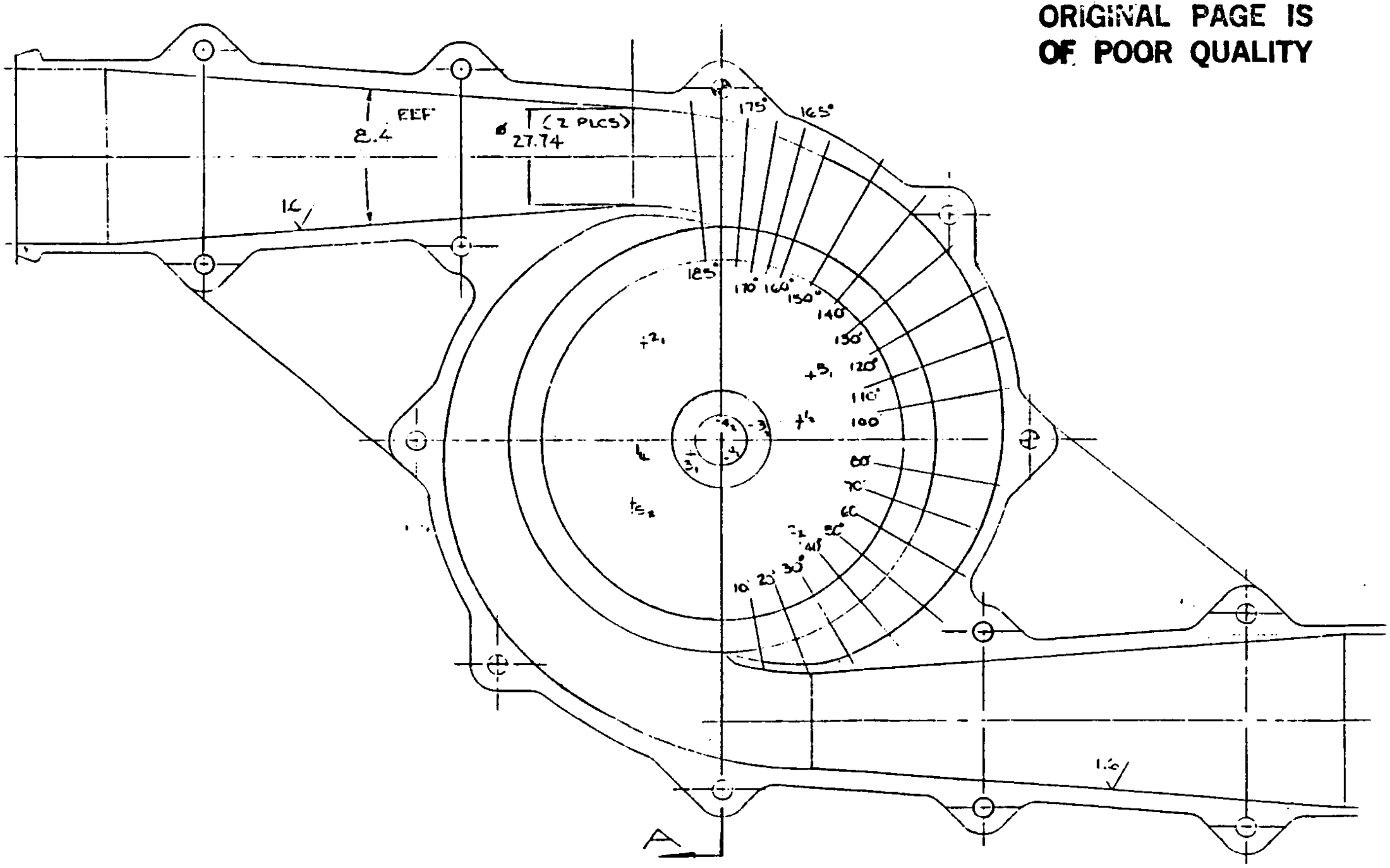

Figure 5-55 Mod II Blower Volute Configuration

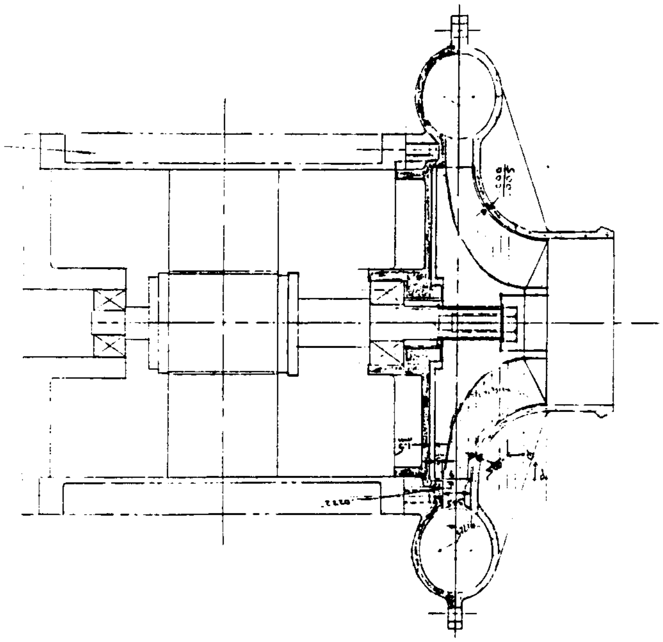

Figure 5-56 Mod II Blower/Motor Assembler

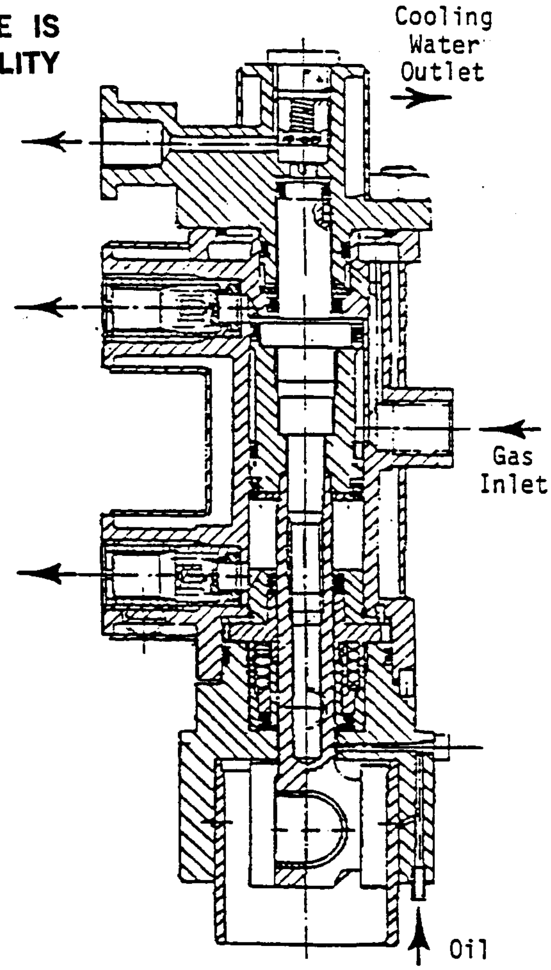

Figure 5-58 Mod II Hydrogen Compressor Design

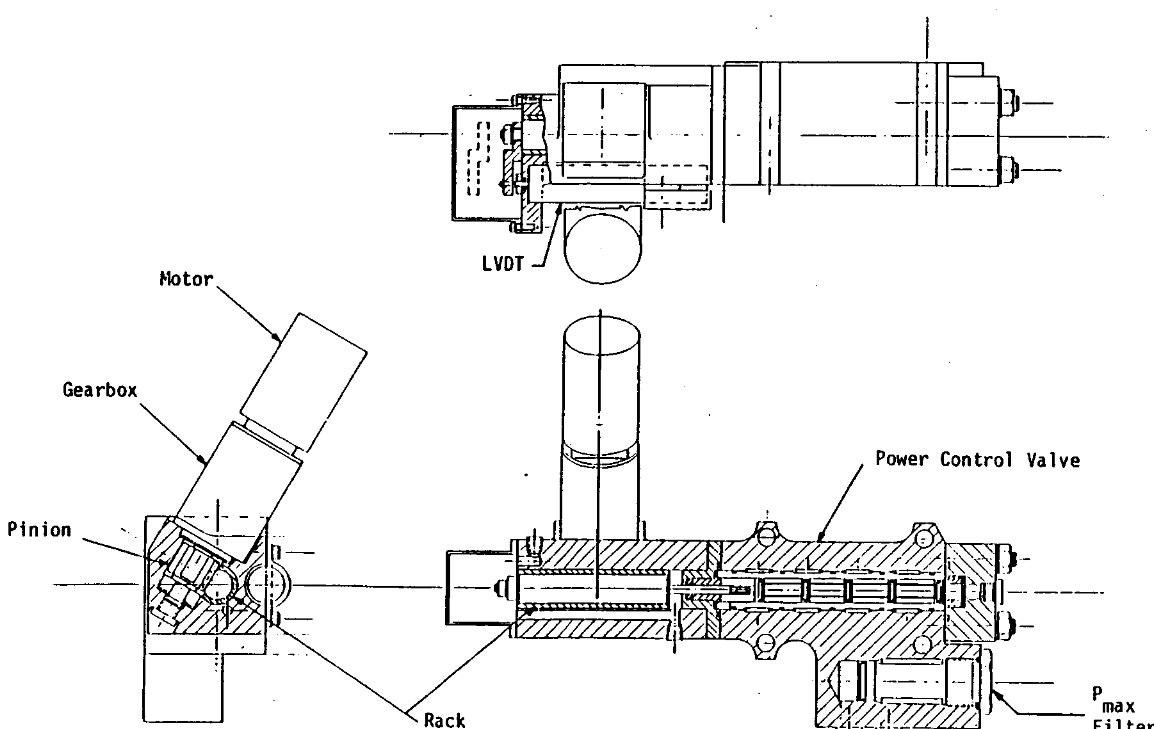

Figure 5-57 Mod II Electric Motor Actuated PCV

VI. TECHNICAL ASSISTANCE

Hot Piston Ring

The concepts for a hot piston ring were reviewed and one was selected for the final design. In this concept the piston ring(s) have a step joint and radial loading is augmented by an internal expander ring. A wavy type spring is also included to prevent axial motion of the piston ring in the groove. To facilitate assembly the the piston dome is split at the piston ring groove and the two parts are connected by a tie bolt.

