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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM

QUARTERLY TECHNICAL PROGRESS REPORT FOR PERIOD: OCTOBER 4 - DECEMBER 31, 1980

Mechanical Technology Incorporated

September 1981

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

for
U.S. DEPARTMENT OF ENERGY
Conservation and Solar Applications
Office of Transportation Programs

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AUTOMOTIVE STIRLING ENGINE DEVELOPMENT PROGRAM TECHNICAL PROGRESS REPORT FOR PERIOD: OCTOBER 4 - DECEMBER 31, 1980

STIRLING ENGINE SYSTEMS DIVISION
Mechanical Technology Incorporated
Latham, New York 12110

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Washington, D.C. 20545
Under Interagency Agreement EC-77-A-31-1040

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1.0 SUMMARY

The DOE/NASA "Automotive Stirling Engine Development Program" has been underway for approximately 31 months. This is the eleventh quarterly report to be issued, and it covers the period of October 4 - December 31, 1980.

Prior reports [1, 2, 3, 4, 5, 6, 8, 9, 10]* discussed component and subsystem effort, the Reference Engine System Design and the program's first and second Stirling engine-powered vehicles, the 1977 Opel and the 1979 AMC Spirit containing United Stirling of Sweden (USS) P-40 Stirling engines. The P-40 engine is the program's baseline Stirling engine originally developed as a stationary laboratory engine. It is heavy and underpowered for the vehicles, but was installed in the vehicles in order to obtain engine-vehicle integration experience, experience with Stirling powered vehicles in testing, and to demonstrate the concept of Stirling engines applied to automobile propulsion.

The technology of Stirling engines as applied to automobile propulsion is presented in MTI's report "Assessment of the State of Technology of Automotive Stirling Engines" [7]. This very comprehensive report gives the background and history of the Stirling engine; it discusses the technology, materials, components, controls, and systems; and it presents a technical assessment of automotive Stirling engines.

The previous quarterly report [10] presented information on the Reference Engine design and the design of the ASE Mod I (Automotive Stirling Engine Model No. I). The report also reviewed component and subsystem development activities, the testing of the baseline P-40 engines, and the progress being made on computer code developments. NASA has approved the updated Reference Engine and ASE Mod I designs, and manufacturing/procurement of the ASE Mod I has started. The ASE Mod I is the program's first Stirling engine designed specifically for automotive use; it is a stepping-stone to the program's final prototype engine, the ASE Mod II.

Program engine operating hours through the end of this quarterly period (through December 31, 1980) reached the following:

Engine No.	Total Hours
ASE 40-4 (High Temperature Endurance Test Engine)	5,620.7
ASE 40-5 (Opel Engine)	250.0
ASE 40-7 (MTI Test Engine)	202.4
ASE 40-8 (Spirit Engine)	245.4
ASE 40-12 (Engine for the Concord)	119.5
Total	6,438.0

*References are listed by number in Section 4.0

2.0 INTRODUCTION

The Automotive Stirling Engine Development Program is directed at the development of technology and knowledge related to the application of Stirling engines to automotive use, and the transfer of Stirling engine technology to the United States. The high efficiency and low emissions potential of the Stirling engine makes it a prime candidate for automotive propulsion. This contract is directed towards developing the necessary technology, by 1984, to demonstrate these potentials.

MTI, the prime and systems contractor, is responsible for overall program management, alternative and high risk component and systems development, engine and vehicle testing and evaluation, computer code development, and transfer of Stirling engine technology to the United States.

The engine development program is based upon the extensive technological achievements, capabilities, and background knowledge in Stirling engines of KB United Stirling (Sweden) AB & Co. (USS), acting as a subcontractor to MTI.

AM General Corporation (AMG), a wholly owned subsidiary of American Motors Corporation, is the subcontractor responsible for automotive selection, design, integration, and evaluation of Stirling engines installed in passenger cars.

2.1 Final Program Objectives

The final Program objectives are to develop and demonstrate, by September 1984, an Automotive Stirling Engine System which when installed in a late-model production vehicle will meet the following objectives:

1. Using EPA test procedures, demonstrate at least a 30% improvement in combined metro-highway fuel economy over that of a comparable production vehicle. The comparison production vehicle will be powered by a conventional spark-ignition engine. Both the Automotive Stirling and spark-ignition engine systems will be installed in identical model vehicles and will give substantially the same overall vehicle driveability and performance. The improved fuel economy will be based on unleaded gasoline of the same energy content (Btu/gal).

   It is intended that identical model vehicles be used for the comparison. However, a difference in inertia weight between the two vehicles is acceptable if the difference results from the substitution of the Automotive Stirling engine system for the spark ignition powertrain system. The transmission, torque converter, and drivetrain may also differ in order to take advantage of Stirling engine characteristics.

2. Show the potential of gaseous emissions and particulate levels less than the following: $NO_x = 0.4$, $HC = .41$, $CO = 3.4$ gm/mile and a total particulate level of 0.2 gm/mile after 50,000 miles.

   The potential need not be shown by actual 50,000 mile tests, but can be shown by Contractor projections based on available engine, vehicle, and component test data and emissions and particulate measurements taken at EPA using the same fuel as used for the EPA fuel economy measurements.

The emissions and particulate measurements will be based on EPA procedures for the metro cycle and will use the same fuel used for fuel economy measurements.

In addition to the above objectives that are to be demonstrated quantitatively, the following design objectives are considered goals of the program.

1. Ability to use a broad range of liquid fuels from many sources, including coal and shale oil. This objective will be pursued initially in the combustor development effort and later in engine and vehicle testing. The candidate alternative fuels and their characteristics, to be considered in this Program, will be identified based on the DOE Alternative Fuels effort. Until these specific fuels and their characteristics are identified for inclusion in the Program, diesel fuel, gasohol, kerosene, and No. 2 heating oil will be used as a representative range of alternate fuels. Engine tests with the alternate fuels will not be initiated for the ASE Mod I and ASE Mod II engines until satisfactory operation, performance, and emissions have been achieved on the baseline fuel -- gasoline. Testing will then be conducted with the selected alternative fuels to determine the extent of any detrimental effects on engine operation, performance, emissions, or fuel economy and to determine the degree of modifications or adjustments that might be required in switching from one fuel to another.

2. Reliability and life comparable with powertrains currently on the market.

3. A competitive initial cost and a competitive life-cycle cost with a comparable conventionally-powered automotive vehicle.

4. Acceleration suitable for safety and consumer considerations.

5. Noise and safety characteristics that meet the currently legislated or projected Federal Standards for 1984.

2.2 Program Major Milestones

Progress toward achieving these Final Program Objectives, which will be demonstrated by dynamometer and vehicle testing, will be assessed at several points in the program. Specific milestones will be:

1. ASE Mod I Basic Engine design freeze prior to March 31, 1980.

2. Dynamometer characterization of the first build of ASE Mod I engine at the Contractor's facility prior to September 30, 1981.

3. Dynamometer characterization of ASE Mod I (updated) engine prior to September 30, 1982.

4. ASE Mod I engine system test in a vehicle at EPA prior to September 30, 1983.

5. Dynamometer characterization of ASE Mod II engine at the Contractor's Facility prior to September 30, 1983.

6. Complete ASE Mod II engine system test in a vehicle at EPA prior to September 30, 1984.

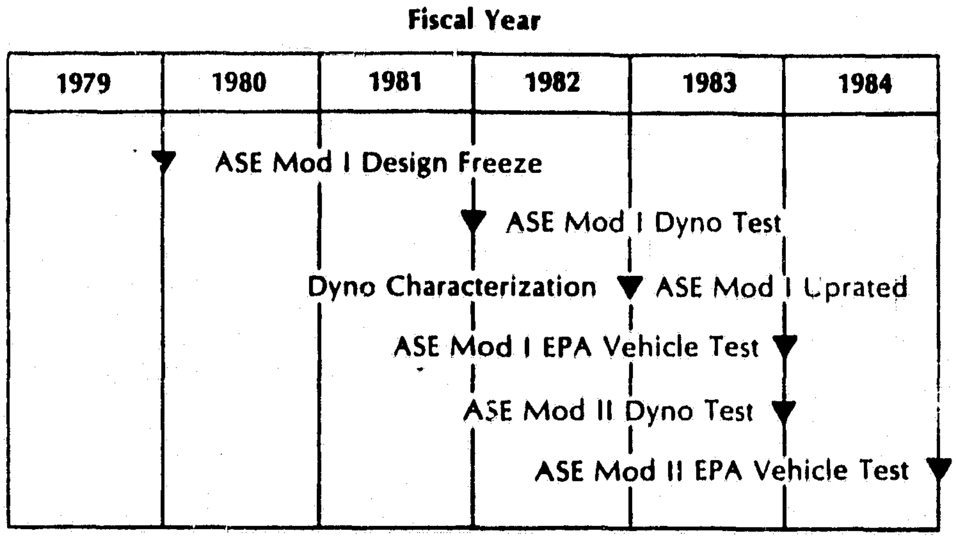

Figure 2.0-1 Program Milestones

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2.3 Task Description

MAJOR TASK 1 - REFERENCE ENGINE

This task is intended to guide component, subsystem, and engine system development. A reference engine system design will be generated and continually updated to reflect the best contemplated approaches and the latest technology to meet the final program objectives. The reference engine system will be the focal point to guide development, will be based on approved engine system concepts, and will include anticipated 1990 vehicle power level and size for equivalent spark ignition, diesel, and stratified charge engines.

Task 1.1 Initial Technology Assessment

A comprehensive technical assessment will be made of the present status and level of technology of Stirling engines as candidates for automotive power plants. This assessment will be directed at, but not limited to, the status of United Stirling of Sweden's engine design and development technology. When completed, the Initial Technology Assessment will be used as a basis for a detail study and reevaluation of the overall technical program plan.

- This task has been completed through the issuance of a final report in September 1979.

Task 1.2 Reference Engine System Design

A series of conceptual Stirling engine system designs will be produced to meet the final program objectives. Analyses and drawings will be prepared in sufficient detail to be able to assess the potential advantages and disadvantages of the candidate concepts, including material costs. Available transmission technologies, accessory systems, auxiliary systems, and alternate power control systems will be evaluated. Transmission and drive train arrangements and vehicle installation will be assessed. Sufficient design and analyses will be performed to establish performance requirements of the engine and components to meet the required vehicle performance and the final program objectives.

The Reference Engine System Design (RESD) will be the best engine design that can be generated at any given time that will provide the best possible fuel economy and will also meet or exceed all other Final Program Objectives. It will be designed to meet the requirements of the projected reference vehicle, which will be representative of the class of vehicles for which the engine might first be produced. It will utilize all new technology that can reasonably be expected to be developed by 1984 and which is judged to provide significant improvement relative to the risk and cost of its development.

In general, all technology advancements that are to be worked on in the Program will be incorporated into the Reference Engine System Design (RESD) and their payoff will be quantified prior to initiation of the technology development. Since there may be more than one attractive technology option for a given component, subsystem, or system which merits development, it may be desirable and necessary to generate one or more alternative designs in addition to the primary design of the RESD. Such alternative designs might range from incorporating high risk, high pay-off alternatives, to providing more conservative reduced risk back-up approaches. As the development program proceeds, actual test experience may eliminate one or more of these back-up approaches, or it may dictate a reduction in performance relative to initial expectations.

- The RESD was generated early in the Program. It will be continually updated to reflect development experience and technology growth.

MAJOR TASK 2 - COMPONENT AND SUBSYSTEM TECHNOLOGY DEVELOPMENT

Development activities will be conducted on all required component and subsystem development tasks as guided by the Reference Engine System Design (RESD) to support the various Stirling Engine Systems (SES) being developed. The component and subsystem development activities will include conceptual and detail design and analyses, hardware fabrication and assembly, and component and subsystem testing in laboratory test rigs. When an adequate performance level is achieved, the component and/or subsystem design will be configured for in-engine testing and evaluation in appropriate engine dynamometer and vehicle test installations. Design efforts will be carried out with consideration of cost and manufacturing feasibility.

Effort will proceed according to an overall Program Component and Subsystem Development Plan and detailed, individual, Component and Subsystem Test Plans, which will be submitted to the NASA Project Manager for review and approval. At the completion of each significant component or subsystem development effort, a report will be prepared and submitted to the NASA Project Manager. These reports will define the designs, define new fabrication techniques developed, describe the development effort, and present the results.

The following development activities will be carried out to advance the technology in terms of durability, reliability, performance, cost, and fabrication:

- Task 2.1 Combustion Technology Development
- Task 2.2 Heat Exchanger Technology Development
- Task 2.3 Materials Development
- Task 2.4 Mechanical Component Development of Seals
- Task 2.5 Mechanical Component Development of the Engine Power Chain
- Task 2.6 Controls Technology Development
- Task 2.7 Auxiliaries Development
- Task 2.8 United Stirling Project Support
- Task 2.9 United Stirling Component Development (Directed towards ASE Mod I and ASE Mod II)

  - Effort on components and subsystems are underway.

MAJOR TASK 3 - TECHNOLOGY FAMILIARIZATION (BASELINE ENGINE)

The existing USS P-40 Stirling engine will be used as a baseline engine for Stirling engine familiarization and as a test bed for component and subsystem performance improvement. It will also be used to evaluate current engine operating conditions and component characteristics, and to define problems associated with vehicle installation. Four P-40 Stirling engines will be built and delivered to the United States team members, with one installed in a 1979 AMC vehicle. A fifth P-40 Stirling engine will be built and installed in a 1977 Opel sedan.

The baseline P-40 engines will be tested in dynamometer test cells as well as in the automobiles. Test facilities will be planned and constructed at MTI to accomodate the entire program.

Task 3.1 - Baseline Engine (P-40)

USS will manufacture four P-40 engines, including spare parts. Engine/dynamometer testing will include full and part power operation, transient and cyclic operation, start-stop cycles, and endurance testing. Complete engine performance maps of fuel consumption, emissions, power, and torque versus engine speed over the full range of engine operating pressure levels will be obtained over the entire anticipated range of operating heater head temperatures, combustor flows, inlet temperatures, coolant temperatures, coolant flows, and coolant inlet temperatures.

Tests will be run with the complete Stirling engine system as designed (with all auxiliaries installed and operating off engine power). Where appropriate, selected auxiliaries and/or ducting may be simulated, or compensated for. Tests will also be run with all auxiliaries removed and their functions provided by test facilities, or compensated for.

AMG will modify an AMC vehicle for the P-40 engine, in the first year of the program, thereby gaining experience and knowledge on the integration problems and requirements associated with the installation of a Stirling engine in a passenger car. Limited vehicle testing will be conducted by AMG to establish baseline vehicle-affected engine performance, such as: fuel consumption, emissions, and under-hood environment. The vehicle installation and test is designed to familiarize AMG and other team members with a Stirling engine equipped vehicle and its performance and operation. It will also establish baseline performance for the total program, including durability.

- P-40 engines have been delivered by USSw, installed in vehicles and dynamometer test stands, and testing is well underway.

Task 3.2 - Facilities

The test facilities and equipment necessary to completely evaluate engines and components will be designed, built, and procured at MTI. It is anticipated that this will include installation at MTI of two engine test cells with appropriate data acquisition equipment and five component test cells to be used for component development purposes.

- Facility construction and installation are progressing on schedule.

Task 3.3 - P-40/Opel Test Vehicle

One P-40 engine will be manufactured and installed in a 1977 Opel Rekord 2100D diesel engine-powered automobile to establish baselines for comparison with other program generated Stirling engine-powered automobiles. Vehicle tests will be conducted on a chassis dynamometer and by road testing, in order to measure parameters such as fuel economy, emissions, driveability, and noise.

- This task was completed, and reported in January, 1979. [6,7]

MAJOR TASK 4 - ASE MOD I ENGINE SYSTEM

A first generation Automotive Stirling Engine (ASE) will be developed. ASE Mod I will use the United Stirling P-40 and P-75 engines as a basis for improvement. The prime objective will be to improve power density and overall engine performance. The ASE Mod I engine will be an experimental version of the RESD. It will be limited by the technology which can be confirmed in the time available. It need not achieve any specific fuel economy improvement, but will be utilized to verify the basic RESD and to serve as a stepping stone toward the ASE Mod II engine. It will provide an early indication of the potential to meet the Final Program Objectives. A preliminary design and analysis will be made of the engine and its installation in an automobile which will include the preparation of detailed layout drawings defining critical features, dimensions, materials, and fabrication techniques. Appropriate analyses will be performed to predict engine system and component performance, in-vehicle performance of the engine system, and appropriate stress and thermal loads. Potential problem areas will be identified.

A Design Review Meeting will be held with NASA to review the results of the engine preliminary design. Information to be presented at the design review will include layout drawings, materials, fabrication techniques, and the results of performance, stress, and thermal analyses.

Seven engines and adequate spares will be manufactured by USS and the engines will be tested in dynamometer test cells to establish performance, durability, and reliability. Continued testing and development may be necessary in order to meet the preliminary design performance predictions. One additional ASE Mod I engine will be manufactured in the United States; USS drawings will be used, but United States vendors will be used to manufacture the engine.

Engine/dynamometer testing will include full and part power operation, transient and cyclic operation, start-stop cycles, and endurance testing. Complete engine performance maps of fuel consumption, emissions, power, and torque, versus engine speed over the full range of engine operating pressure levels, will be obtained over the entire anticipated range of operating heater head temperatures, combustor flow, inlet temperatures, coolant temperatures, coolant flows, and coolant inlet temperatures.

Tests will be run with the complete Stirling engine system as designed (with all auxiliaries installed and operating off engine power). When appropriate, selected auxiliaries and/or ducting may be simulated or compensated for. Tests will also be run with all auxiliaries removed and their functions provided by test facilities or compensated for. The full range of engine transient characteristics will be determined, including startup, shutdown, and typical power and speed transients. Tests will be run both with and without the selected vehicle transmission system, as appropriate.

Four production vehicles will be procured and modified to accept the manufactured engines and the engines will be installed in the vehicles. One of the four vehicles will be an engineering-evaluation front-wheel drive vehicle. Tests will be conducted on the engine-powered automobiles to establish engine-related driveability, fuel economy, noise, emissions, and durability/reliability. Tests will be performed under various steady state, transient, and environmental conditions. One vehicle will be delivered to EPA prior to March 31, 1983, for vehicle assessment by EPA.

- A design review was held in May 1980 and the design was approved by NASA.

- Assembly and test activities are underway to have the first Mod I SES available for acceptance test on March 31, 1981.

MAJOR TASK 5 - ASE MOD II ENGINE SYSTEM

The second generation engine will be designed, fabricated and tested. It will be power rated according to the reference engine system studies, using the first generation engine system as the basis for improvement. The prime objective will be to upgrade the first generation engine system to improve efficiency, and to improve durability and reliability.

Only high confidence level component and subsystem developments will be used. The design will reflect the use of automotive engineering design and fabrication techniques to the maximum extent possible. Emphasis will be on performance and durability/reliability. The ASE Mod II engine could differ from the RESD by the use of small quantity fabrication techniques and special provisions for instrumentation, parts replacement, and servicing.

A preliminary design and analysis will be made of the engine and its installation in an automobile which will include the preparation of detailed layout drawings defining critical features, dimensions, materials, and fabrication techniques. Appropriate analyses will be performed to predict engine system and component performance, in-vehicle performance of the engine system, and appropriate stress and thermal analyses. Potential problem areas will be identified.

A Design Review Meeting will be held with NASA to review the results of the engine preliminary design. Information to be presented at the design review will include layout drawings, materials, fabrication techniques, and the results of performance, stress, and thermal analyses.

Five engines and adequate spares will be manufactured and the engines will be tested in dynamometer test cells to establish performance, durability, and reliability. Continued testing and development may be necessary in order to meet the preliminary design performance predictions.

Engine/dynamometer testing will include full and part power operation, transient and cyclic operation, start-stop cycles, and endurance testing. Complete engine performance maps of fuel consumption, emissions, power, and torque, versus engine speed over the full range of engine operating pressure levels, will be obtained over the entire anticipated range of operating heater head temperatures, combustor flows, inlet temperatures, coolant temperatures, coolant flows, and coolant inlet temperatures.

Tests will be run with the complete Stirling engine system as designed (with all auxiliaries installed and operating off engine power). When appropriate, selected auxiliaries and/or ducting may be simulated or compensated for. Tests will also be run with all auxiliaries removed and their functions provided by test facilities or compensated for. The full range of engine transient characteristics will be determined, including startup, shutdown, and typical power and speed transients. Tests will be run both with and without the selected vehicle transmission system, as appropriate.

Three late-model front-wheel drive production vehicles will be procured and modified to accept the manufactured engines and the engines will be installed in the vehicles. Tests will be conducted on the engine-powered automobiles to establish engine-related driveability, fuel economy, noise, emissions and durability/reliability. Tests will be performed under various steady state, transient and environmental conditions. One vehicle will be delivered to EPA prior to April 30, 1984 for EPA assessment of the vehicle to meet the Final Program Objectives of fuel economy and exhaust emissions.