High temperature pin-on-disk type friction and wear tests were carried out in inert atmospheres at MTI and NASA to identify appropriate material combinations. The best combination was found to be Stellite 6B running against a PS200 coating and it was decided to make the piston ring from Stellite 6B and apply a PS200 coating to the mating surface of the cylinder housing. The final detail design drawings are in the process of completion.

VII. PRODUCT ASSURANCE

Quality Assurance Overview

The status of the ASE program Quality Assurance Reports (QARs) as of June 30, 1985 is presented below:

Open QARs (pending further analysis and/or NASA approval)	271
Closed QARs (total to date)	1194
P-40 QARs	263
Mod I QARs	558
Upgraded Mod I QARs	579
Preliminary Mod II QARs (Durability Rig)	47
Mod II QARs	3
Total QARs in system	1465

Program QAR activity for the second half of 1985 is as follows:

New QARs (for six-month period)	297
P-40 QARs	13
Mod I QARs	46
Upgraded Mod I QARs	208
Preliminary Mod II QARs (Durability Rig)	18
Mod II QARs	3
Test Cell/Equipment Related	20

Mod I QAR Experience

A summary of trend-setting problems documented via the QAR system is presented in Table 7-1 (shown on page 8-2), and Figures 7-1 through 8-4. Problems are defined as items that: 1) cause an engine to stop running; 2) prevent an engine from being started; or, 3) cause degradation in engine performance. Problems that fall into these categories must be minimized to provide acceptable engine performance, and maximize the mean time between failures.