- The design effort will start in FY 1981.

MAJOR TASK 6 - PROTOTYPE ASE SYSTEM STUDY

A study will be undertaken to describe the effort required to bring the Automotive Stirling Engine from its expected state of development in September 1984 to the start of production engineering. Engine production cost, life cost, operating condition, in-service maintenance requirements, and vehicle-imposed loads and constraints will be studied. Consideration will be given to mass production fabrication techniques. In addition, the prototype ASE system will incorporate the final levels of technology necessary before going into production.

The results of this study will be incorporated into a plan which will be submitted to the NASA Project Manager by September 30, 1983. The plan will describe the development steps required, the schedule of events, and the estimated cost. In addition, the development risk will be assessed and the plan will include supportive manufacturing and cost information. The plan will form part of the basis for a Government decision regarding the extent of its support, if any, for system development activities beyond the scope of this contract.

- This task will start in 1982.

MAJOR TASK 7 - COMPUTER PROGRAM DEVELOPMENT

Analytical tools will be developed which are required to simulate and predict engine performance, as well as to aid in the design, development, optimization, and evaluation of engine hardware. This effort will include the development of three comprehensive computer programs specifically tailored to: (1) predict Stirling engine system steady state cyclical performance over the complete range of engine operations; (2) optimize the Stiring engine system to maximize and/or minimize specified physical and/or performance characteristics while satisfying given system constraints; (3) evaluate the effects of Stirling engine control system selection on engine transient response to arbitrary power changes. The computer programs will be structured to be user oriented and to have high portability.

The computer programs will be designed and structured to predict the performance of given engine and component configurations and should not be confused with engine and component design computer programs that are used to design physical hardware (i.e., heater head designs, regenerator designs, bearing load computations, stress analysis, dynamics, etc.).

In addition to delivering the source codes for the library of computer programs developed, complete documentation will be provided to describe the logic structure, detailed theory, assumption, operating procedures, demonstrated validity, ranges of applicability, sample problems, etc. for each program. In addition to delivering the final, fully verified version of each program, interim partially verified versions of each program will also be delivered.

The programs will be improved and verified on a continuing basis throughout the course of the program, using data from component, subsystem, and engine system test activities. The test data so utilized will be identified and provided for each program.

In addition to the engine configurations to be specifically investigated, the performance prediction and optimization programs will be correlated against the three engine configurations and performance data to be supplied by NASA.

It is anticipated that several engine systems will be investigated over the course of the contract. The engine performance prediction program will allow for either separate or simultaneous engine/drive system analysis. In addition to the determination of engine piston dynamics, the drive system modeling will include evaluation of the bearing, slip, windage, and pumping losses associated with each drive system concept.

- Effort is underway and the first programs are expected to be completed early in 1981.

MAJOR TASK 8 - TECHNICAL ASSISTANCE

Technical assistance to the Government, as requested, will be provided pursuant to the Technical Direction Clause of the contract. This effort will include: Stirling engine and/or vehicle systems for DOE/NASA demonstration purposes; models and displays for use at Government and professional society technical meetings; computer program assistance to evaluate various NASA specified engine modifications, parametric engine variations and engine operating modes; training of personnel in the operation, assembly and maintenance of Stirling engine systems and vehicles delivered to NASA; appropriate communication media including brochures, audio-visual materials, other literature, and independent studies after approval from NASA.

- Effort is underway pursuant to specific Technical Directives received from NASA-LeRC.

MAJOR TASK 9 - PROGRAM MANAGEMENT

This task defines the total program control, administration and management, including reports, schedules, financial activities, test plans, meetings, reviews, seminars, training, and technology transfer.

Task elements include:

- Program management.
- Technical direction.
- Product Assurance.
- Monitoring of technical and financial progress.
- Report preparation, publication and distribution.
- Preparation of test plans, work plans, design reviews, etc.
- Coordination of monthly meetings, review meetings, etc.
- Transfer of technology to the United States.
- Training of personnel.
- Seminars and technical society presentations.
- Attendance and coordination of government meetings and presentations
- Engineering drawings and installation, operation and maintenance manuals.
- Other items related to overall program management and control.

Figure 2.0-2 is the Work Breakdown Structure of the Automotive Stirling Engine Development Program at the level of reporting to NASA.

- Effort will continue throughout the program.

1.0 REFERENCE ENGINE

- 1.1 Initial Technology Assessment
- 1.2 Reference Engine System
    - 1.2.1 Project Engineering
    - 1.2.2 USS Engineering Assistance
    - 1.2.3 AMG Engineering Assistance
    - 1.2.4 Reference Engine Analysis
    - 1.2.5 Advanced Concepts Studies

2.0 COMPONENT & SUBSYSTEMS DEVELOPMENT

- 2.1 Combustion Technology Development
- 2.2 Heat Exchanger Technology Development
- 2.3 Materials Development
- 2.4 Mechanical Component Development (Seals)
- 2.5 Mechanical Component Development (Power Chain)
- 2.6 Controls Technology Development
- 2.7 Auxiliaries Development
- 2.8 USSw Projects
- 2.9 USSw Component & Subsystems Development
    - 2.9.1 Baseline Engine
    - 2.9.2 ASE Mod I Engine
        - 2.9.2.1 External Heat System
        - 2.9.2.2 Hot Engine System
        - 2.9.2.3 Cold Engine System
        - 2.9.2.4 Engine Drive System
        - 2.9.2.5 Controls & Auxiliaries
        - 2.9.2.6 Stirling Engine Systems
        - 2.9.2.7 Vehicle Applications
    - 2.9.3 ASE Mod II
        - 2.9.3.1 SES Component/Subsystems Development
            - 2.9.3.1.1 External Heat System
            - 2.9.3.1.2 Hot Engine System
            - 2.9.3.1.3 Cold Engine System
            - 2.9.3.1.4 Engine Drive System
            - 2.9.3.1.5 Controls & Auxiliaries
        - 2.9.3.2 Materials Development
        - 2.9.3.3 P-40 Annular Regenerator
        - 2.9.3.4 Full-Scale Mod II Involute Heater
        - 2.9.3.5 BSE Mod I Components Testing

Figure 2.0-2 Work Breakdown Structure

3.0 TECHNOLOGY FAMILIARIZATION

3.1 P-40 Program

3.1.1 Project Engineering
3.1.2 Mfg. and Assemble Engines
3.1.3 Evaluate Engines
3.1.4 Evaluate Engine/1979 Spirit

3.2 Test Facility at MTI

3.2.1 Project Engineering
3.2.2 Design of Integrated Facility
3.2.3 Equip Engine Test Cell
3.2.5 Construct Integrated Facility
3.2.7 Maintenance & Repair

3.3 P-40 Opel

4.0 ASE Mod I

4.1 Project Engineering
4.2 Analysis & Design
4.3 Manufacture Engines

4.3.1 External Heat System
4.3.2 Hot Engine System
4.3.3 Cold Engine System
4.3.4 Engine Drive System
4.3.5 Controls & Auxiliaries
4.3.6 Stirling Engine Systems

4.4 Assembly & Acceptance Test
4.5 Engine Test Program

4.5.1 Engine #1
4.5.2 Engine #2
4.5.3 Engine #6

4.6 Vehicle Test Program

4.6.1 Engine #3/1979 Spirit
4.6.2 Engine #5/Vehicle Evaluation (AMG)
4.6.3 Engine #4/Vehicle Evaluation (MTI)
4.6.4 1981 FWD Vehicle/Engine #7 Evaluation

4.7 USA Engine

4.7.1 Manufacture/Procurement
4.7.2 Assembly & Test

Figure 2.0-2 Work Breakdown Structure (Cont'd)

5.0 ASE Mod II

5.1 Project Engineering
5.2 Analysis & Design
5.3 Manufacture Engines

5.3.1 External Heat System
5.3.2 Hot Engine System
5.3.3 Cold Engine System
5.3.4 Engine Drive System
5.3.5 Controls/Auxiliaries
5.3.6 Stirling Engine System

5.4 Assemble & Acceptance Test
5.5 Engine Test Program

5.5.1 Engine #1 Evaluation (USSw)
5.5.2 Engine #4 Evaluation (MTI)

5.6 Vehicle Test Program

5.6.1 Vehicle/Engine #3 Test (USSw)
5.6.2 Vehicle/Engine #2 Test (AMG)
5.6.3 Vehicle/Engine #5 Test (MTI)

6.0 MANUFACTURING & MARKET STUDIES

7.0 COMPUTER PROGRAM DEVELOPMENT

8.0 TECHNICAL ASSISTANCE

9.0 PROGRAM MANAGEMENT

9.1 MTI Program Management
9.2 AMG Program Management
9.3 USS Program Management

Figure 2.0-2 Work Breakdown Structure (Concluded)

3.0 PROGRESS SUMMARIES

The description of the work to be performed under the contract was presented in Section 2.0. This section of the repor presents the details of the work accomplished on each task during the period of October 4 - December 31, 1980.

MAJOR TASK 1 - REFERENCE ENGINE

In October, MTI made a technical presentation to NASA to discuss the results of the Advanced Concept Study. Later in December, several items were identified which will be included in the NASA review in March, 1981. They were:

- The structural integrity of the heater head and piston dome will be investigated to assure that the design is consistent with the Reference Engine criteria. This item was approximately 80% complete by the end of December.
- Structural analyses will be utilized to update the conduction path resistance values.
- The updated First-Order Code will be used to update the conduction values, to revalidate the Reference Engine base input data, and to revise the thermodynamic predictions.
- The net engine performance and vehicle mileage predictions will be revised and updated.
- The generic differences between mean pressure control and stroke control will be determined.
- The control system for response, for the logic at operational extremes, and for fail-safe features will be investigated.

The initial work on the control system operation, fail-safe design, and response uncovered basic problems associated with the current design layout. Additional design work is required in order to proceed further in this area.

Efforts on the optimization study continued throughout the quarter. In October, attention was paid to the criteria and the overall modelling in order to observe the essential parameter influences on the resulting fuel economy. For instance, changes in the cold start penalty and engine weight followed the changes in the independent parameters as closely as possible. The main portion of the criteria model consisted of:

- Stress criteria for hot parts (creep and buckling);
- Auxiliary characteristics criteria;
- Weight criteria (also for cold parts);
- Cold start penalty;
- Bearing dimensioning criteria (affects friction);
- Engine geometry criteria.

All these criteria were programmed in analytical form, so that most changes could be reduced to a few changed input values for the computer code. A rough mileage number was also calculated, which made it possible to let the computer maximize the mileage while directly taking the change in weight and cold start penalty into account. However, such optimizations did not replace real CVS simulations. Combinations of pressure, speed, and temperature, which were agreed to be examined by USSw, are found in Figure 1.0-1. However, with iron-based heater material, the combination of 15 and 17.5 MPa and 870°C were not possible due to thermal stress. The middle combination (15 MPa, 4000 rpm and 820°C) will be used as a reference point. To make the number of cases reasonable, it was suggested that all nine (p, n) combinations be checked at 820°C and then the reference combination of (p, n) be checked for 770°C. This totalled ten cases. Values between the discrete values were also checked. The computer model allowed for a parameter value combination within reasonable limits, as the criteria are described as analytical expressions.

In November, the engine system model for the optimization computer code was completed. The model was extended to cover the external heat system efficiency, which depended on the chosen heater temperature. The mathematical model of the Stirling engine geometry was refined to represent the real hardware geometry in more detail than before. This was necessary due to the early optimization runs which revealed that the heater did not couple with the cylinder/regenerator geometry in a satisfactory manner. The mathematics of the involute heater tubes was included in the computer code so that criteria for non-interference and covering angle would be taken into account. In summary, the following capabilities are now included in the computer code for optimization:

- Stress criteria for hot parts (creep and buckling);
- Wall thicknesses, etc. can be automatically calculated to correspond to the current combination of diameter, pressure, and temperature;
- Auxiliaries characteristics criteria, where the power consumption curve in some cases is dependent on the engine speed range;
- Weight criteria:
  - The weight of the hot parts can be calculated using the dimensions from the stress criteria (W_{hot});
  - The weight of some of the cold parts can be scaled with swept volume and maximum mean pressure as:

$$W = W_o \cdot \frac{V}{V_o} \cdot \left(\frac{P}{P_o}\right)^{2/3}$$

  - This relation follows from a simple constant stress hypothesis, and will probably have a slope value in the right order of magnitude.
  - The total engine weight is:

$$W_{tot} = W_{const} + W_{cold} + W_{hot}$$

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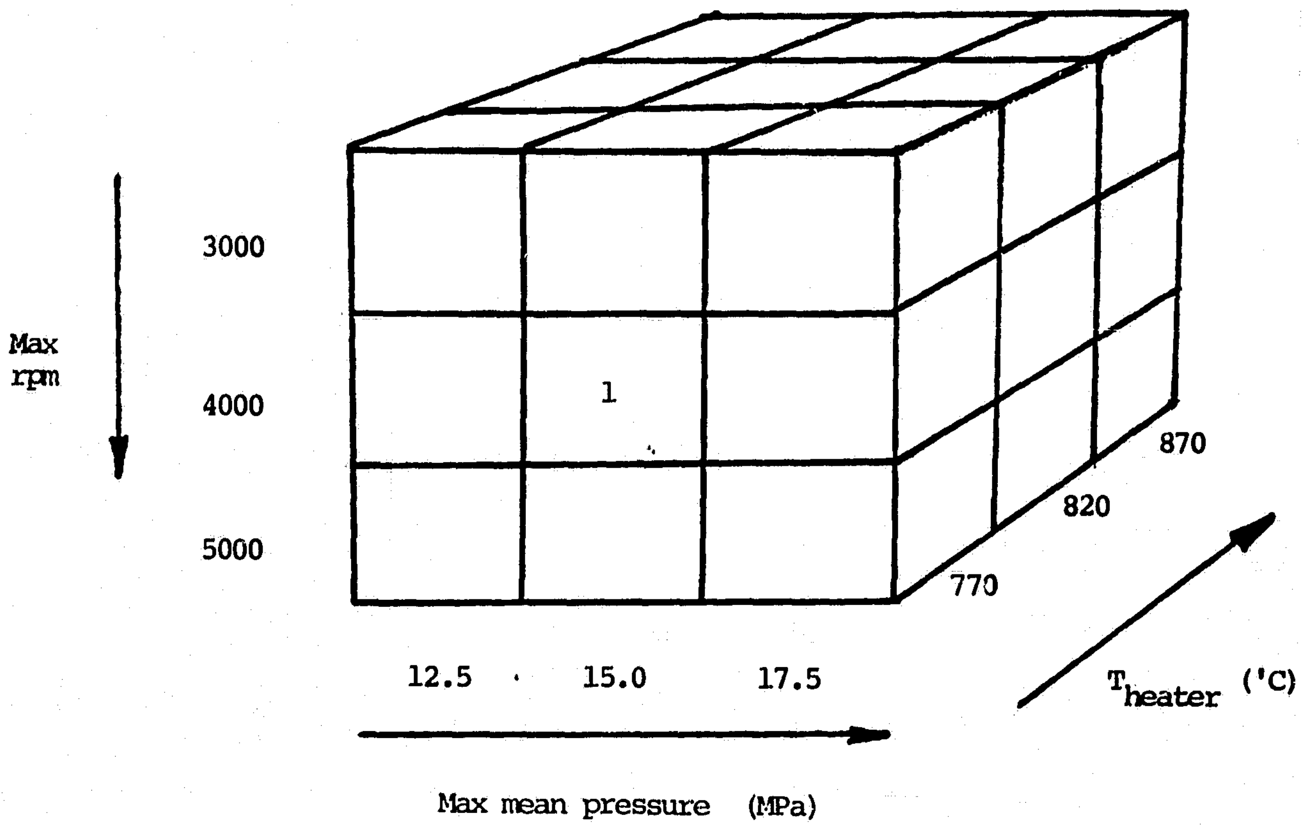

Figure 1.0-1 Pressure, Speed, and Temperature Combinations to Be Analyzed

- Cold start penalty, which can be calculated at the stored amount of heat multiplied by a factor (f) in the interval: $0.43 < f \leq 1$,

- The code includes an equation for heat storage calculations for time dependent variations in $C_p(t)$ for all engine masses separately. The following expression is used, assuming uniform mass distribution for each engine mass ($dm/dx = 0$):

$$E = \frac{m}{T_2 - T_1} \int_{T_1}^{T_2} \int_{t_o}^{T} Cp(t)\,dt\,dT$$

T = time, and t = temperature.

Bearing Dimensioning Criterion

Previously, the bearings were dimensioned for constant surface load with respect to maximum mean pressure. This criterion gave a conservative friction tion loss response to an increase in mean pressure, as the cycle pressure ratio (p_{max}/p_{min}) decreased with decreasing mean pressure level. This,'in turn, (as was shown before) resulted in an increased mileage with decreasing mean pressure. In the new model, the dimensioning criteria is for a constant oil film thickness at the operating point (for instance 1000 rpm, full pressure). Here, the pressure ratio was taken into account, and therefore, the relationship of friction versus pressure was flattened out.

Engine Geometry Criteria

The cylinder distance and the regenerator position was calculated from a constant distance between the outer wall surfaces of the canisters.

The length of the cold connecting duct was coupled to the regenerator/ cylinder distance.

The cooler tube length was coupled to the stroke.

The complete involute heater mathematics was included. The tube bend extension was also included in the non-interference criterion. The relative position of the hot manifolds was free to be optimized within certain limits around the canister centers. The height of the heater was calculated and included the upper bend radius. The fin section height was coupled to the regenerator length and the canister top thickness. The volume of the hot connecting ducts (manifolds) was coupled to the heater cross section and pitch radii.

- The independent variables used in the optimization study are shown in Table 1.0-1.

- The mileage was initially calculated from a simplified formula which could be used as an optimization function.

In December, preliminary results of the optimization of the RESD were received from USSw. The results show no significant changes (< 2.0 mpg) in combined mileage over the ranges of pressure, temperature, and rotational speeds explored. Information from USSw clarified some questions

X(1) = Swept Volume

X(2) = Bore/Stroke Ratio

X(3) = Cold Connecting Duct Cross Sectional Area

X(4) = Regenerator Matrix Diameter

X(5) = Regenerator Length/Diameter

X(6) = Regenerator Filling Factor

X(7) = Cold Connecting Duct Vertical Length

X(8) = Cooler Tube Bundle Diameter/Regenerator Diameter

X(9) = XR+1.; XR = Relative Regenerator Manifold Position

Table 1.0-1 Independent Variable Used in the RESD Optimization Study concerning the design adapted in the optimization study. A full engineering evaluation with MTI and NASA is scheduled for January 12-15, 1981.

A Request for Proposal for the RESD Manufacturing Cost Study was submitted to several outside vendors. This study is aimed at developing the cost for the RESD.

The design of the RESD in the X-Car chassis was concluded and final clarifications were made. A report was drafted which defined the design requirements, objectives, and conclusions of the installation. A new task was initiated for the Reference Engine, which incorporated a speed increase between the crankshafts and the drive shaft. To retain full balance, a separate balancer shaft was required.

A meeting with MTI, USSw, and Ricardo was held in Malmo in November to initiate the step-up gear study for the Reference Engine. Different solutions were discussed for the location of the extra balance shaft; it was decided that two different alternatives should be further evaluated by Ricardo. In the first alternative, the extra shaft was located outside one of the crankshafts. This implied a redesign of the crankcase and a rearrangement of the burner blower location. In the second alternative, the extra balance shaft was located concentric to and inside the main shaft. The shaft would be driven by a gear in the front end. The balance weight in the rear end would be located between the main gear and torque converter. The second alternative was judged to have the highest risk, but at the same time, its design would have the highest potential for low weight.

In November, a computer program was generated to simulate a Stirling engine mean pressure control system similar to the P-40/Mod I configuration. This program was not validated at that time. The program concept was geared toward flexibility and modularity, with the mean pressure control program used as a subprogram.

Additional refinements were incorporated into the design of the RESD. The cylinder liner and crosshead guide were redesigned to form an integral part for better alignment between the crosshead and the piston. This redesign resulted in less risk for side forces and wear of the PL seal, and better control of the piston dome gap. The seal housing was threaded into the cylinder and crosshead liner. The integral cylinder/crosshead liner was brazed to the duct plate, which eliminated the two gas O-rings.

The duct plate was also redesigned and now consists of a plate with a depth of about two inches. The crosshead guide is no longer an integral part of the duct plate. The check valves for minimum and maximum cycle gas pressure and supply were repositioned to permit easier access at the time of installation and replacement. The check valves are all of the Mod I-type.

The old gas line connections from the power control valve and between the duct plates were redesigned. This resulted in an improved and simpler ducting design which will make assembly easier. Drawings of the new design are in progress.

Highly stressed components, which influence the Stirling cycle, were analyzed, considering wall thickness and other typical dimensions as functions of the maximum mean pressure and the heater tube temperature. The analysis was performed for the following components: the heater tubes, the heater head housings including the manifolds, the gas coolers, the piston domes, the piston rods, and the crankshafts.

A simple, removable, and cleanable preheater which consisted of a corrugated tube, was added to the outer air duct. With this arrangement, the coldest end of the preheater will move from the main preheater matrix to this secondary preheater. The cold end plate temperature of the original preheater matrix was raised to give a safe margin between the matrix plate temperature and the acid dew point (at the coldest ambient temperature). This redesign increased the outer diameter of the external heat system by not more than 10 mm; however the weight and cold start penalty were not affected.

The combustion system was also reworked. The aim was to move the fuel injection system closer to the center of the combustor and to help the vaporization process by recirculating combustion gases rather than mixing of preheated air and combustion gases. This redesign may prevent the possible flash-back at low loads.

A structural analysis of the Z-crank hot system is almost completed. Remaining work includes the layout of the new cylinder head geometry which incorporates the results of the analysis, and the correction of the conductivity estimates to match the new design.

The calculated loads for the stroke control actuator mechanism of the Z-stroke engine vary from minus 1525 lb_f at idle to plus 3265 lb_f at full power. The load magnitude is manageable but the direction is opposite to that which is desirable, particularly at idle. The preferred trend would be to drive the mechanism to shorter strokes as engine load/speed decreases. Ways of accomplishing this are being investigated.

MAJOR TASK 2 - COMPONENT AND SUBSYSTEM DEVELOPMENT

Task 2.1 - Combustion Technology Development

Design and Analysis

The detailed design of the surface combustion section for the Exploratory Test Rig was completed in October. The report on the surface combustion concept and now it may be applied to the RESD was completed and approved in December.

In November, Parker-Hannifin Co. was contacted in connection with the alternative fuel nozzle inquiry. A drawing which gives the overall dimensions of the fuel injection nozzle area was sent to them for a technical feasibility evaluation.

Suggestions for a revision to the AMG Spirit Emissions Test Plan were made to include measurements of preheater air discharge temperature, air flow, and atomizing air pressure during the steady state portion and air flow and fuel flow during CVS testing.

Fluid Dynamics Rig

In October, the conceptual design of the Air Flow Test Rig was initiated. This test rig will be used for flow visualization, pressure drop measurements, and calibration of flow meters. In November, a review of the water table design was completed. The water table manufacturing review was completed and minor drawing changes were started in December.

Exploratory Test Rig

At the beginning of the quarter, the manufacturing review of the base was completed and a Request for Quotation (RFQ) was issued for the refractory surface combustion test section and for the base unit.

Combustor Development Test Rig

In October, a contract was awarded for the installation of the inlet air and exhaust systems. By November, all of the fabricated parts for the rig were received and assembled. By the end of this quarter, the fuel, cooling water, and atomized air systems were completed. The inlet blower, safety relief valve, bypass valve, and actuator were installed in the rig.

Combustor Endurance Test Rig

In October, the design of the heat extraction system was reviewed and updated to conform to the RESD heater requirements. An RFQ was revised for the heat exchangers. In November, a revised layout of the rig and its subsystems was completed and the major components were identified. The design of the simulated heater head manifold assemblies was completed and a compressor for the cooling loop was selected. Quotations for the heat exchangers for the cooling loop were received. By December, quotations were reviewed for the heat exchangers and control valves. Due to a space problem (the heat exchangers were too big), the heat exchanger quotations had to be revised. Specifications for the remaining components (compressor, pressure vessel, hand valves, check valves) and the instrumentation was started.

Task 2.2 - Heat Exchanger Development

In October, the literature survey of regenerator continued. A test was run in the partially assssembled Pressure Loss Test Rig to compare a dirty P-40 regenerator and one cleaned with acid; no major differences were found. In November, the order for the sintered test section was placed with Facit Enterprises; the delivery of the test section was expected by the end of December. In December, problems with the data acquisition system prevented the regenerator matrix heat transfer tests from being run. A more reliable replacement blower for the Regenerator Heat Transfer Rig arrived and was installed. The P Rig hardware was assembled; utilities and instrumentation will be completed during the next quarter.

Task 2.3 - Materials Development

In October, the second quantity of XF-818 (633 lbs) was shipped to USSw. In November, Inconel 625 tubing was cut to length and heat treated at Klock Inc. (East Hartford, Conn.); the material was then shipped to USSw. Later in this quarter, an order was placed with Superior Tube for the redrawing of the CG-27 tubing received from NASA in November. The scheduled delivery date for this redrawn tubing was January 25. Superior Tube will anneal the material and cut it to proper length for use in the manufacture of heater quadrants. Specimens of casting alloy XF-818 were also machined for tensile testing. Specimens of HS-31 for fatigue testing were cast by Bulten-Kanthal and will be shipped to MTI during the next quarter.

Brazing experiments on Inconel 625 and CG-27 are still in progress. Fixtures which simulate the brazing of tubes to manifold holes, and tubes to fins, were sent to Wall-Colmony. The brazing of these fixtures is scheduled to take place during mid-January.

The failure analysis of the quadrant from the High Temperature P-40 engine received from USSw continued, using Scanning Electron Microscopy (SEM) and Transmission Electron Microscopy (TEM) to examine the crack in the cylinder head manifold.

A failure analysis on the failed Concord engine heater quadrant was completed in December and will be reviewed during the next quarter.

Calculations of the hydrogen permeation loss rate were made for the P-40 engine at temperatures in the range 520°C to 820°C; the data were then compared to the P-40 engine loss rates measured by USSw. The following is a list of calculated values of hydrogen loss rate and the preliminary measurements of USSw as a function of temperature:

Nominal Heater Temperature (°C)	Loss Rates Measured at USSw	Calculated Permeation Loss Rates
520	2.6 Nl/hr	0.747 Nl/hr
670	3.3 Nl/hr	2.95 Nl/hr
820	8.0 Nl/hr	8.02 Nl/hr

The failure report from USSw regarding the failed Mod I piston domes was reviewed in December. Alternative weld configurations to eliminate the failure-prone notch were evaluated. Recommendations for a solution will be made at the next Materials Task Force Meeting, which is scheduled for February 12, 1981.

Task 2.4 - Mechanical Engineering - Seals Design and Analysis

In October, an assessment was made of the Mod I PL seal friction curves utilizing experimental data obtained by USSw on P-40 PL seals (12 mm diameter). The calculated power loss for 5 Mod I PL Seals (four in the engine and one in the hydrogen compressor) is shown in Figure 2.4-1. The initial review at MTI indicated that USSw's results may be 15% low. Further analysis will be performed as data becomes available.

An analysis of piston ring leakage and operation as it relates to test rig conditions, was performed. The analysis indicated that the ambient volumetric leakage rate in the test rig may be large and yet still be compatible with an "acceptable" leakage rate in the engine. This report may be found in Appendix A of this report. An analysis was also completed of the proposed duty cycle; the results are in Appendix B of this report.

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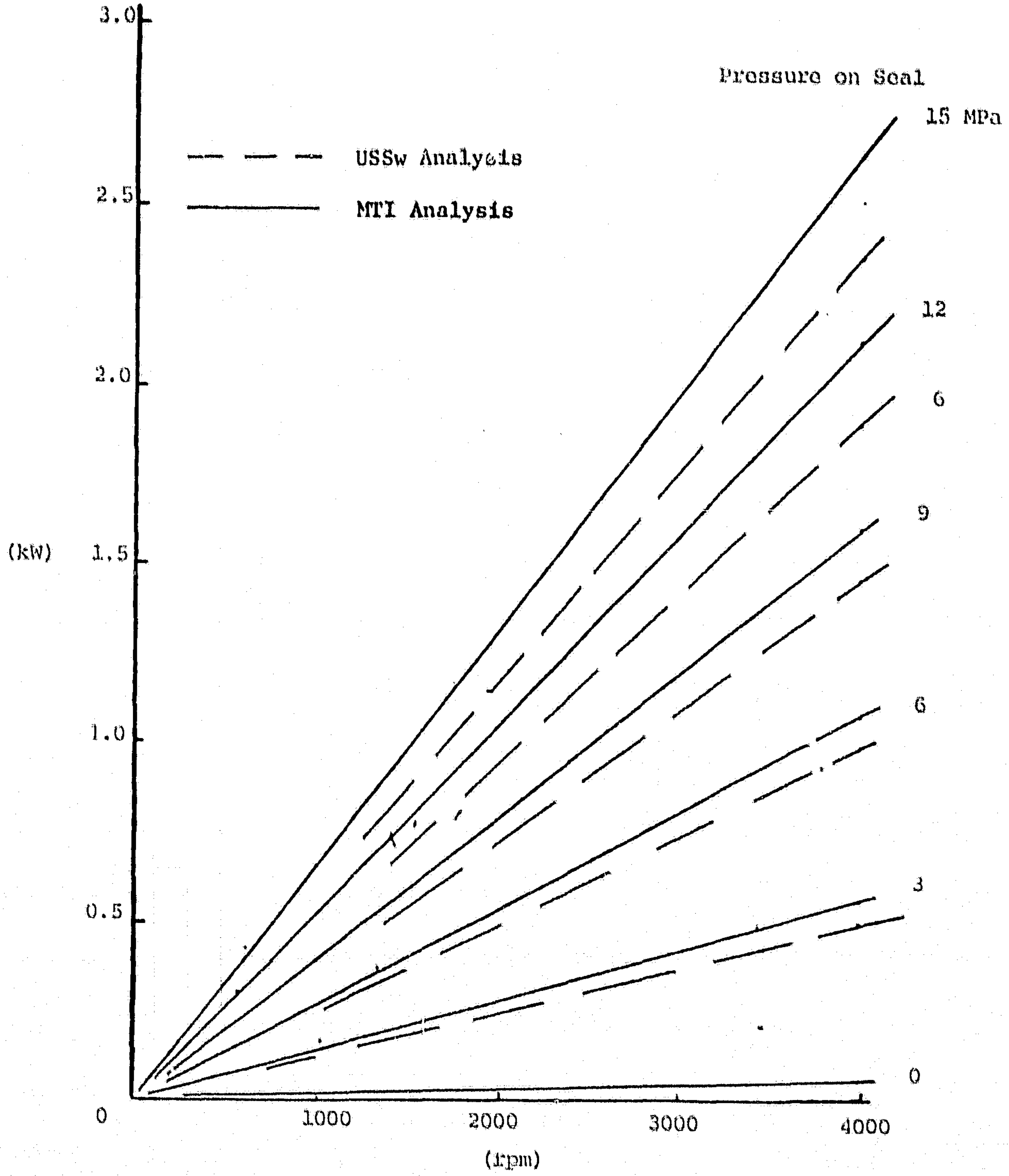

Figure 2.4-1 Power Loss for 5 Mod I PL Seals

Project Engineering

Roy Howarth of MTI presented a paper on Materials Screening Tests at the November, 1981 Contractors Coordinator Meeting in Dearborn, Michigan. Copies of this paper may be obtained from MTI [11].

Materials Screening Tests

In October, tests were carried out to investigate the effects of reciprocating speed on wear rate. The materials tested were: Dixon TFE-GL-HL-800-2, Dixon TFE-GF-H1-800-2, and Rulon E. Testing was carried out with a constant load of 0.69 MPa and at speeds of 300, 600, 1200, and 1800 rev/min. At 1800 rev/min, a flexible coupling and a connecting rod bearing failed and the test was terminated after completing approximately one third of the planned number of cycles (24,000).

In November, the three previously mentioned materials continued to be tested. The duration of the tests was varied so that the number of cycles at each speed was the same (1.44×10^6). For all three materials, the wear rate in terms of wear/cycle (within the limits of measurement) was independent of speed. Tests at 1800 rpm were attempted but the rig was over-stressed and testing was halted after completing 4.32×10^5 cycles. At this reduced number of cycles, the wear rate was the same as the rate at lower speeds. These materials were also tested to establish the effect of load on wear rate. These tests were carried out at 1200 rpm and with contact pressures of 0.68, 1.02, 1.36, and 2.04 MPa. The Dixon TFE-GL-HL-800-2 and Dixon TFE-GF-HL-800-2 materials both gave essentially constant wear rates, which were only slightly higher than these at 0.68 MPa. With Rulon E, the wear rate increased progressively with contact pressure. At 2.04 MPa, the wear rate was approximately ten times greater than at 0.68 MPa.

100 hour wear tests per material were completed on two new materials: Vespel SP211 and Koppers K30W35 (copper fibers normal to the plane of sliding). These basic tests were run at 1200 rpm and at a contact pressure of 0.68 MPa. Repeated tests were also completed on a Crane material, consisting of PTFE with 55% bronze powder and 5% MoS_2 filler. The test data showed that all three materials are in the candidate wear range.

Repeated wear tests on Koppers K30W35 (fibers parallel to the plane of sliding) continued to give wear rates in the candidate range, but not as low as the samples with the fibers normal to the plane of sliding. Repeated tests of two samples of Crane material with bronze powder/MoS_2 filler gave inconsistent results; one sample gave a very low wear rate and the other a very high wear rate. Additional tests with the Crane material again gave inconsistent results (both high and low wear rates).

In November, the Friction Test Rig was rearranged to allow the specimen to be run on the face of the disc. Preliminary tests have shown that the new arrangement gave accurate, reproducible data. Nitralloy discs were ordered in December for test purposes and should be delivered in January.

Exploratory Tests

The Piston Ring Test Head was assembled in November and was installed on the Exploratory Test Rig. Supplies and instrumentation connections were made and shakedown testing was initiated. A problem developed in the drive motor control and the unit had to be returned to the manufacturer for repair. Another controller was temporarily hooked into the system to allow shakedown testing to continue. In December, the problems with the drive motor/control system were resolved and shakedown testing continued.

Some difficulties were experienced in maintaining a uniform, constant temperature in the test head. Modifications improved the temperature distribution, but it was still a problem to set and maintain a specific temperature. This was mainly due to the low temperature of the cooling water and the low flow rates required. Further modifications are being considered to improve the temperature control.

Minor modifications to the PL seal test head drawings are being carried out prior to procurement.

Test Facility Development

In October, the main supply pump in the Seals Test Cell was installed on the platform and the scavenge pump was installed in the test cell; work is underway to complete the piping. While reviewing safety features in the test cell, it was discovered that the air circulation system in the test cell will probably not prevent a build up of hydrogen in the cell. To overcome this potential problem, a separate extractor fan will be installed and connected to one of the existing exhaust ducts in the roof.

Task 2.6 - Controls Development Systems Analysis

Systems Analysis

Work continued in October on generating a global systems analysis and interrelating all control systems. A first transfer function representation, including thermal time constants in the external heat system, was prepared. In November, P-40 tests were specified to give data necessary to qualify the Mean Pressure Control System Power Response Code. Air/fuel system transfer functions applicable to Mod I/Mod II engines will be inferred from the P-40 tests. A preferred test procedure to determine P-40 hydrogen permeation and total loss rates was also defined using a mixture of hydrogen and helium.

Low Cost Transducers

The fabrication of test fixtures for the Pressure and Position Transducer Test Rig was initiated in November. The transducers which will be evaluated were procured.

Combustion Control

In October, work began on the design of the air and fuel flow sensor test fixtures. These fixtures will be used to characterize the steady state and dynamic performance of air and fuel flow sensors, which will be applied to the Mod II air/fuel control. Testing of the Bosch K-Jetronic systems may also be possible. In November, the design of the Air and Fuel Flow Calibration Test Rig was completed and all parts for the Fuel Flow Calibration Test Rig were ordered. By December, the construction of the Air Flow Calibration Test Rig was completed and shake-down tests were conducted. As a result of the shake-down testing, a new higher capacity blower was purchased; the Datametrics airflow sensor was sent out for NBS calibration; a set of orifice plates and a mounting fixture was also purchased.

In October, Chrysler agreed to supply MTI with one of their vortex-shedding air flow sensors for evaluation. MTI will test this device along with the J-TEC unit. The Chrysler device can only be operated over a limited portion of the desired Stirling engine air flow range. Chrysler's design is currently being applied to the 8-cylinder Imperial, which has higher low and high end flow detection requirements than the Mod I. In November, the Chrysler Air Flow Sensor was received and preparations were made to test it on the Air Flow Calibration Test Rig.

Eaton Corporation showed an interest in working with MTI to develop an air/fuel control system. A trip is planned to Eaton in January to discuss their possible involvement.

The design of the air/fuel control electronics was started this quarter and will continue at a low level of effort until the characterization of the air and fuel metering devices is completed.

In December, the operator panel and memory-mapped digital input and output hardware for the generalized controller for the air/fuel control system was completed. Debugging will occur next month.

Electrical Actuator

The assembly of the Electrical Actuator and Test Rig continued as parts were received. Analysis was performed on the valve/actuator system so that a suitable electronic control could be designed and purchased.

Task 2.9 - USSw Component Development

Subtask 2.9.2 - ASE Mod I Engine

Heat Generating System

In October, the CVS Opel tests of the CGR-combustion system were completed. Typical emissions over the CVS-cycle (1975 procedure) were:

HC	1.0 g/mile
CO	3.0 g/mile
NO_x	0.6 g/mile

The main purpose of these tests was to verify that the CGR-principle could be used at transient conditions. Some uncertainties of the dynamics in the CGR-circuit have now been eliminated. Transients do not affect the CGR-system in a significant way. No_x emissions corresponded very well to measured values from engine dyno tests and combustor rig tests, while CO and HC emissions were higher due to the starts. It was also noticed that the atomizer was overheated at idle speed, which gave additional CO emissions. It must be emphasized that the P-40 CGR combustor is quite different from the Mod I combustor and therefore conclusions concerning Mod I emissions call for further development of the start sequence.

Two Burner Test Rigs were delivered to USSw in November. Emissions testing on Rig No. 1 was started and 7 hours of operation were accumulated. The tests were temporarily stopped due to a major failure (Failure Notice #U-48) of the preheater. The analysis of the failure showed evidence of a fire in the preheater. The heater head was blocked by soot due to non-stoichiometric combustion, and the pressure drop across the heater reached a level where the CGR circuit ceased to function. This caused part of the preheated air to escape through the combustion gas side of the preheater without passing through the combustor. The damaged preheater was replaced and testing started again in early December. So far, emission results of Rig No. 1 have been confusing and very different from data obtained in the AP-80 scale rig. The general feeling is that the flame temperature was too low, which resulted in very low NO_x, high CO, and high HC emissions. Soot was also noticed at some loads. Possible explanations were:

- Two much CGR;
- Two low an inlet temperature to the combustor;
- Poor temperature distribution of the preheated air;
- The influence of the geometrical differences between the AP-80 scale combustor and the Mod I combustor (such as the conical shape).

The analysis of this problem has the highest priority.

The engine cold start tests in the Low Temperature Test Rig continued in December. The first part of the test program was adapted to room temperature start tests in order to define a suitable start sequence.

Power Control

By the end of October, the Mod I hydrogen compressor operated for 930 hours for a total of 7765 hours; the total accumulated number of cycles was 1,059,316. Testing of this version of the Mod I compressor was completed and the rig will now be moved to the new component laboratory; testing of the improved Mod I with the new piston/piston rod design will begin.

The Power Control Valve Test Unit No. 2 operated for 159 hours, for a total of 4476 hours and 273,590 cycles.

The development work of the Mod I solenoid valves at Valcor Inc. (NJ) continued in October. The testing program for the first prototype has started and Valcor expects to have two complete sets of valves ready by mid-December and the balance in January, 1981.

The design and drawings of the Power Control Block No. 1, 2, and 3 were completed in October and the design of the Power Control Block No. 4 was completed in November. Early in the quarter, Moog started the design of the new electro-hydraulic type actuator, which included a mechanical feedback system. In November, Moog and USSw discussed the new design in more detail.

Burner Blower

The endurance testing of the Mod I blower was completed in November after 1000 accumulated hours. The inspection after testing showed some cracks in the belt surface, much like those found in earlier tests; but no malfunction was noticed. The cracks were probably caused by the temperature rise in the blower.

The endurance test of a new shipment of Siegling flat-belts was started in November. The flat belt was somewhat thinner than the previous ones which were tested. A number of tests with the Siegling flat belt indicate that this belt cannot be used on the Mod I blower. During these tests, belt life varied from 20 to 700 hours. The belt seemed to be running off the center of the pulleys; the reason for this will be determined. During the same test conditions, the Fredfors belt did not indicate any problems at all.

Atomizer Air Compressor

A compressor endurance test at 2500 rpm and 70 kPa output pressure was started in October. Tests were also performed with the built-in oil pump; the tests indicated that an increase of capacity might be necessary. In order to verify these circumstances, further tests have started. In November, the pump was redesigned to attain a higher pumping capacity. An inspection after 500 hours of endurance testing showed normal wear on the carbon vanes. The test will continue until 1000 hours of running time is reached.

Air Control Concept Evaluation

Testing of the new air flow transducer in October indicated a malfunction with this particular transducer, and contacts with the manufacturer were made. After this transducer was repaired, testing continued in November with good performance results.