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

Table 7-2 is a summary of the operating times versus failures for all active ASE program Mod I/Upgraded Mod I engines.

Table 7-2 is a summary of the operating times versus failures for all active ASE program Mod I/Upgraded Mod I engines.

TABLE 7-2

OPERATING TIMES VERSUS FAILURES AS OF JUNE 30, 1985

Engine No.	Operation Time (hr)	Mean Operating Time to Failure (hr)
3	1870	110
5	1364	114
6	1576	315
7	3403	851
8	954	477
9	253	63
10	81	-

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TABLE 7-1 - MAJOR PROBLEMS SUMMARY

Established Prior to 6/30/83	% of Total	Reports From 6/30/83 to 12/31/83	% of Total	Reports From 1/1/84 to 6/30/84	% of Total	Reports From 6/31/84 to 12/31/84	% of Total
Moog Valve	82.8	1	4.3	1	4.3	1	4.3
Heater Head	43.4	3	10.0	3	10.0	6	20.0
Check Valves	40.0	3	15.0	2	10.0	3	25.0
Combustion Blower	42.9	6	21.4	6	21.4	1	3.6
Fuel Nozzle	31.0	6	14.3	13	30.9	5	11.9
Igniter	62.5	1	12.5	0	--	0	-
Preheater	30.4	7	30.4	1	4.4	1	4.4
Atomizing Air Comp./ Servo-Oil Pump	50.0	1	8.3	2	16.7	2	16.7
Combustor	31.8	4	18.2	2	9.1	5	22.7
Flameshield	26.1	3	13.0	0	--	4	17.4
PL Seal Assembly	29.7	3	11.1	8	29.6	1	3.7
Crankcase/Bedplate	--	2	40.0	0	--	2	40.0
Piston Rod	--	3	33.3	4	44.5	1	11.1

Mod I hours accumulated prior to 6/30/83 - 2689 % of hours - 23.6
Mod I hours accumulated from 6/30/83 to 12/31/83 - 1881 % of hours - 16.5
Mod I hours accumulated from 1/1/84 to 6/31/84 - 1848 % of hours - 16.2
Mod I hours accumulated from 6/30/84 to 12/31/84 - 2263 % of hours - 19.8
Mod I hours accumulated from 1/1/85 to 6/30/85 - 2731 % of hours - 23.9

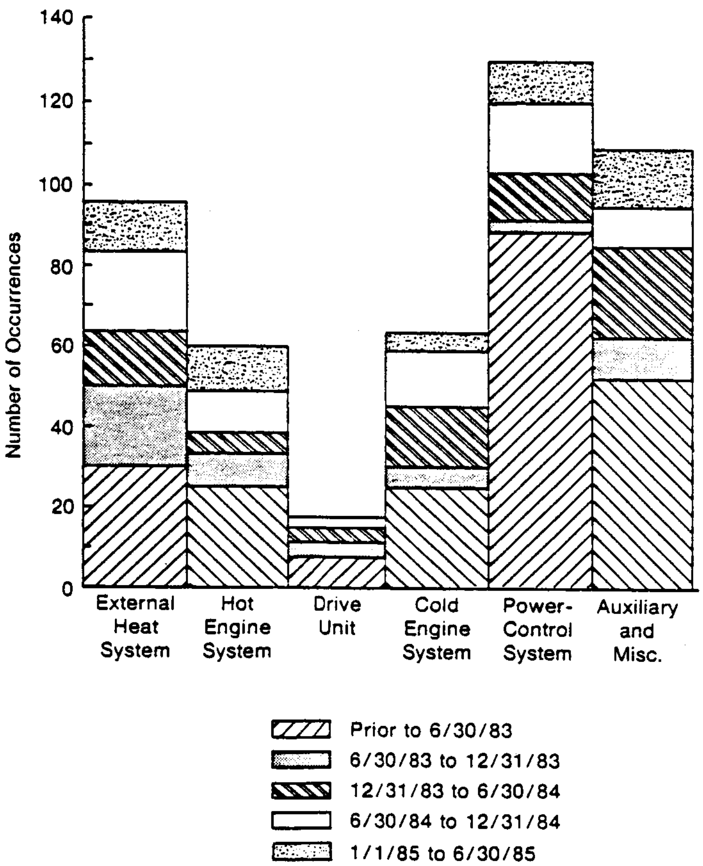

Figure 7-1 Upgraded Mod I Engine Major Failures and Discrepancies Through June 30, 1985

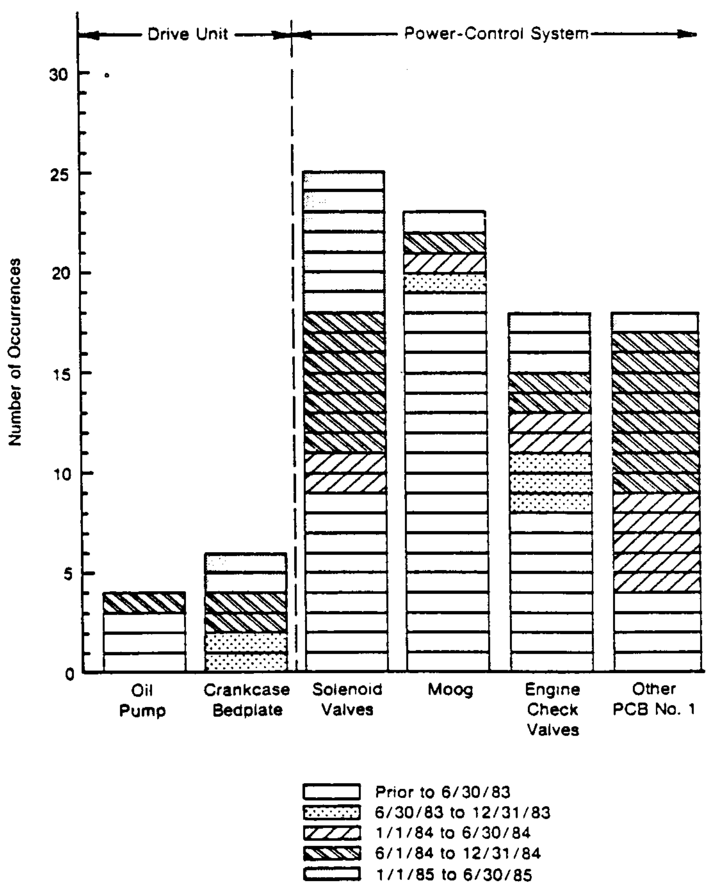

Figure 7-2 Drive Unit and Power-Control System Failures and Discrepancies Through June 30, 1985