Figures 2.9-1 and 2.9-2 are results from earlier tests with the "Fluid Inventor" air flow transducer. As can be seen, the pressure drop over the transducer approximates the specified goal of 1.5 kPa at 79 g/s of air flow. The repeatability is acceptable; with a rather simple correction curve, the accuracy is also acceptable. Testing will continue in December and a preliminary test report will be prepared in January.

Variable Belt Drive

Testing of two new solenoid valves for variator control was completed after 50,000 test cycles. One of the tested valve types will be chosen for the Mod I variator control.

Electronics

The microprocessor system was tested in October with the simulation of all inputs. Small adjustments were made to the program and the hardware. The program for monitor communication with engine electronics was

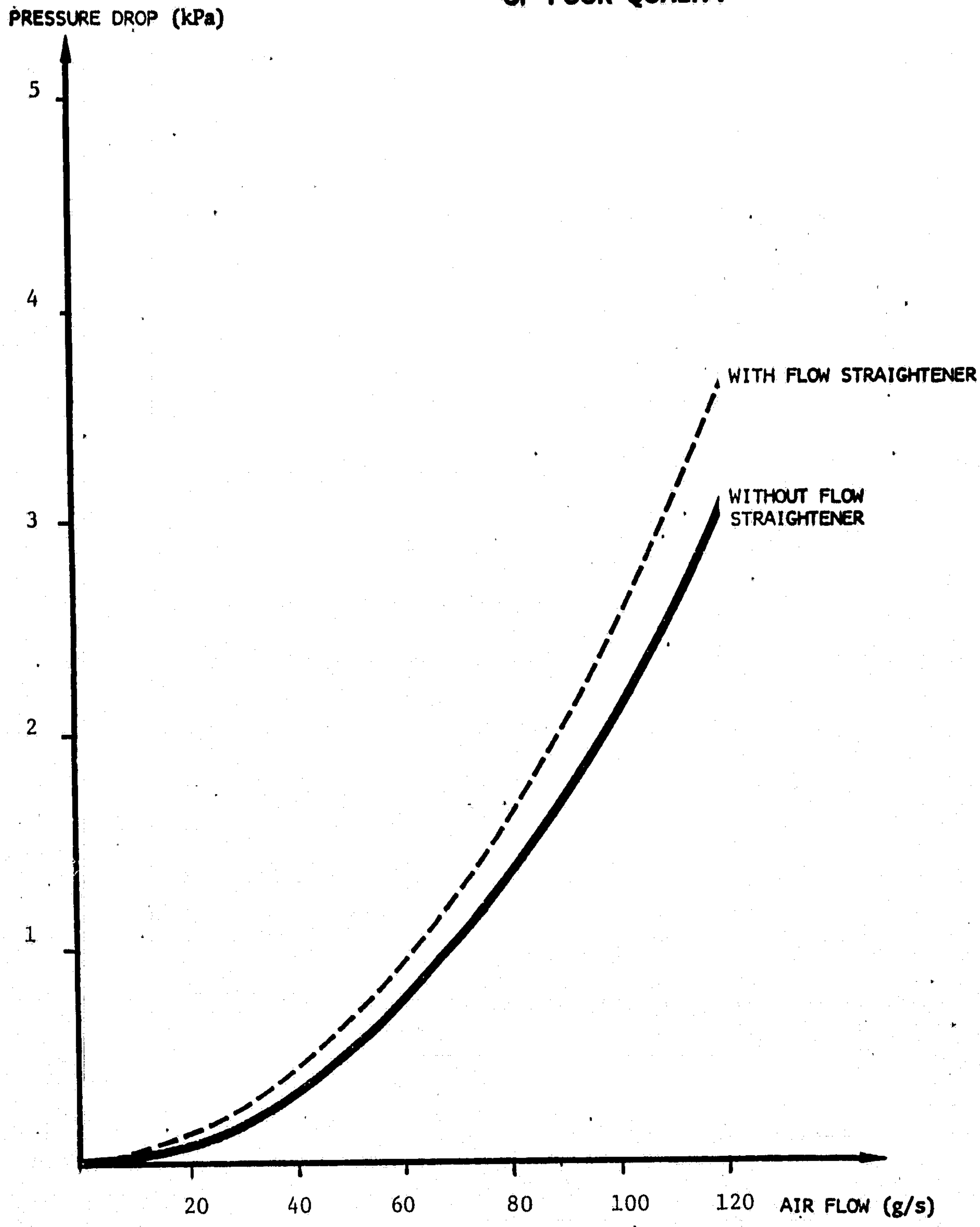

Figure 2.9-1 Test with "Fluid Inventor" Airflow Transducer

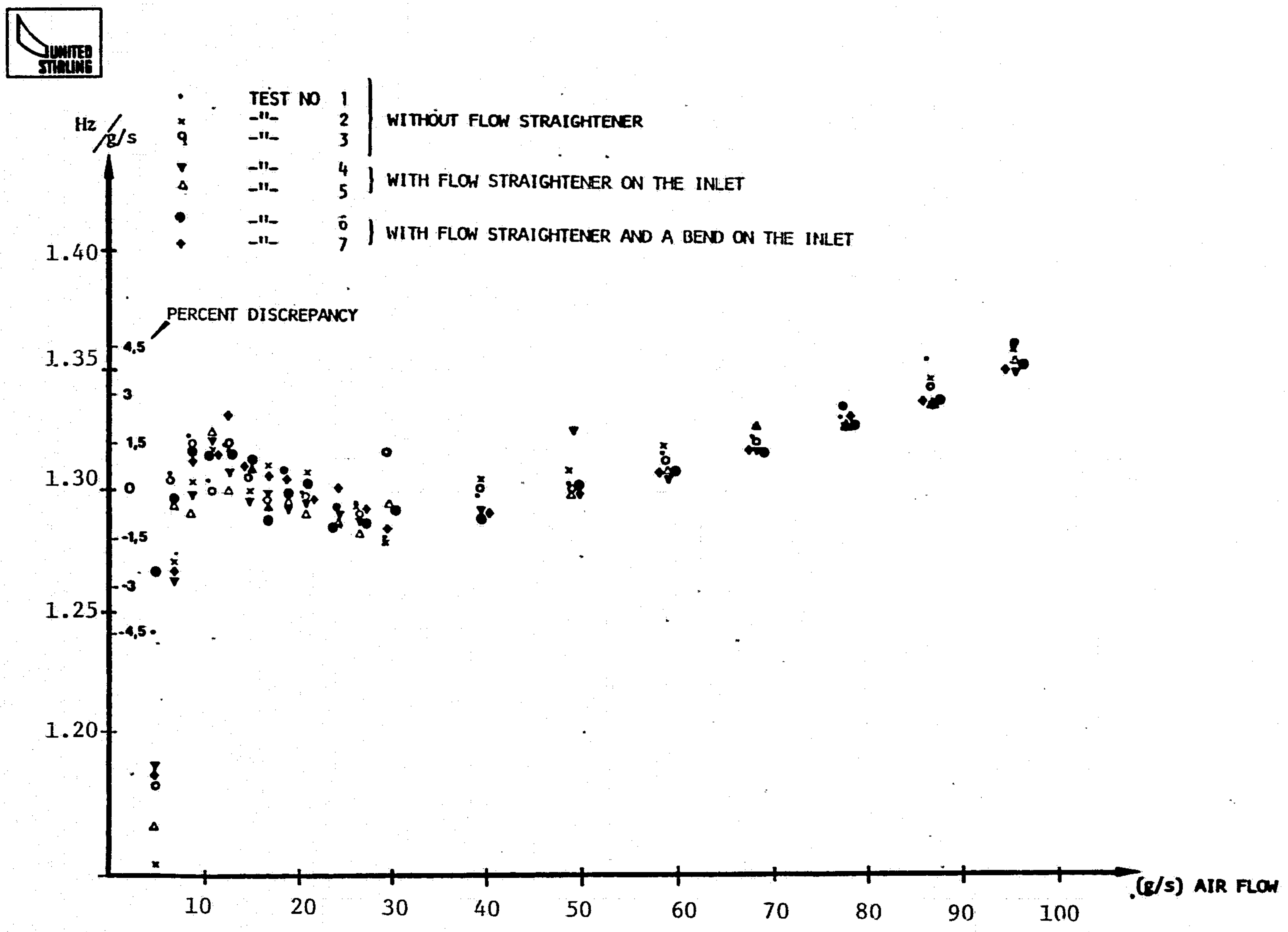

Figure 2.9-2 Test with "Fluid Inventor" Airflow Transducer

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written and all systems in the total program and hardware were checked. The logics, watch-dog, and safety systems were also tested along with a new pressure transducer from National Semiconductor. This is a low price transducer which is supposed to have excellent resolution. The transducer was then mounted in a separate housing.

The monitor capacity and how all information will be treated by the monitor was evaluated. To program the PROM's (Program Read Only Memory), a special program was written for the computer to deliver data to the PROM-programmer. Loading PROM's can now start. The calibration program, which must be completed, will be used for setting the correct Moog valve position and also for the servo valve setting. The testing of the complete microprocessor unit will continue into the quarter.

In December, the monitor program was completed. Hardware for the transducers for pressure, water, cold junctions, oil temperature, and servos (throttle and Moog) were also completed. The programming of PROM's will be completed in the next quarter. A short time will then be needed to reconstruct all cables and connectors so that the motor tests can start at the end of January, 1981.

Subtask 2.9.3 - ASE Mod II Engine

Stress Analysis of the Heater

The three dimensional photo-elastic analysis was started in October on the Mod I cylinder housing, including the manifold. The analysis was performed with models in the scale of 2:1, and cast in araldite. The manufacturing of wooden casting models also continued. Work began on a three dimensional (3-D) linear finite element analysis of the manifold region of the Mod I cylinder head. In the first approach, the model consisted of 136 20-nodes isoparametric solid elements which corresponded to 935 nodal points or approximately 2800 degrees of freedom. The finite element mesh of the model is shown in Figure 2.9-3. Both internal pressure and thermal transient loading were applied to the model, and stresses were compared to the results from the photo-elastic analysis of the cylinder head of the Mod I engine. The conclusions from the 3-D finite element analysis and the photo-elastic analysis represent an important input to the design of the Mod II engine.

To make an optimal design of the cylinder/regenerator manifolds of the Mod II engine, it was necessary to know all the loads which affect the manifolds. One of the loads came from thermal expansion of the heater, and therefore, the effect of thermal expansion of one heater quadrant was analyzed as follows:

- The geometry used was based on the Mod I design and the heater quadrant was analyzed using the finite element method. All of the heater tubes were modelled using 648 beam elements, as shown in Figure 2.9-4. The manifolds, the cylinder head, and the regenerator housing were also modelled with beam elements of different stiffnesses. The fins on the outer tube circle were simulated

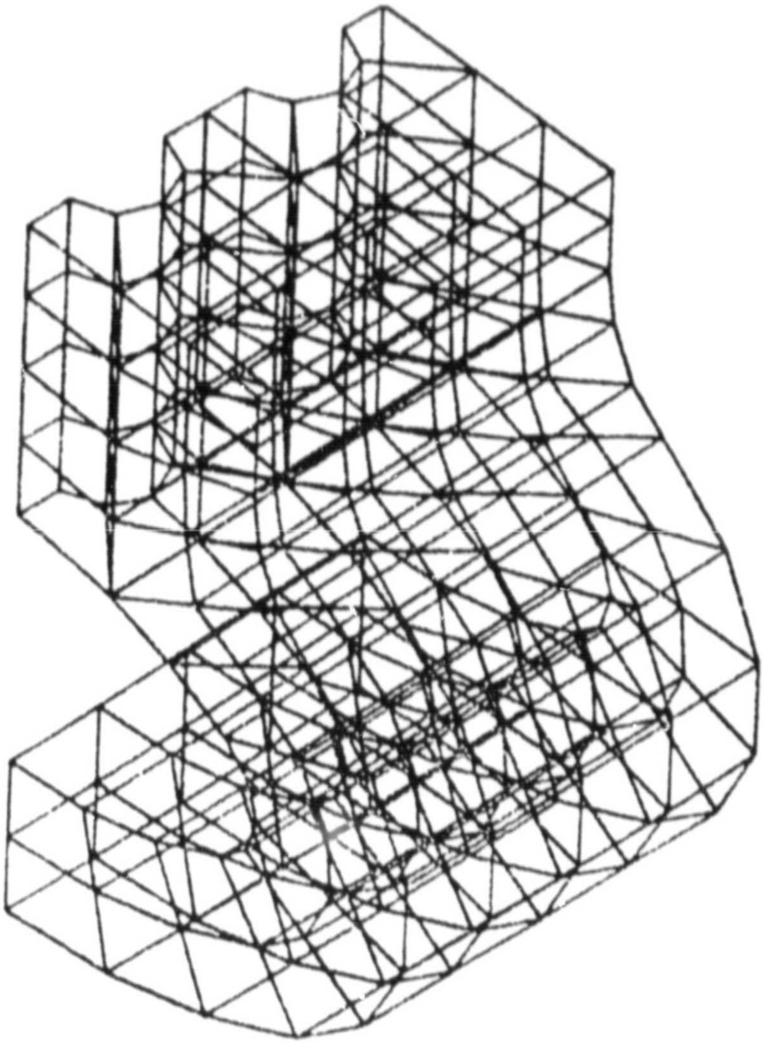

Figure 2.9-3 Finite Element Model of the Manifold Region of the Mod I Cylinder Head

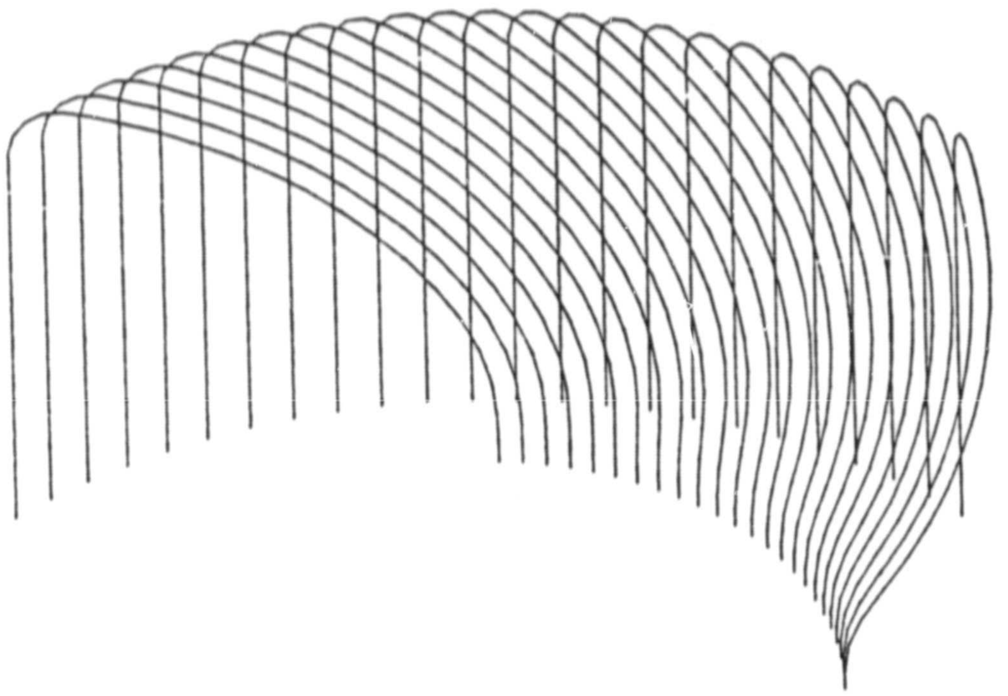

Figure 2.9-4 Heater Tubes in One Quadrant Modelled with Beam Elements

using beam elements, as shown in Figures 2.9-5 and 2.9-6; therefore, the total model consisted of 788 beam elements and 680 nodal points corresponding to 4080 degrees of freedom. In the connection between the cylinder head and the cylinder manifold, the analysis gave an effective stress of: $\sigma_e = 20\ N/mm^2$, which corresponded to a shear force of 389 N and a twisting moment of 9.7 Nm. The effective stress along various heater tubes is shown in Figure 2.9-7.

Heat Generating System

The development of the alternative atomization system progressed during November. Different prevaporized/partly premixed fuel systems were tested in the AP-80 scale rig. In December, the use of preheated air in a "conventional" atomizer was studied. This work was performed in order to find a low mass system to reduce the cold start penalty (CSP).

1-Cylinder Capseal Material Screening Test Rig

The following rods were run against a Rulon LD seal in October: WC sputter ion coating (Harwell); nitrided steel; WC sputter ion coating (Harwell); Metco 101 (94 $Al_2 O_3$ $Ti O_2$) plasma sprayed; Metco 101 (94 Al_2 O_3 $Ti O_2$) plasma sprayed; Metco 101 (94 Al_2 09_3 $Ti O_2$) plasma sprayed. Metco 136 F, Metco 136 F, Metco 111, and Metco 111 were run against a Rulon LD Seal in November and one rod (NEDOX) was run against a Rulon J seal. The first series of tests with different rod materials was run in December against a Rulon LD seal. 30 rods were run and surface hardness measurements were performed. Three different surfaces were selected for further testing against different seal materials:

1. Nitrided steel, Vickers Hardness (HV) = 1100, R_a = 0.2 - 0.4 µm.
2. Plasma spray coated Metco 505, 0.1mm thick.
3. Plasma spray coated Metco 101F, 0.1mm thick.

The first test series with nitrided rods was started in December. Two rods were run against Rulon LD as a baseline test.

1- Cylinder Cold Start Test Rig

The 1- Cylinder Cold Start Test Rig was function-tested in November and was moved into a cold chamber where it was rotated at 140 rpm. Visicorder measurements were made of pressures in the upper and lower cycle and between the piston rings. Figure 2.9-8 shows a section through the test rig.