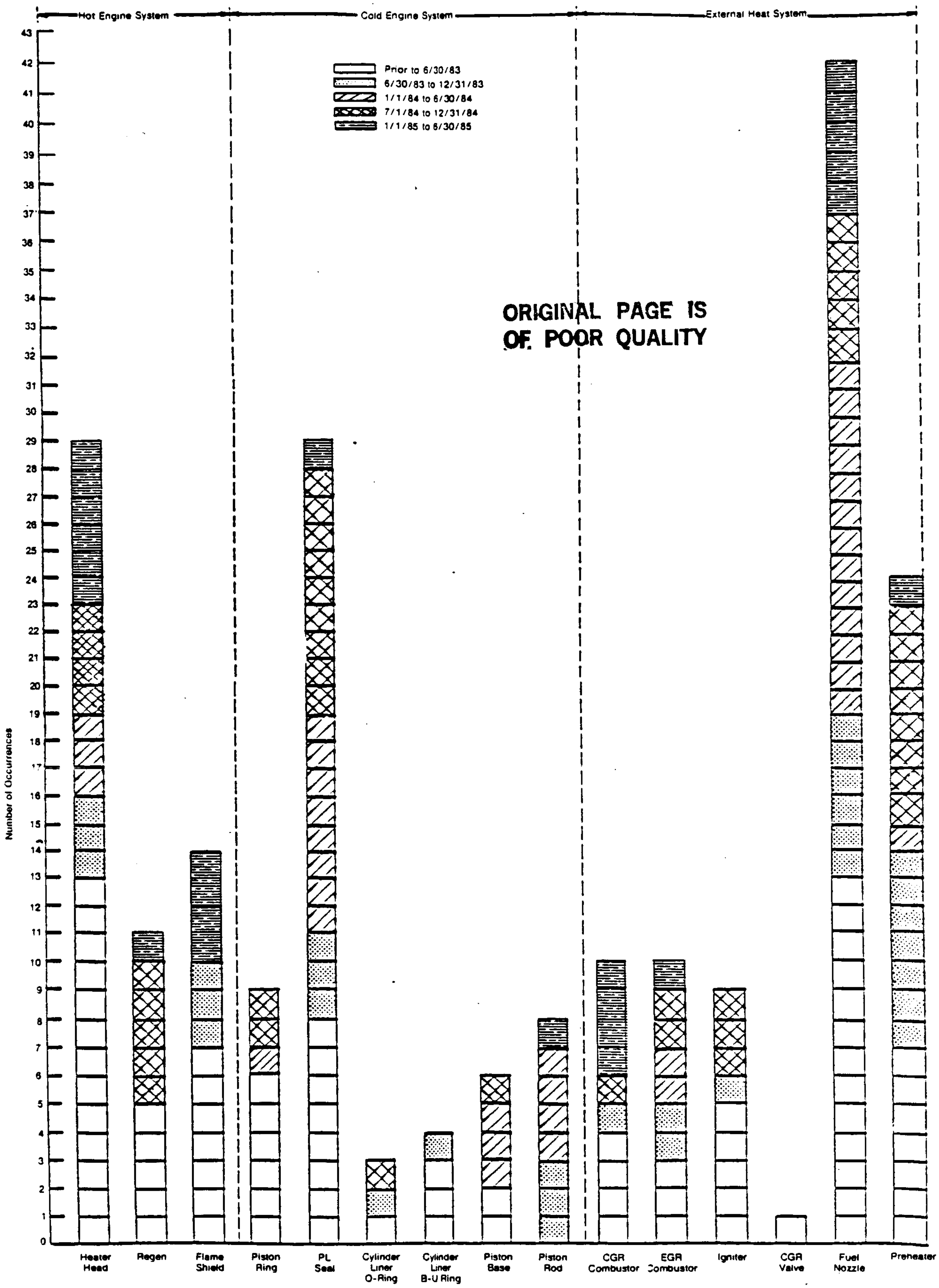

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Figure 7-3 Hot Engine, Cold Engine, and EHS Failures and Discrepancies Through June 30, 1985

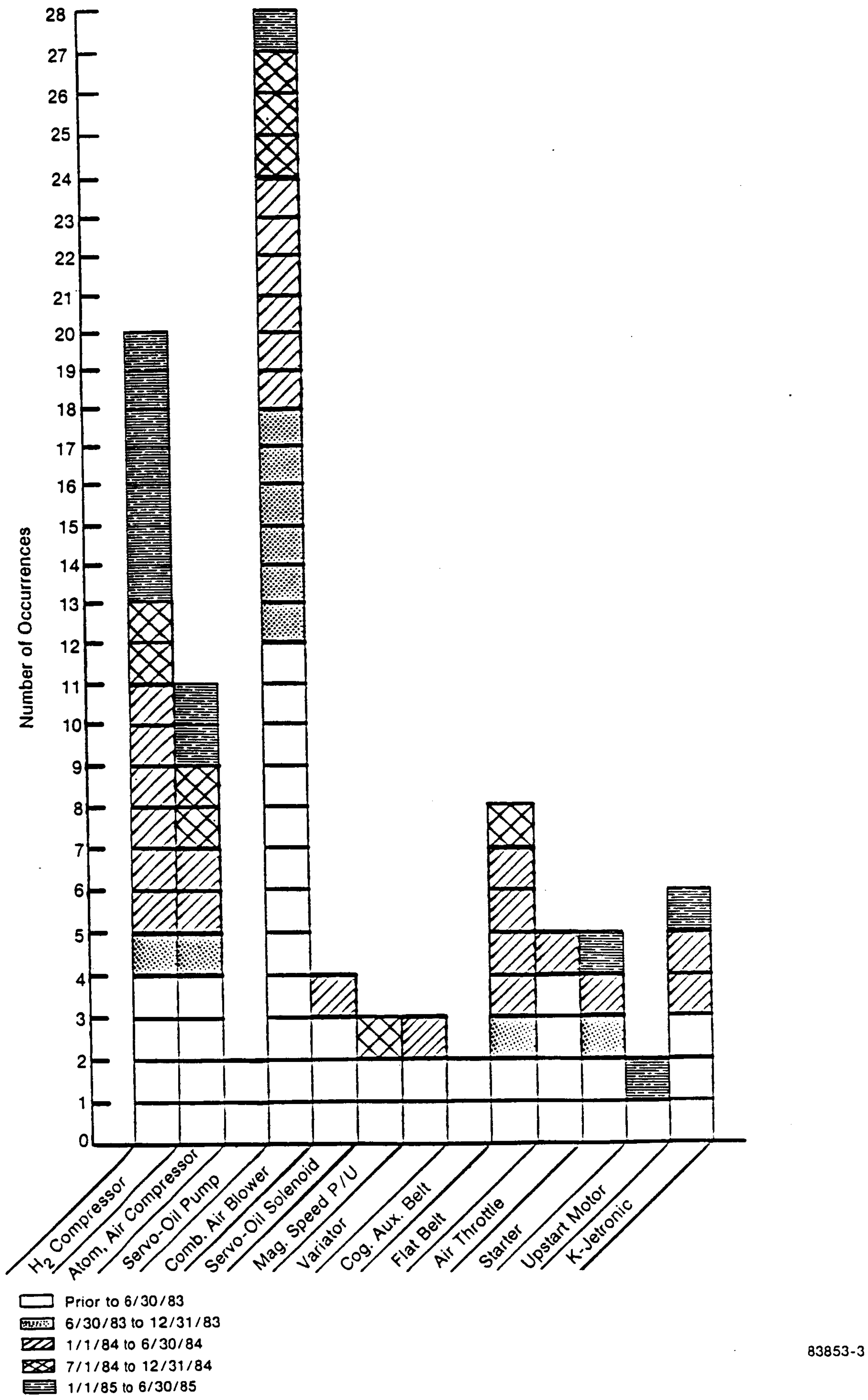

Figure 7-4 Auxiliaries and Miscellaneous Items, Failures and Discrepancies Through June 30, 1985

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15. Supplementary Notes

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16. Abstract

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

This report summarizes development test activities on Mod I engines and the final design of the Mod II engine as a basic Stirling engine and a Stirling engine system. Development activities covered include: conical nozzle and combustion gas recirculation system, a single manifold heater head, single-solid piston rings and a complete set of stirling controls and auxiliaries. Overall program philosophy is outlined and data and test results are presented.

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