Endurance Testing on the High Temperature P-40 Engine

The total operating time for October was 29.6 hours, totaling 5620.7 hours. 2330.7 hours were accumulated on Heater 2.12739 (No. 17 Quadrant 1, 3 and 4) of which 2160.8 hours were run with hydrogen as the working gas. The heater quadrant from Heater 2.12739 No. 14 failed. This heater was connected to Cylinder No. 2 when the original quadrant in heater No. 17 failed. Heater No. 14 was connected to the engine during 1895.9

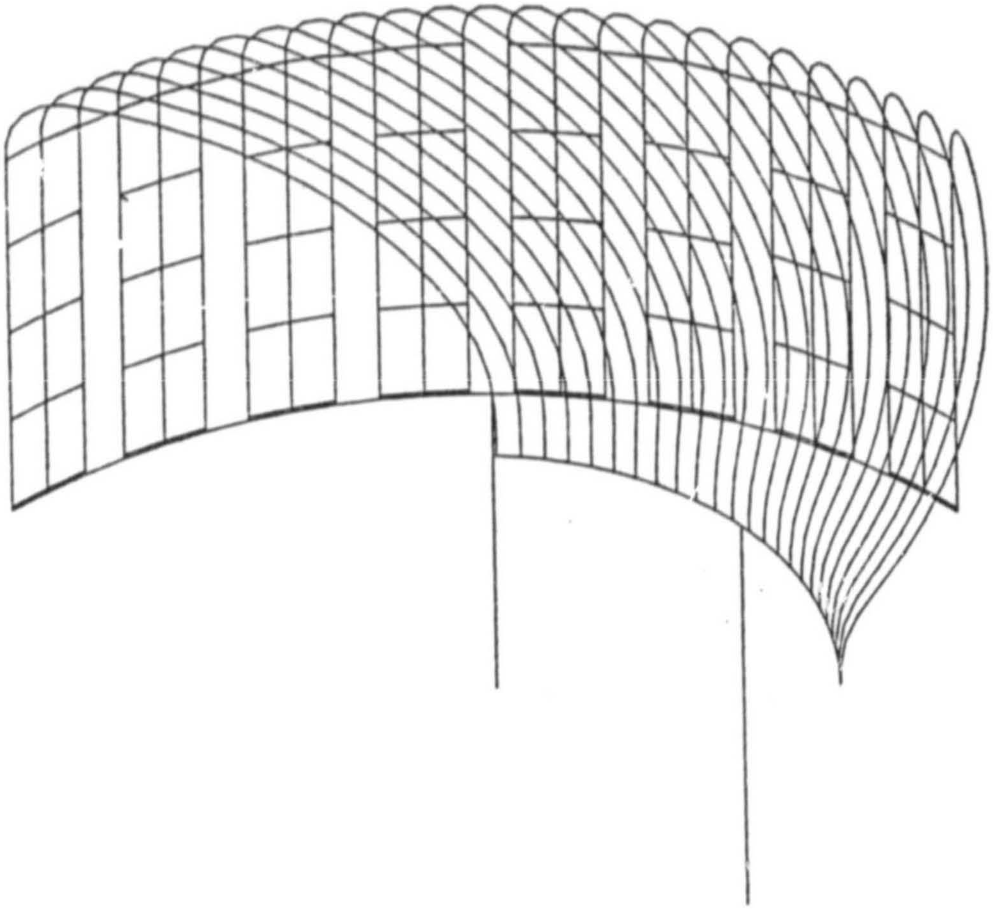

Figure 2.9-5 Finite Element Model of a Heater Quadrant

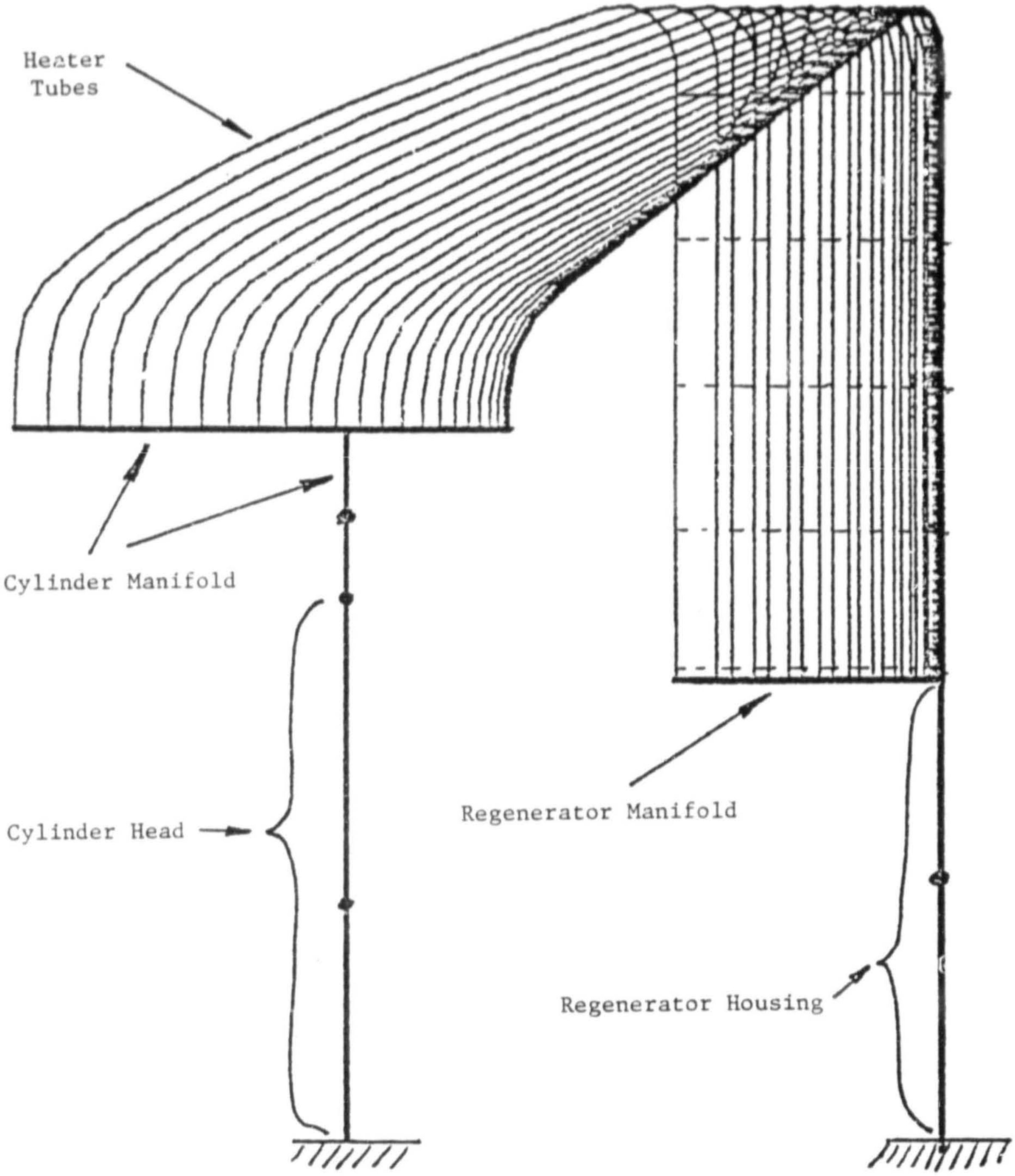

Figure 2.9-6 Finite Element Model of a Heater Quadrant

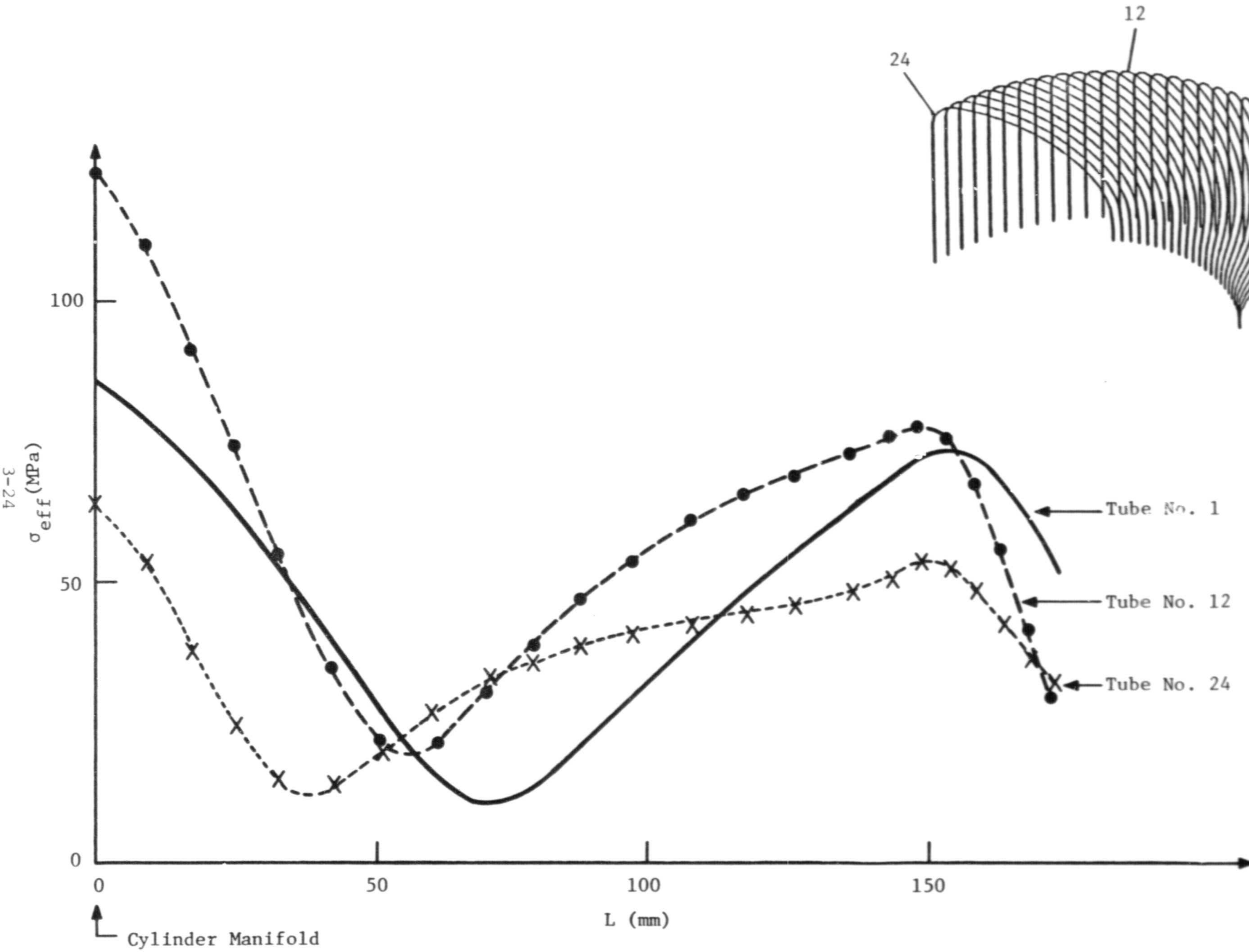

Figure 2.9-7 Effective Stress Due to a Thermal Expansion along Various Heater Tubes

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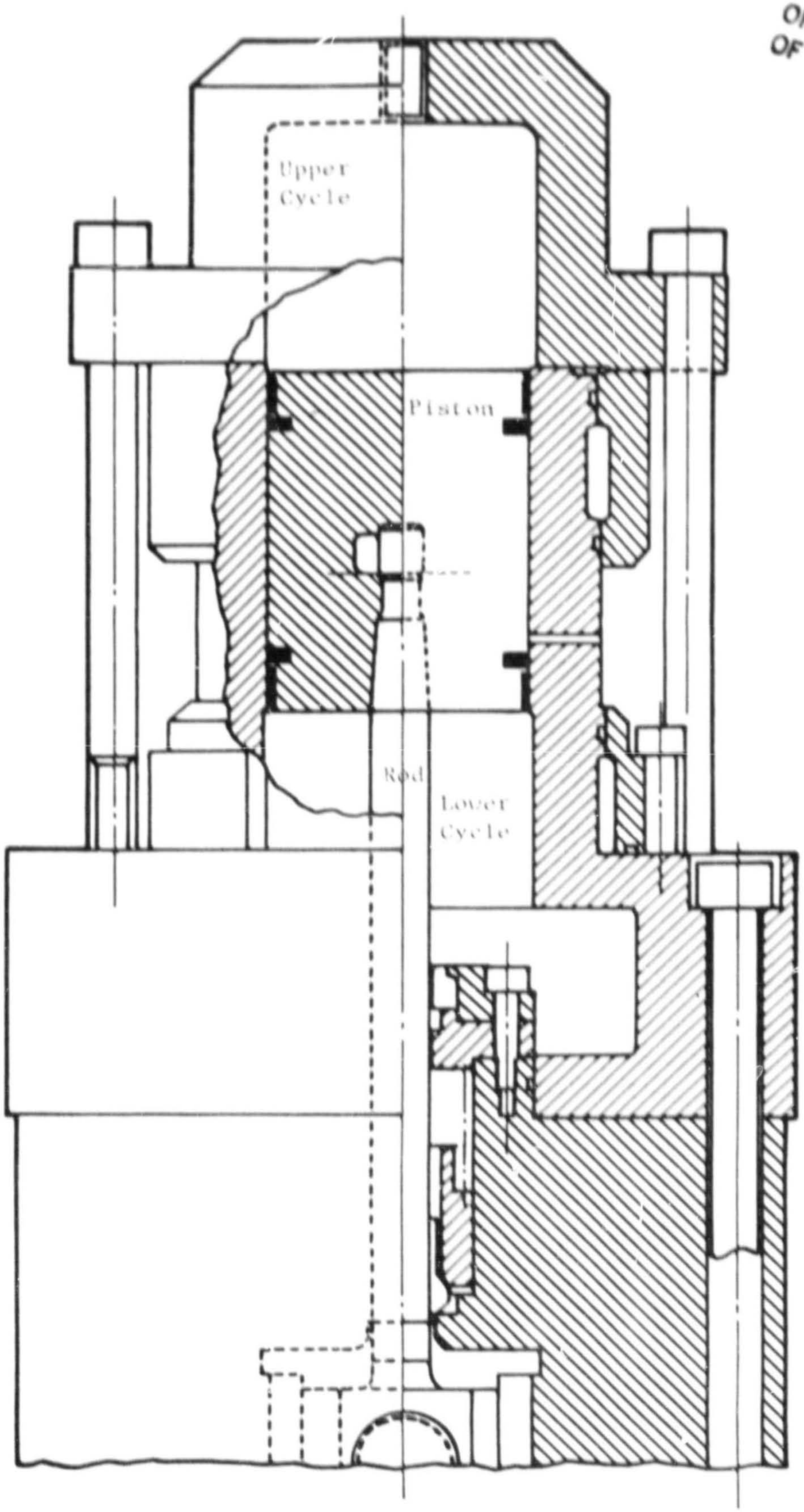

Figure 2.9-8 1-Cylinder Cold Start Test Rig

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hours of Stage 1 testing. The failure occurred after 684.9 hours, when a working gas leakage was located in the area between two tubes in the manifold center of the cylinder. The high temperature endurance tests were terminated because a replacement quadrant was not available. A spare quadrant from MTI was sent to USSw. Efforts were made to check the possibility of measuring the working gas loss as a function of the heater tube wall temperature. For measuring the working gas loss, there was no need for extra test equipment to be connected to the engine. The gas needed for recharging the storage vessels was calculated and the time between the recharging periods was checked. The heater tube temperature was measured and data were recorded during more than 100 hours at 520°, 820°, and 670°. The results are shown in Figure 2.9-9. Based on this data, the only valid conclusion was that a significant connection was noted between the loss of working gas hydrogen and the heater tube wall temperature for the engine during testing. No further data were available to show from which part or parts of the engine the increased working gas loss occurred when the heater tube temperature was elevated. A test program will be proposed to show to what extent the heater is contributing to the total working gas loss.

The material selections for the two heater heads designed for non-strategic material evaluation were finished in November and drawings for the machining of the various parts were started. The design of the longer piston domes was completed and manufacturing was started. The regenerator housings made of Haynes Stellite-31 were cast.

The test program to determine the permeation loss rate of hydrogen and the sum of all other hydrogen leaks advanced, in December, to the stage of running-in new components. The control system was modified in order to reduce external leaks. New PL seals, piston rings, and O-rings were mounted. Testing is scheduled to begin in the next quarter.

Testing of Annular Regenerator Tube Heater for P-40 Engine

All tests with the present heater design were completed during October. The proof casting of the new heater was completed in mid-November. Final casting is planned to take place in mid-December. No manufacturing problems were encountered this quarter. Nine of the ten castings passed crack detection and X-ray examinations. Surfaces will be machine-finished early in the next quarter.

P-40 No. 15 Motored Engine

The crankcases of the P-40 No.6 and No.15 were switched in November in order to fit the electric motor drives. Subsequently, the identification numbers were also changed because the identification numbers were on the crankcases. The P-40 No.15 motored drive is planned for constant speed and mean pressure tests. The test rig was assembled and wear tests of piston rings and capseals were performed. Most parts for the test rig were procured, except for the cylinder liners.

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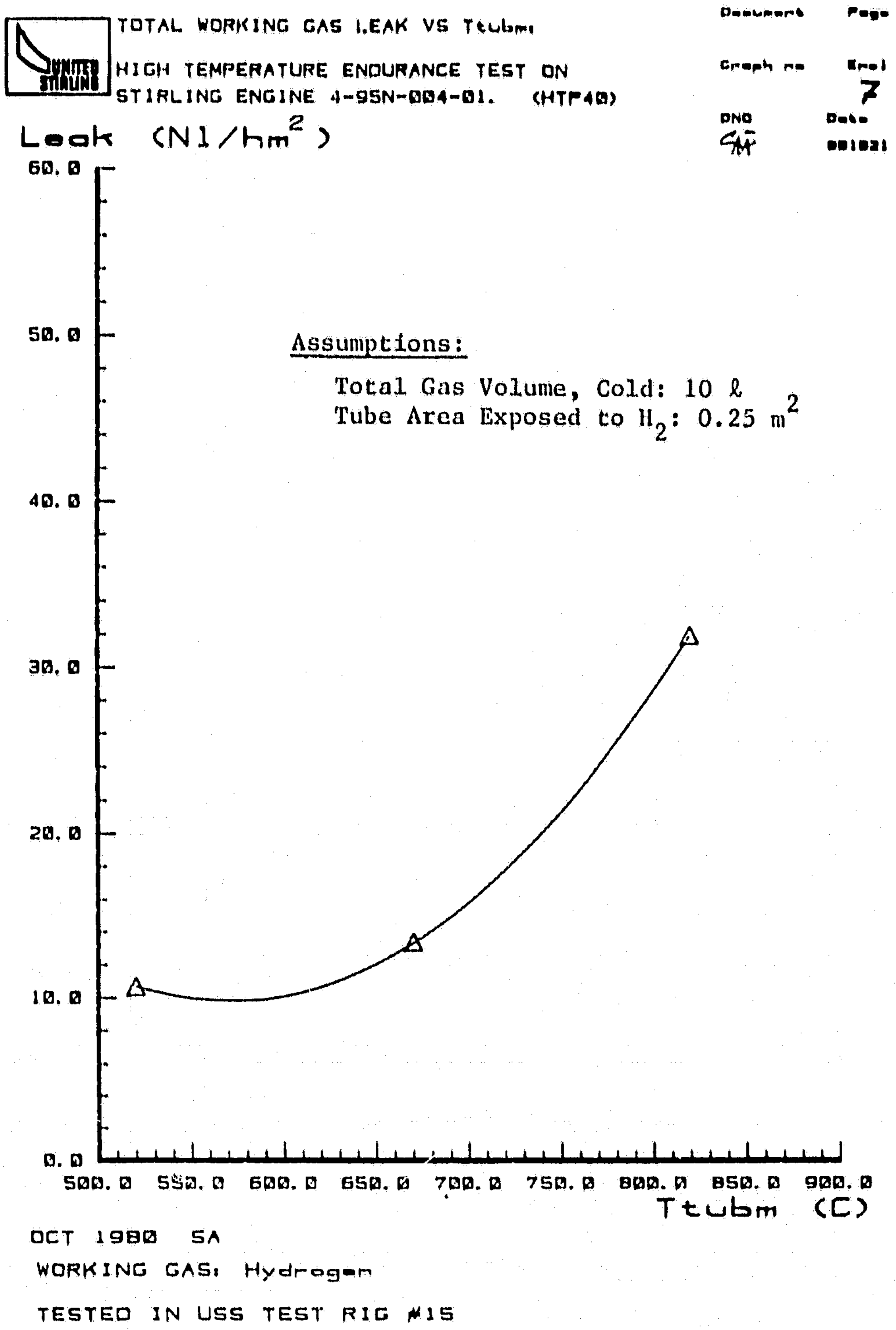

Figure 2.9-9 High-Temperature Endurance Test — Measurement of Heater Tube Temperatures

Piston Dome Development

Because of former problems in welding of domes made of high-temperature material, a dome investigation was initiated. Test rings will be manufactured of different materials, and different material combinations will then be test welded. Results are scheduled to be available in February, 1981.

Pre-Mod II Engine

A preliminary layout for the hot engine system, and also for the cold connecting duct and the water jacket was completed in December.

Air Preheater

The detailed design of the preheater was about 75% finished at the end of this quarter. A preliminary layout of the External Heat System was completed for those parts for which the dependent dimensions of the preheater matrix were set. The necessary combustion chamber design specifications were not yet available.

Heater

A precision-molded Mod I cylinder unit will be used. The dimensions of the heater cage and the surface extension of the rear tube row were settled.

Regenerator

The regenerator housing will be of a new construction, and the detailed design was 80% complete by the end of the quarter. The main dimensions of the tubes for the regenerators and gas coolers were determined.

A fatigue test was performed with a P-75 regenerator housing having an EB-welded plug (diameter about 30 mm) at the top. The pressure was 15.75 ± 5.25 MPa. The test was terminated after 8.6 million cycles due to a bolt failure. The test will continue during the next quarter after a redesign of the test equipment.

Gas Cooler Analysis

The gas cooler and the flow plate will be brazed together. A ring nut will be used to fasten the cooler and the flow plate to the regenerator housing. The assembly was analyzed using the finite element analysis. The analysis showed that the yield and fatigue requirements were fulfilled. The maximum radial stress in the brazed joint was only 74 MPa at maximum pressure. The total finite element mesh is shown in Figure 2.9-10. The three major parts (the lower part of the regenerator housing, the integral cooler, and the flow plate and ring nut) are shown separately in Figure 2.9-11. For simplicity, the cooler tubes will be replaced by three large, thin, concentric tubes having the same total cross sectional area as the cooler tubes. The boundary conditions, the loading, and the couplings between the parts are also shown in Figure 2.9-12. The brazed joint between the cooler and the flow plate is shown

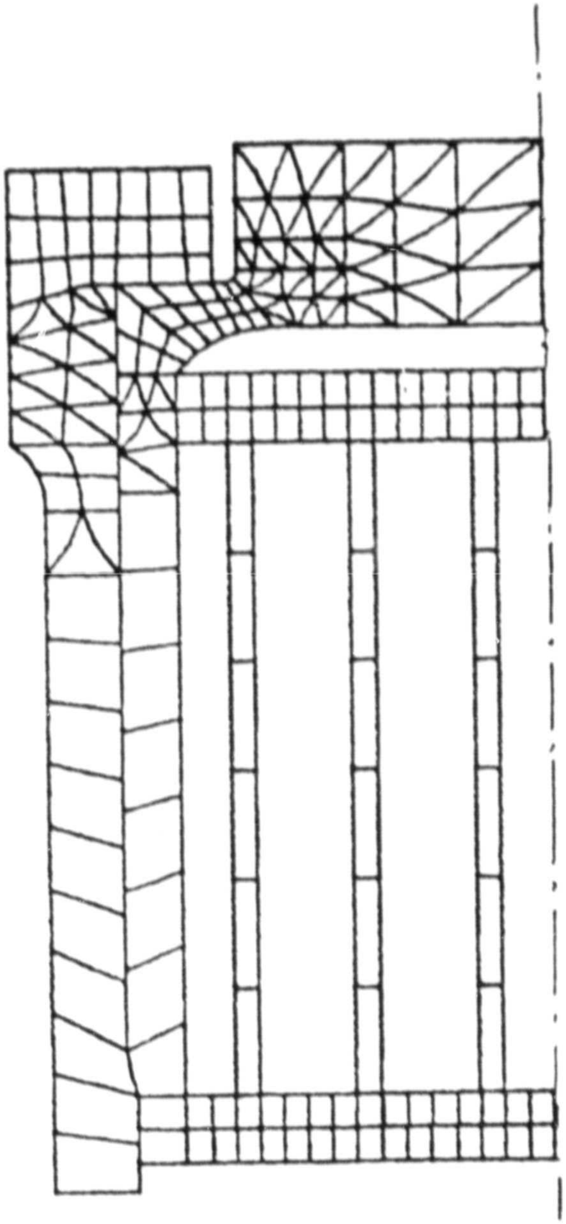

Figure 2.9-10 Finite Element Mesh for the Regenerator Housing, Gas Cooler, and Flow Plate Assembly

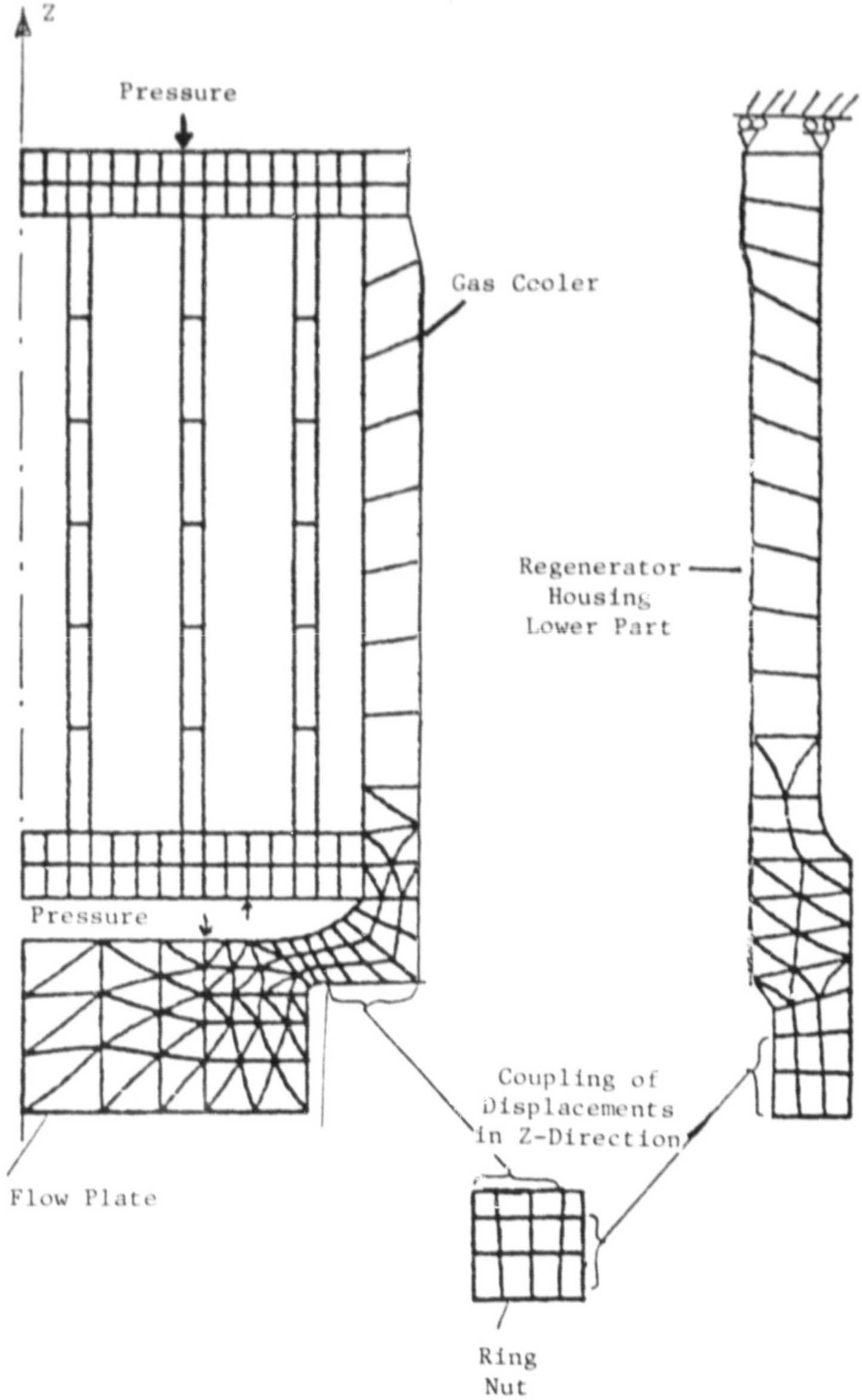

Figure 2.9-11 Boundary Conditions, Loading and Internal Coupling of the Regenerator Housing, Gas Cooler and Flow Plate Assembly

in Figure 2.9-12; the cooler and the flow plate are modelled as one part (i.e., a notch free joint).

Cylinder Block

A preliminary layout of the cylinder block was completed in December. The detailed design work will start in January, 1981.

MAJOR TASK 3 - TECHNOLOGY TRANSFER (BASELINE ENGINE)

Task 3.1 - Baseline Engine System (P-40)

MTI Engine Testing (ASE 40-7)

Engine characterization tests were initiated in October and a complete engine map was recorded for hydrogen as a working fluid and non-EGR operation In addition, a partial engine map with helium as a working fluid was also obtained before a major gas leak forced a stoppage in testing. Other problems encountered during testing were variator wear and oil and water leaks. The hydrogen leak was repaired by replacing the seal housing O-rings between the engine block and the crankcase. Some of the characterization data is shown in Figures 3.1-1 through 3.1-4.

The EGR system was made operational and engine characterization testing was reinitiated. However, variator performance deterioration limited maximum power output to 37 kW. Testing was stopped and the variator was repaired.

During December, the water pump pulley bolt sheared during testing. The failure caused the heater head, cooler, and piston dome O-rings to overheat, and as a consequence, a major hydrogen leak developed. During the engine teardown, contaminants were found in the combustion system; these contaminants will be analyzed to determine the potential source.

The engine was removed from the test cell during the rebuild in order to make the dynamometer available to check the performance of ASE 40-12 (P-40 Concord engine).

A total of 81 operating hours were accumulated during the quarter. The following is a summary of operating times to date.

Build 1	35.0 hours	(USSw)
Build 2	10.9 hours	(MTI)
Build 3	57.5 hours	(MTI)
Build 4	67.7 hours	(MTI)
Build 5	31.3 hours	(MTI)
Total	202.4 hours	

Spirit Engine (ASE 40-8)

A detailed, EPA approved, coastdown test was performed on the P-40 Spirit to determine the road load characteristics of the vehicle. The testing reconfirmed the CVS dynamometer power setting of 10.1 horsepower (without air-conditioning) that was previously used.

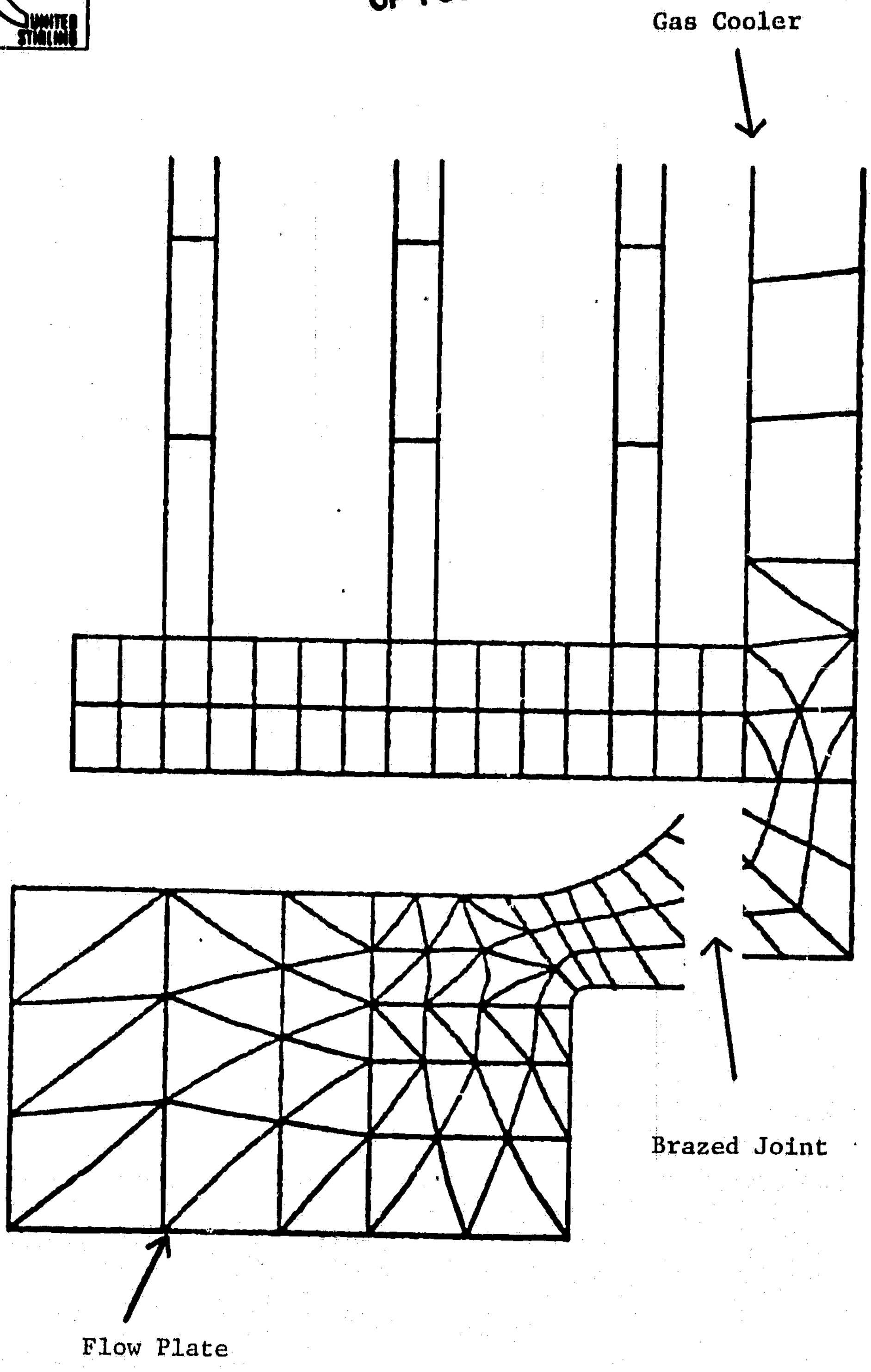

Figure 2.9-12 Gas Cooler — Flow Plate Brazed Joint

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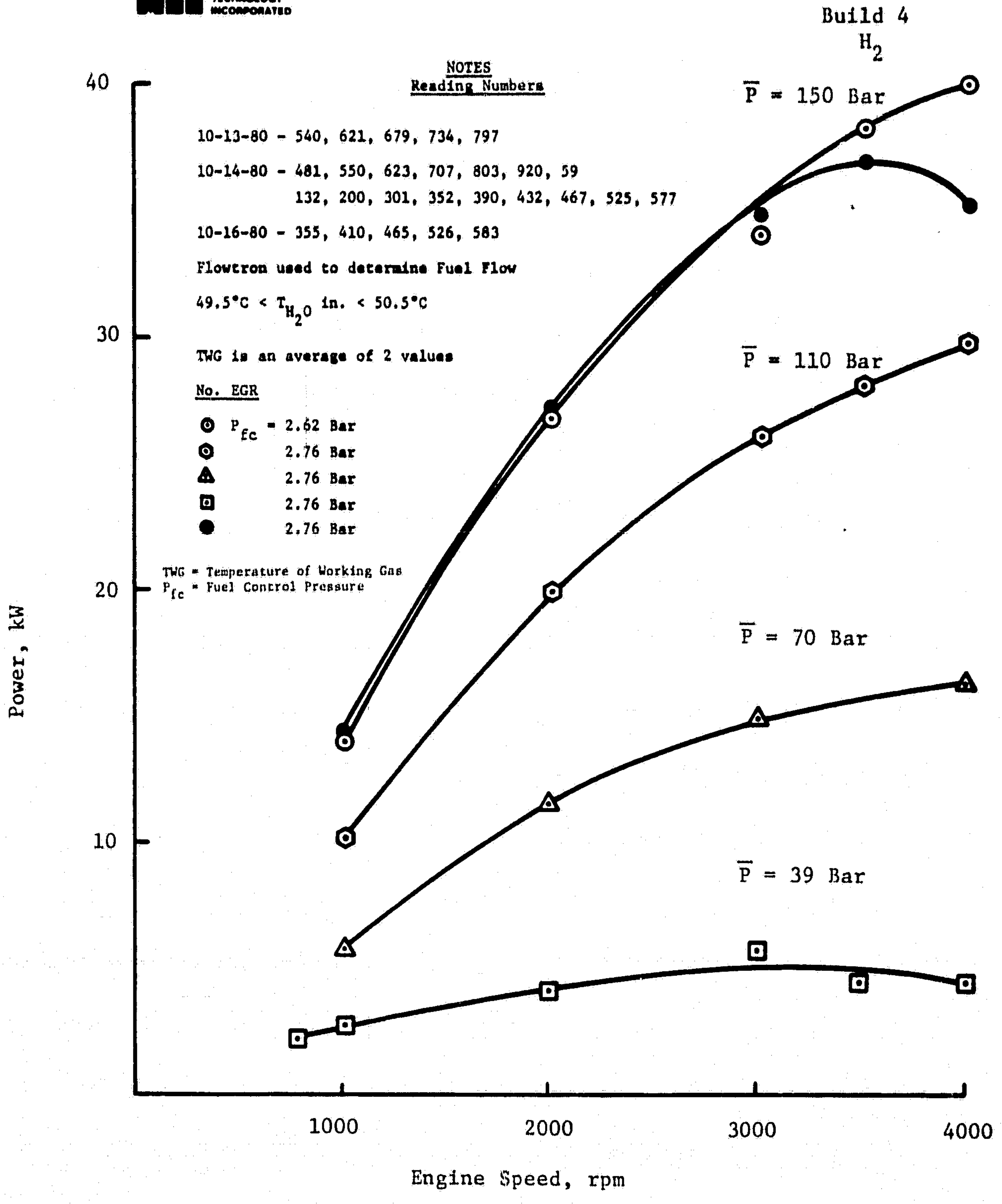

Figure 3.1-1 Performance Data: Power vs Speed (ASE 40-7)

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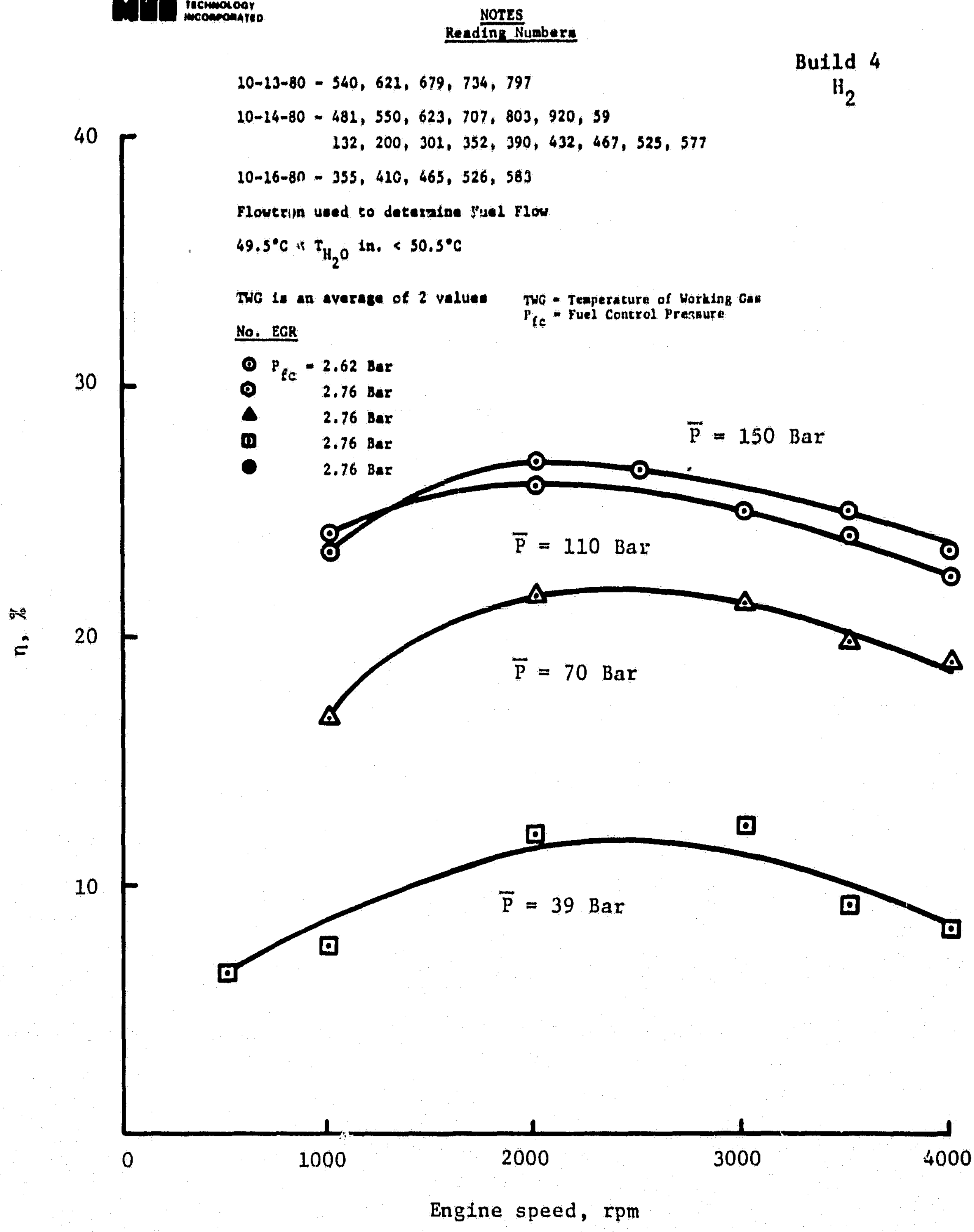

Figure 3.1-2 Performance Data: Efficiency vs Speed (ASE 40-7)

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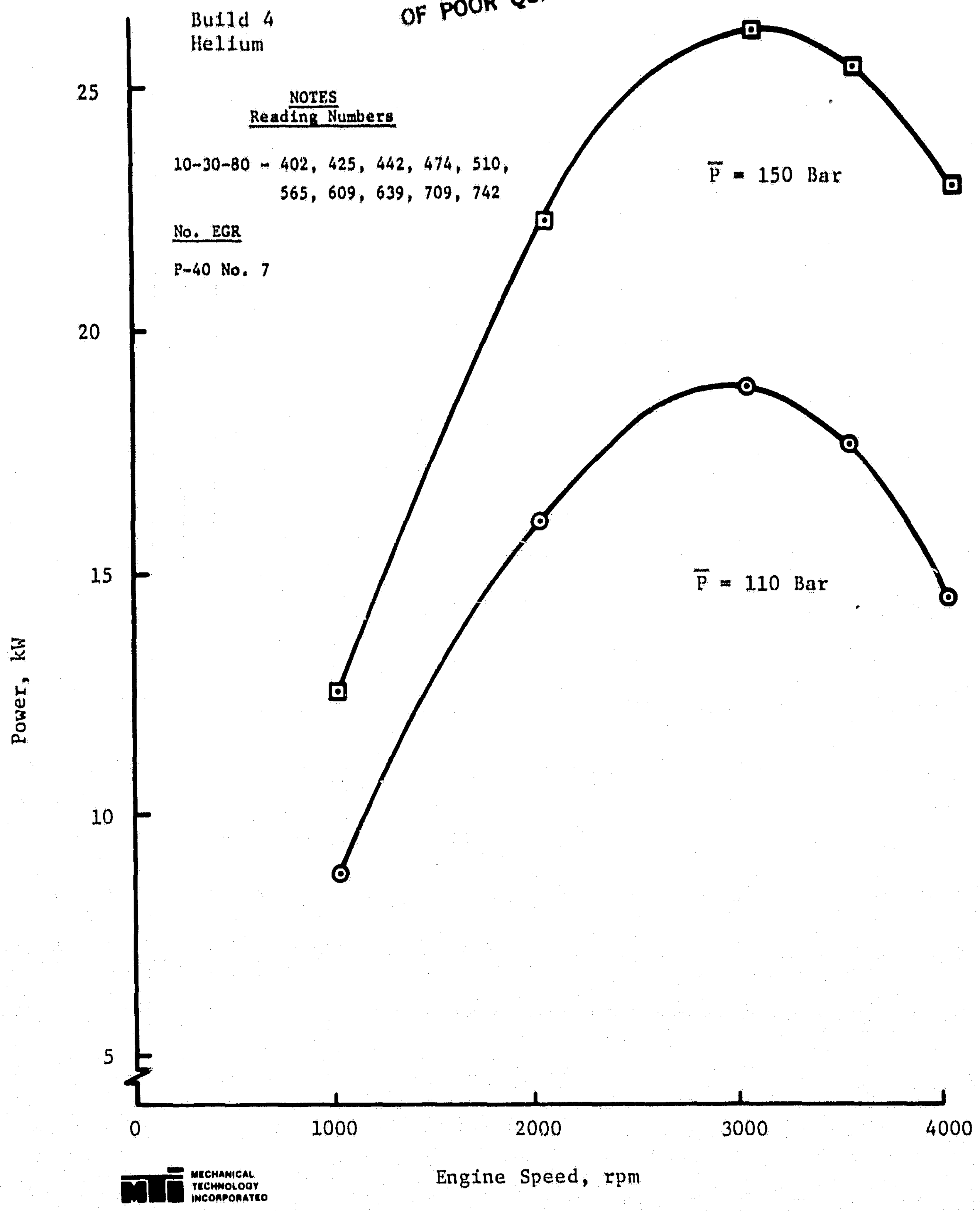

Figure 3.1-3 Performance Data: Power vs Speed (ASE 40-7)

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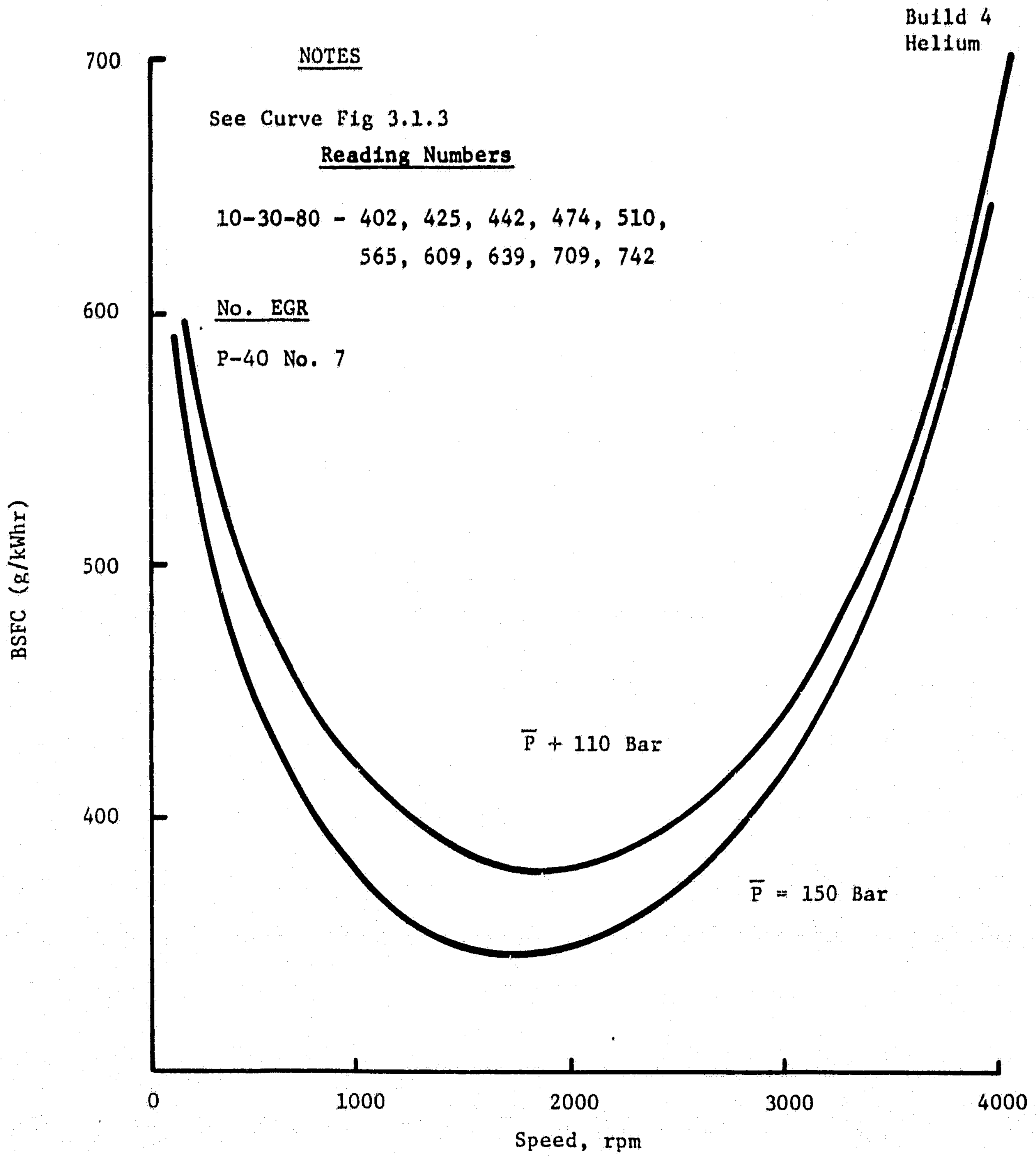

Figure 3.1-4 Performance Data: BSFC vs Speed (ASE 40-7)

A baseline CVS test was run on the vehicle. After the CVS test, a 0-60 mph acceleration was run on the vehicle dynamometer. It required approximately two minutes to reach 60 mph. Based on this poor vehicle performance, a decision was made to stop testing and repair the engine.

The engine was torn down and rebuilt in November. The problems encountered included:

- Glazed piston rings.
- One failed dome O-ring.
- Worn cap seals.
- One failed cap seal housing O-ring.
- Piston dome scuffing.
- Dirty coolers.

After the engine was reassembled, the power control valve was found to be leaking and was repaired. CVS tests were conducted but were aborted due to excessive engine vibrations. The condition cleared itself and was believed to be caused by dirt in a check valve. However, during subsequent operation, the moog valve acted erratically and the problem was diagnosed as misalignment with the electronic control.

The power control valve was realigned and baseline CVS and engine assessment tests were run. During the testing, problems were encountered with: combustion flameout, hydrogen leaks, variator performance, and fan clutch operation. Testing was finally stopped due to a high rate of hydrogen leakage. Leaks were found in two heater head quadrants: one leak was due to a crack between the tubes in the cylinder head manifold, which appears to be similar to that experienced on the HT P-40 engine; other leak is in the second row of tubes near the regenerator manifold. Because of the presence of the fins for the second row, the actual location of this leak could not be determined without further disassembly.

MAJOR TASK 4 - ASE MOD I

Design and Analysis

Heat Generating System

The following activities were completed during this quarter:

- All drawings of the heat generating system including the heater;
- Detail design of the CGR (combustion gas recirculation) bypass valve;
- Inclusion of USSw's heat generating system into the MTI microfilm library.

Testing of the CGR valve revealed a jammed condition at 150°C. Clearances on the valve were enlarged which eliminated the problem. The force required to operate the valve was 30 N.

The preparation for a CGR valve actuation method was started during this quarter. Two alternatives, hydraulic and electrical, will be investigated.

Cylinder Block and Gas Coolers

The pressure test equipment for the cylinder liner and gas coolers were analyzed. Calculations were performed in accordance with the Swedish Pressure Vessel Code and in the case of the block, finite element analysis was used.

Piston/Piston Rod Assembly

Durng the motoring test, it was evident that a slight redesign of the seal housing and the bolt connection for the dummy head was necessary. This was accomplished and new components were manufactured and mounted on the engine.

MTI and USSw seal materials screening tests and all other relevant literature show that the optimum surface finish for the mating surface of an unlubricated filled PTFE seal is an order of magnitude greater than what has been used until now. Therefore, the surface finish of the piston rod part which contacts the capseal was changed from a maximum of 0.063 m Center Line Average (CLA) to 0.2 -0.4 m CLA.

Engine Drive System

The detailed drawings of the revised crankshafts, counter balance weights, and reciprocating masses which are required for motoring tests were completed and checked. An additional drawing of the dynamic balancing assembly was prepared.

One way to reduce friction losses in the Mod I motoring unit would be to reduce the diameter of the big-end and main bearings. In November, stress calculations were made on the crankshaft to determine if the crank is strong enough to allow the reduced bearing sizes. The bearing friction were also calculated and design work was started on a reduced main bearing crank.

Engine Manufacturing and Procurement

USSw Mod I procurement is summarized in Table 4.0-1.

The following Mod I components were photographed and are included in this report:

- Piston Rod and Crosshead (Figure 4.0-1);
- Flange with Retaining Rings (Figure 4.0-2);
- Drive Unit (Figure 4.0-3);
- Motoring Test (Figure 4.0-4).

Engine Drive System

In December, two rebalanced crankshaft sets, which were modified to accept higher reciprocating masses, were received at Ricardo. One set was fitted to the No. 4 crankcase assembly and the remaining set, with front and rear drive shaft balance weights, was forwarded to USSw for fitting into the No. 2 Drive Unit.

USS PROCUREMENT PLANS FOR ASE MOD I

Task	Item description	Qty	Cont.type	Date of MR	Est cost SeCr x 10^3	Vendor(s)
84410	EXTERNAL HEAT SYSTEM					
84411	Heat generating system	10	17864	80-04-02	1330	FFV
84412	Preheater & housing	10	17854	80-04-02		FFV
84419	Air inlet manifold	10	17865	80-05-22	35	Karl Schmidt
"	" machining	10		80-06-04	10	Kockums
84420	HOT ENGINE SYSTEM					
84421	Cylinder/regenerator hsgs	40	17835/ 17837	80-01-17	434	Bulten-Kanthal
"	Heater	10	17268	80-04-02	800	FFV
84422	Regenerator	40	17041	80-03-11	54	Deran
84423	Gas cooler	40	17021	80-05-05	100	Leifa

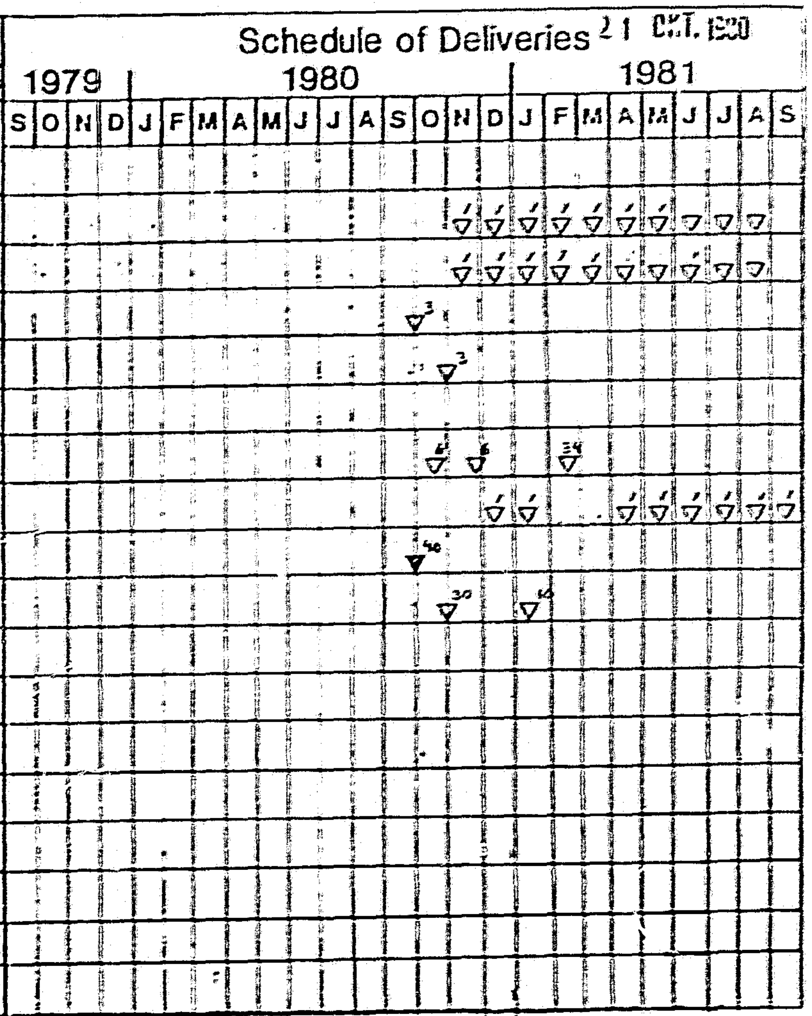

Table 4.0-1 Mod I Procurement Schedule

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USS PROCUREMENT PLANS FOR ASE MOD I

Task	Item description	Qty	Cost type	Date of LSR	Est.cost S-Cr x10^3	Vendor(s)	Schedule of Deliveries 1979–1981
84430	COLD ENGINE SYSTEM						
84431	Cold connecting duct	50	17025	80-05-05	13	Lidköping mek verkst	6, 44
	" machining	40		80-06-26	35	Cram-mekano	4, 12, 24
	Water jacket	14	17045	79-11-23	18	Karl Schmidt	14
	" machining	18		80-07-17	75	Cram-mekano/ Kockums	1, 1
	Cylinder liner	40	17231	80-06-12	20	Blomqvist Levang	32
84432	Seal housing	40		80-06-26	40	H Jönsson	4, 8
	Seals	40		80-05-1[illegible]	6	Frö-mekano	4
84433	Piston dome	40	17116	80-06-04	160	Kockums/ FFV	5, 6
	Piston & piston rings	40	17047	80-04-17	54	Kockums/ USS	4, 5
	Piston rod (forgings)	45	16078			FFV	45
	" machining	45		80-03-18	32	Addo	4, 4
84439	Dummy-heads	3	17126	80-06-13	60	Kockums	1, 2
84440	ENGINE DRIVE SYSTEM						
	Engine drive system	9					
	" (castings)	11			2*137	Ricardo	

Table 4.0-1 Mod I Procurement Schedule (Continued)

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USS PROCUREMENT PLANS FOR ASE MOD I							Schedule of Deliveries																								
Task	Item description	Qty	Cont type	Date of MR	Est.cost	Vendor(s)	1979				1980												1981								
							S	O	N	D	J	F	M	A	M	J	J	A	S	O	N	D	J	F	M	A	M	J	J	A	S
84450	CONTROLS & AUXILIARIES																														
84451	POWER CONTROL																						First unit available								
	Gas compressor	11		80-09-01																	▽										
	Control valve block	10		80-08-15																	▽										
	Subcomponents	various		80-08-01																	▽										
84452	AIR/FUEL SYSTEM																														
	Burner blower impeller/ Volute housing	12		80-08-15																	▽										
	Burner blower drive	12		80-10-15																	▽										
	Atomizer compressor	12		80-08-15																		▽									
	Subcomponents	various		80-08-01																	▽										
84453	AUXILIARIES																														
	Alternator	5		80-07-01																	▽										
	Power steering pump	3		80-07-01																	▽										
	A/C compressor	1		80-07-01																	▽										
	Starter motor	8		80-07-01															▽												
	Subcomponents	various		80-08-01																	▽										
84459	CONTROLS & AUX. NOT SPEC.																														
	Cooling system comp.	various		80-06-15																	▽										

UNITED STIRLING

Table 4.0-1 Mod I Procurement Schedule (Continued)

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Figure 4.0-1 Mod I Piston Rod and Crosshead

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Figure 4.0-2 Mod I Flange with Retaining Rings

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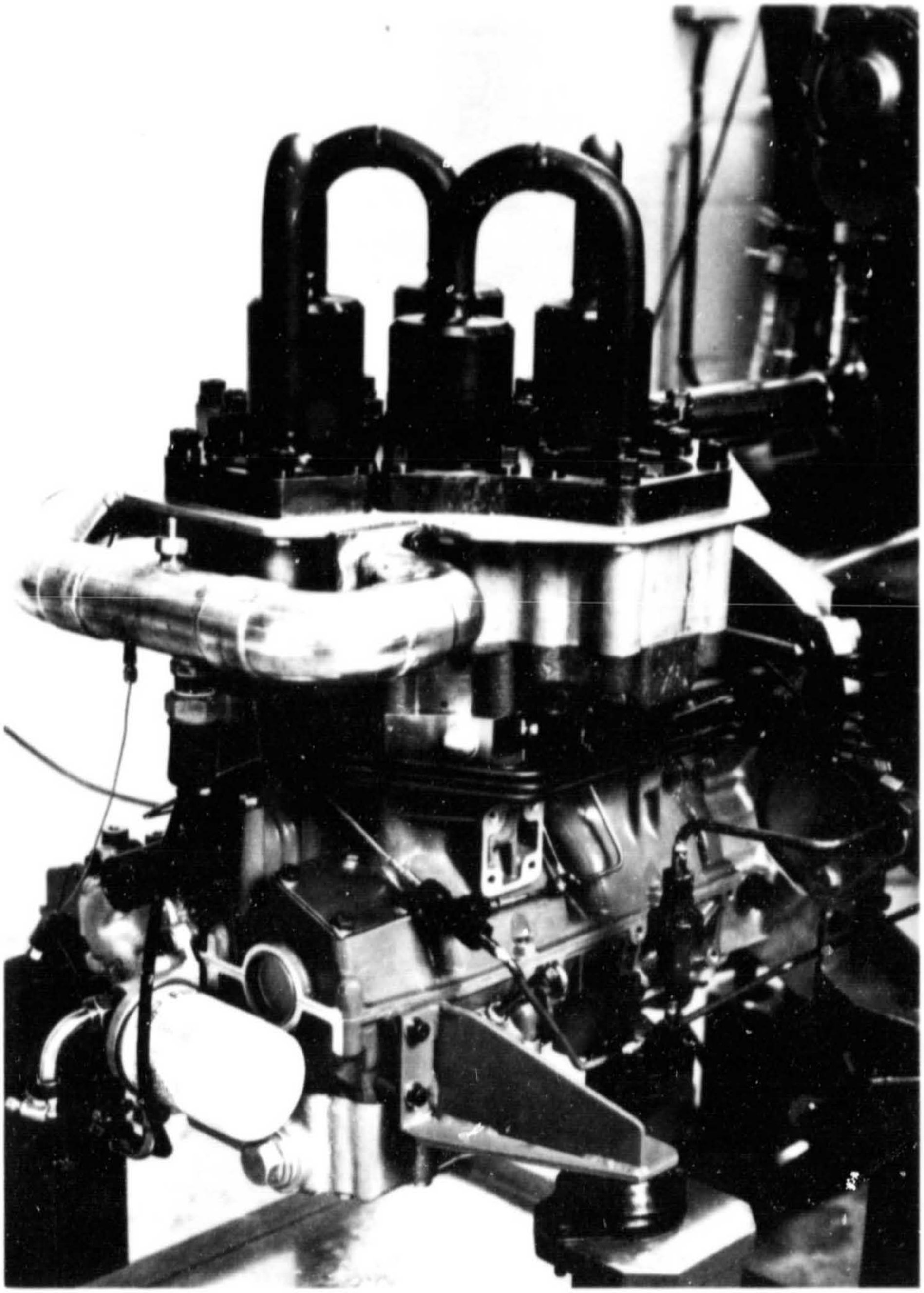
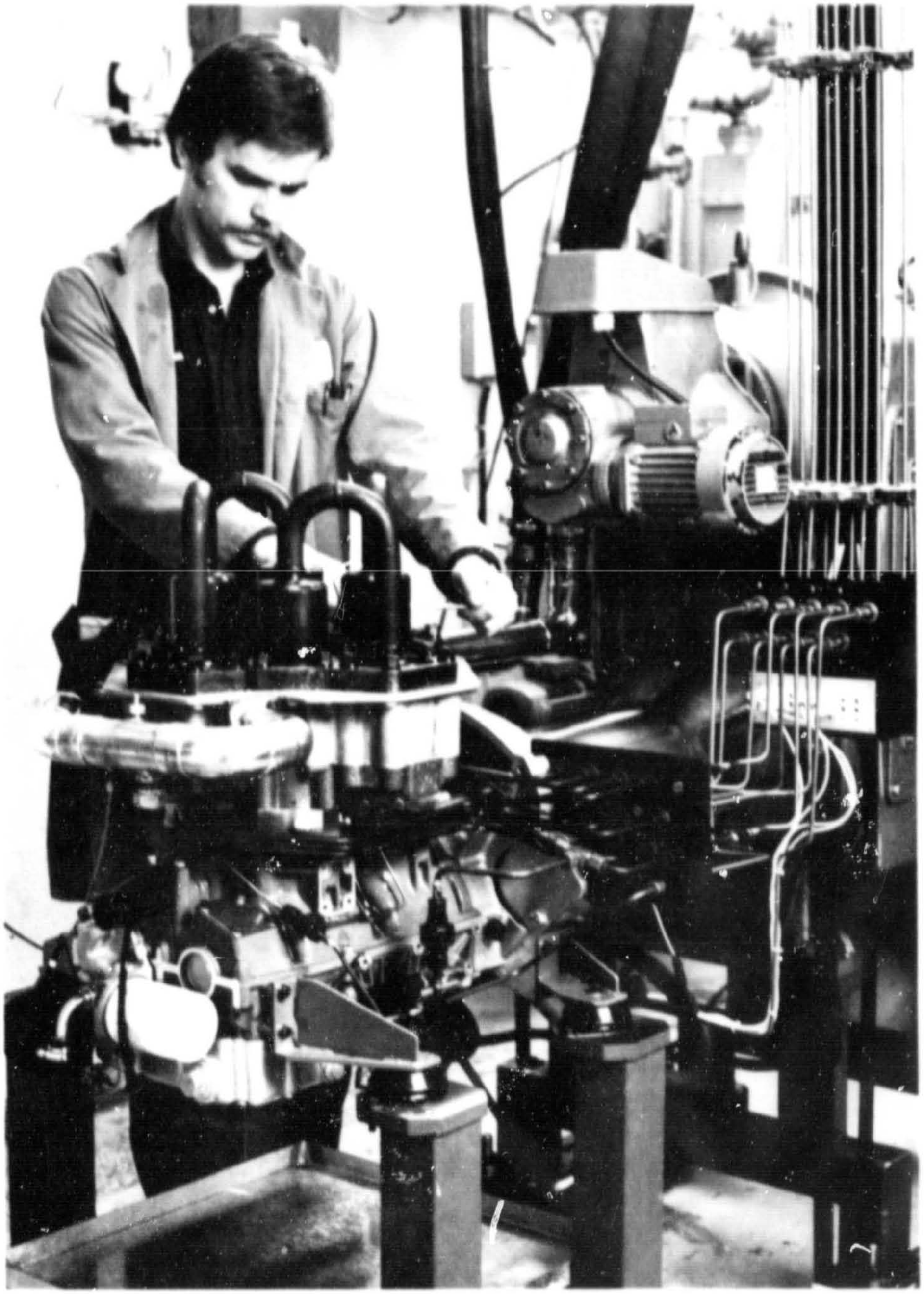


Figure 4.0-3 Mod I Drive Unit



UNITED STIRLING

Figure 4.0-4 Mod I Motoring Test

Remachined fine-tooth gears were also received, which enabled the Mod I drive units to be assembled.

At the end of this quarter, five sets of 0.8 module gears were undergoing final finishing after being corrected by a hard chrome treatment.

Motoring Test - Mod I No. 1

The motoring test started on October 2, 1980. The first four weeks were used for breaking in, functional tests, and preliminary measurements. During November, the motoring unit was functioning well enough to produce continuous motoring power curves. Near the end of November, all but the nitrogen measurements were performed. Testing was halted to replace the original design with an improved design for the piston domes, which were susceptible to weld cracks. The test measurements were then completed.

An O-ring failure and a seal housing replacement due to a missing oil hole were the only testing problems encountered. The failed O-ring was a gas O-ring on a cylinder liner, which was mounted there by mistake. The motoring unit was dismounted from the test rig was being rebuilt to become a basic engine.

Motoring Test - Mod I No. 2

The motoring unit was assembled and installed in the test rig. The motoring test was started on December 17, 1980. The breaking-in period of 10 hours of testing was passed on December 23. The motoring unit will be inspected early next quarter.

Motoring Unit - Natural Frequency Analysis

During the motoring test with the Mod I drive unit, higher friction losses were noted at different speeds. This may indicate a resonance problem for the piston rod/piston/dome system. A finite element analysis for both the P-40 and Mod I engines was performed and the results are shown in Table 4.0-2.

	Lowest Critical Natural Frequency (rpm)		
	Boundary Conditions		
Engine	Case I	Case II	Case III
P-40	7423	6896	3425
Mod I	9349	8731	4043

Table 4.0-2 Results of the Natural Frequency Analysis

As shown in Table 4.0-2, the Mod I had larger values than the P-40 in all cases. In conclusion, a resonance problem was not the cause of the higher friction losses.

Vehicle Modification and Engine Installation

Cooling System

A computer code for non-uniform airflow distribution over a radiator core was generated. Its output was compared with the results of radiator performance tests conducted at Blackstone Linden in November. It was found that the handbook Nusselt (Nu) versus Reynolds (Re) relationships give pessimistic predictions compared to the test results. In order to achieve better agreement between predicted values and test results, efforts were concentrated on obtaining appropriate correlations. Performance maps were generated using these correlations as input data.

Vehicle Modifications

The procurement of modified vehicle components was started and the 1981 Concord was received. Modifications were also initiated in preparation for installed radiator airflow tests.

Mod I - USA Build

The welded dome configuration was stopped because of tearing in the welds during final machine operations. C.B. Kaupp & Sons Inc. of Maplewood, N.J. was selected to deep draw the Mod I domes. Contracts were let for the heater head to Precision Castparts Corporation (PCC). The first proof castings are expected in May, 1981 with production peices by July, 1981. All pieces were on order for the drive unit with the exception of crankshafts and gears. Since the crankshafts for the US manufactured drive unit did not have the provision to accept chain or link drives, new crankshaft drawings were made in November. Gear orders, which require a six month lead time, will be placed. Machined castings for the cold connecting duct were ordered from USSw. Castings for the water manifold were also ordered from USSw and machining will begin when they arrive. Fans for the combustion air blower were completed along with machined housings. Cast housings which were due in December were rescheduled for a January delivery. Forgings of piston rods are on order from USSw. Matrix stampings for the preheater are currently on order from USSw and samples of the regenerator matrix are on order from Facet Industries (Filter Products Division).

MAJOR TASK 7 - COMPUTER PROGRAM DEVELOPMENT

The major effort during this period was directed toward:

- Developing the Engine Performance Code (STENSY);
- Revising of the Mechanical Drive Code (KINE);
- Analyzing of the appendix gap losses;
- Further development of the harmonic (first order) Stirling engine analysis code.

Engine Performance Code (STENSY)

After reformulating the cycle analysis relative to mass conservation, STENSY was applied to analyze a thirteen control volume model of a typical ASE Stirling engine. The simulation of a full cycle was successfully achieved, as shown in Figure 7.0-1. This was accomplished at a time step rate of 250 time-steps per cycle. Beyond the first cycle however, numerical noise resulted in unsatisfactory results, as shown in Figures 7.0-2 and 7.0-3. A number of changes were progressivelyly made to the program in an effort to eliminate the numerical noise problem. These included:

- Reducing the time-step rate to 100;
- Adding an EPISODE integration formula with a backward difference formula (BDF) up to order 5;
- Removing the Nordsieck history array from the Episode package time-step aligorithm, which implements formulas to order 5 in an implied manner.

This effort resulted in the elimination of the numerical integration main problem over multiple cycles.

Potential inadequacies in the iteration convergence criteria at each time step were also investigated. The iteration convergence criteria were modified to consider errors in both the integration parameters and their first time derivatives. Previously, the calculation of integration parameters at specified times not coincident with the automatic integration process resulted in either numerical instability or inefficient integration. As a specified time was approached, the time-step from the automatic integration process was altered to hit this time exactly. When the altered time-step was much smaller than the preceeding time-steps, numerical instability could result when automatic integration resumed at the next time-step. This instability resulted from error tolerances associated with the large time-steps being inconsistent with the smaller, altered time-step. This problem was resolved by implementing a linear interpolation scheme to obtain integration parameters at intermediate times.

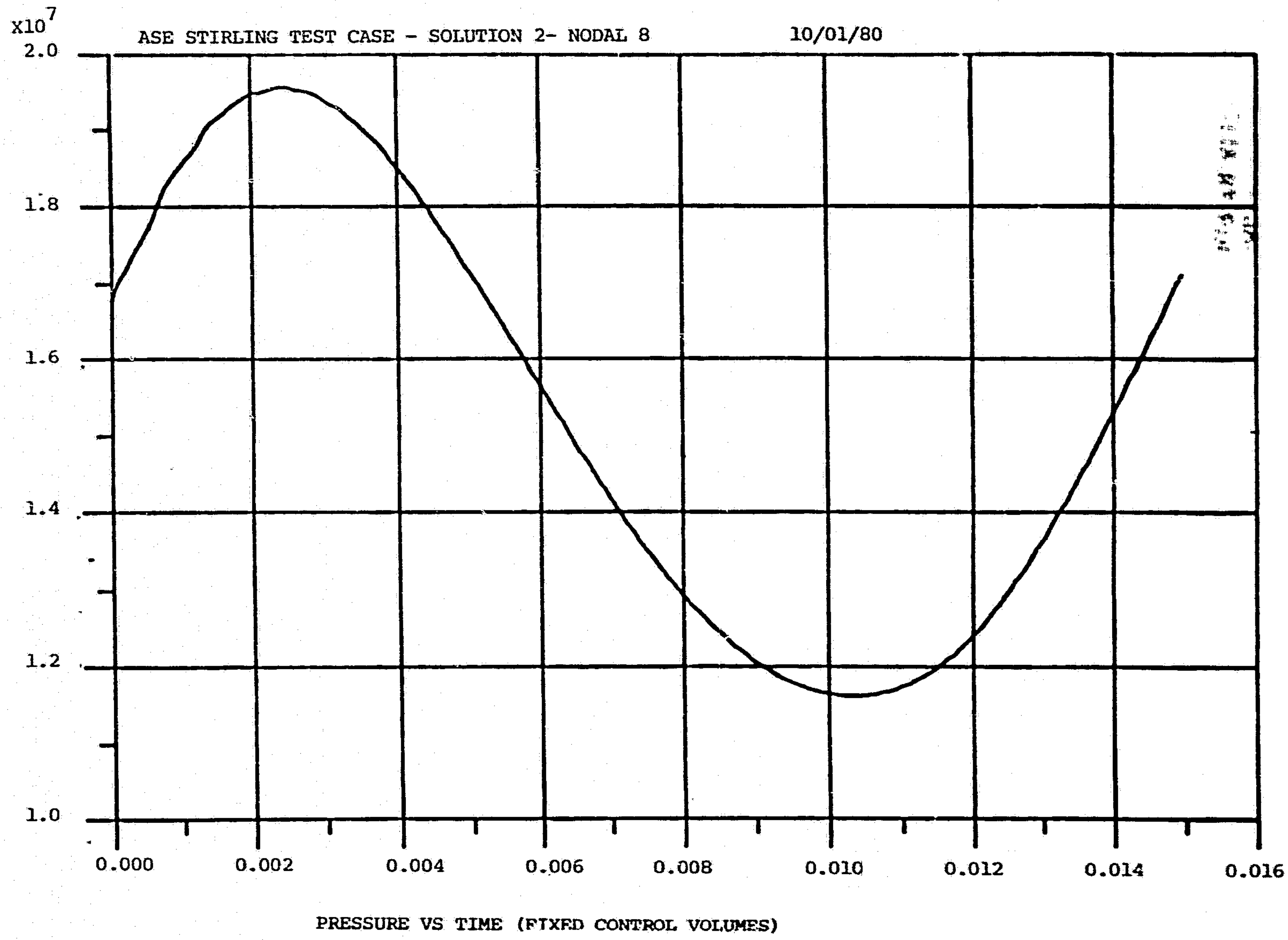


Figure 7.0-1 Compression Space Pressure Time History (250 Steps/Cycle)

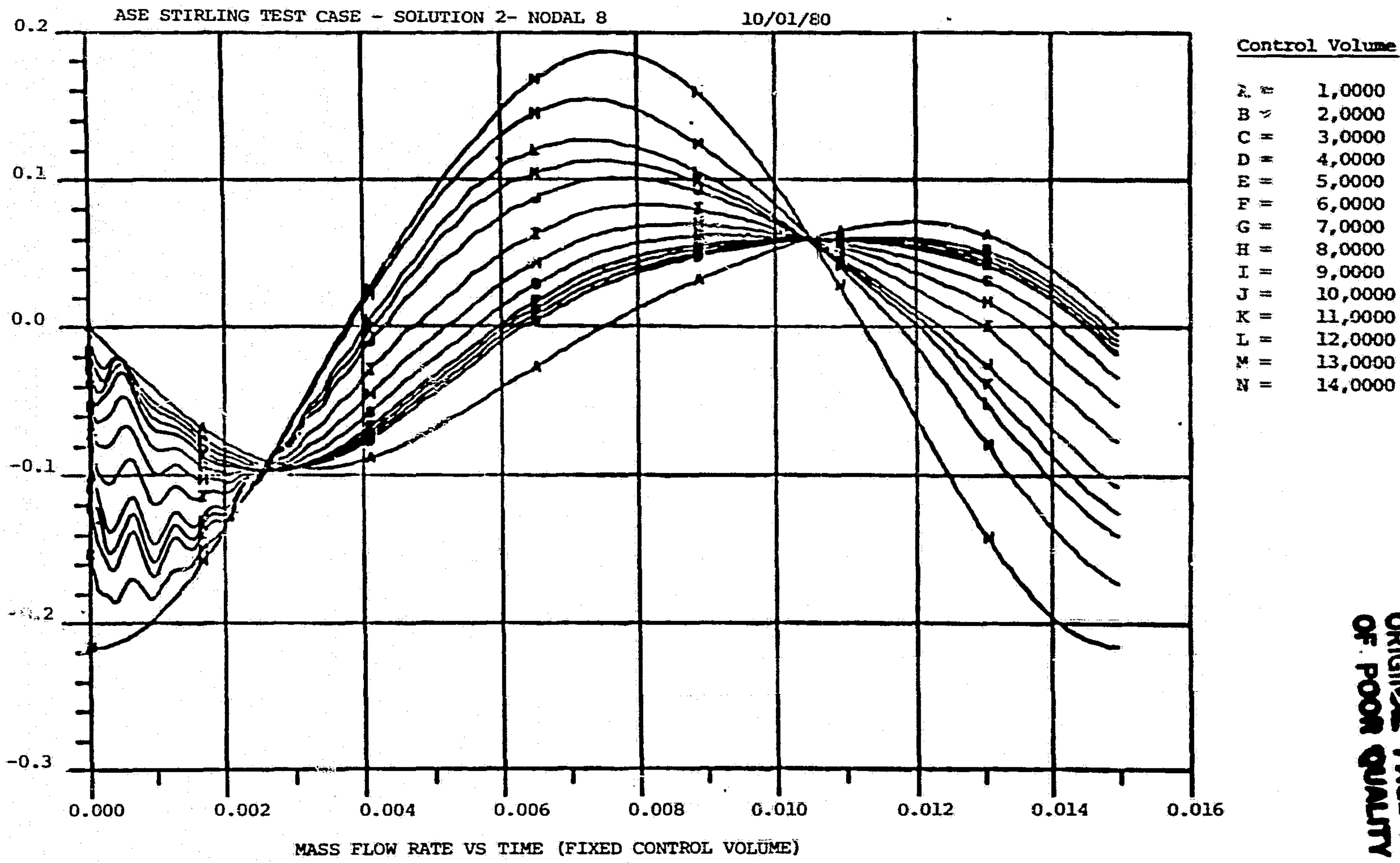


Figure 7.0-2 Mass Flow Rate Time Histories (250 Steps/Cycle)

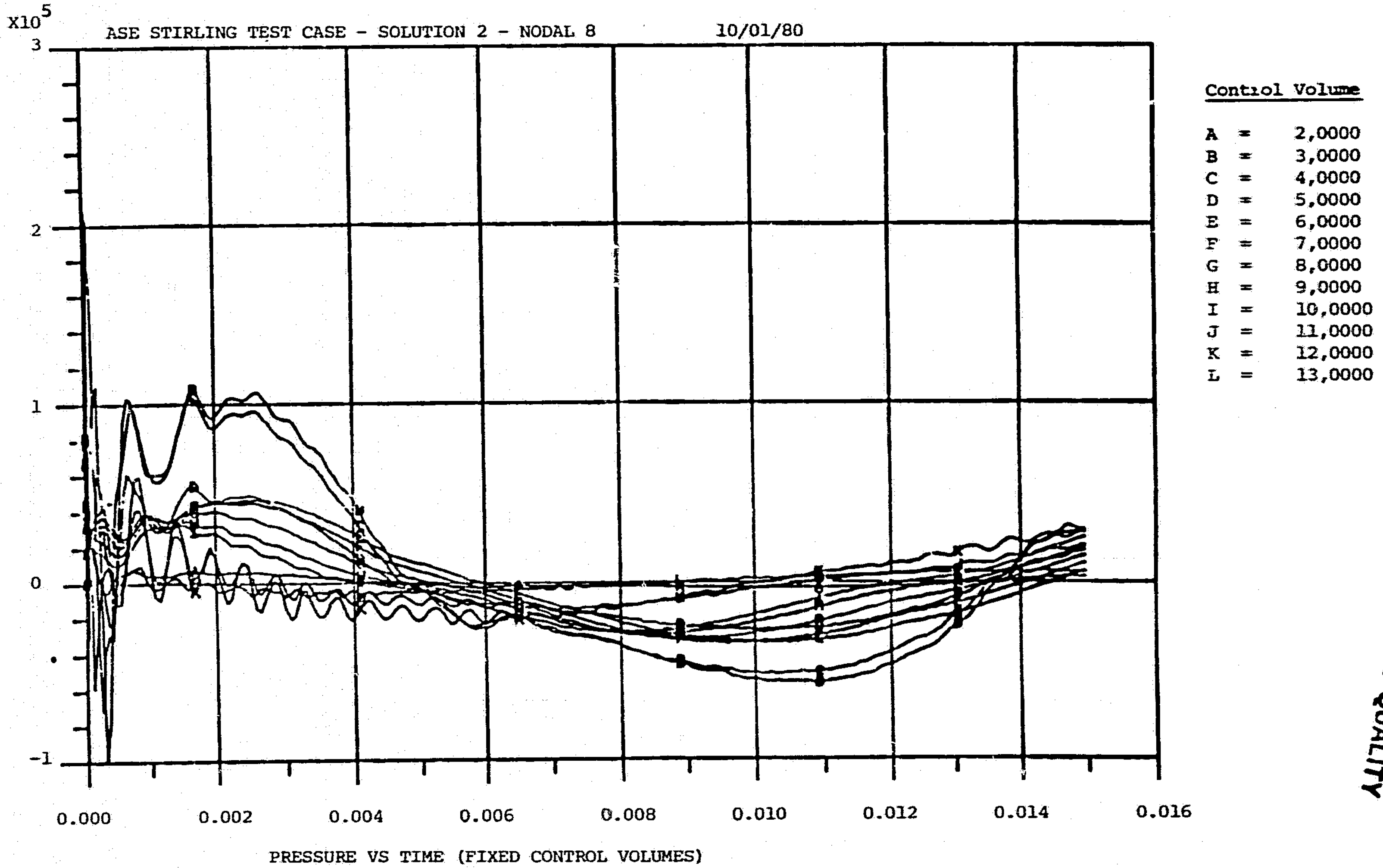


Figure 7.0-3 Pressure Drop Time Histories (250 Steps/Cycle)

Mechanical Drive Code (KINE)

The revision of KINE was completed and documentation of the code was started.

Appendix Gap Losses

The appendix gap loss is essentially composed of the following three components: shuttle heat transfer in the displacer wall; gas enthalpy transport down the appendix gap; and hysteresis work associated with heat transfer from the gas to the cylinder. A closed form analysis of these losses was formulated based on the following assumptions:

- Negligible leakage past seal;
- Two dimensional rectilinear flow in gap;
- Viscous flow;
- Prescribed pressure variations;
- Constant gas properties;
- Sinusoidal displacer/piston motions;
- Identical and linear axial temperature profiles in displacer and cylinder;
- Parallel-sided flow channel;
- Significant convective heat transfer.

This analysis was applied to P-40 geometry and the operating conditions corresponding to the USSw appendix loss tests. The present analysis qualitatively correlates with the USSw appendix loss test results. Figures 7.0-4 and 7.0-5 show both the new appendix loss analysis as a function of non-dimensionalized parameters and actual USSw test data points. The experimental data used is summarized in Table 7.0-1. For comparison purposes, the evaluation assumes correlation at a nominal gap of 0.3 mm and that all changes in engine output power result solely from changes in appendix losses. It can be seen that there is a general agreement betwen the test data and the prediction technique.

Harmonic Stirling Code

The development of harmonic models for the heat exchangers and working spaces was initiated with a literature survey. The objective of this effort is to improve the "single blow" models which are currently being used in the MTI Harmonic Stirling code.

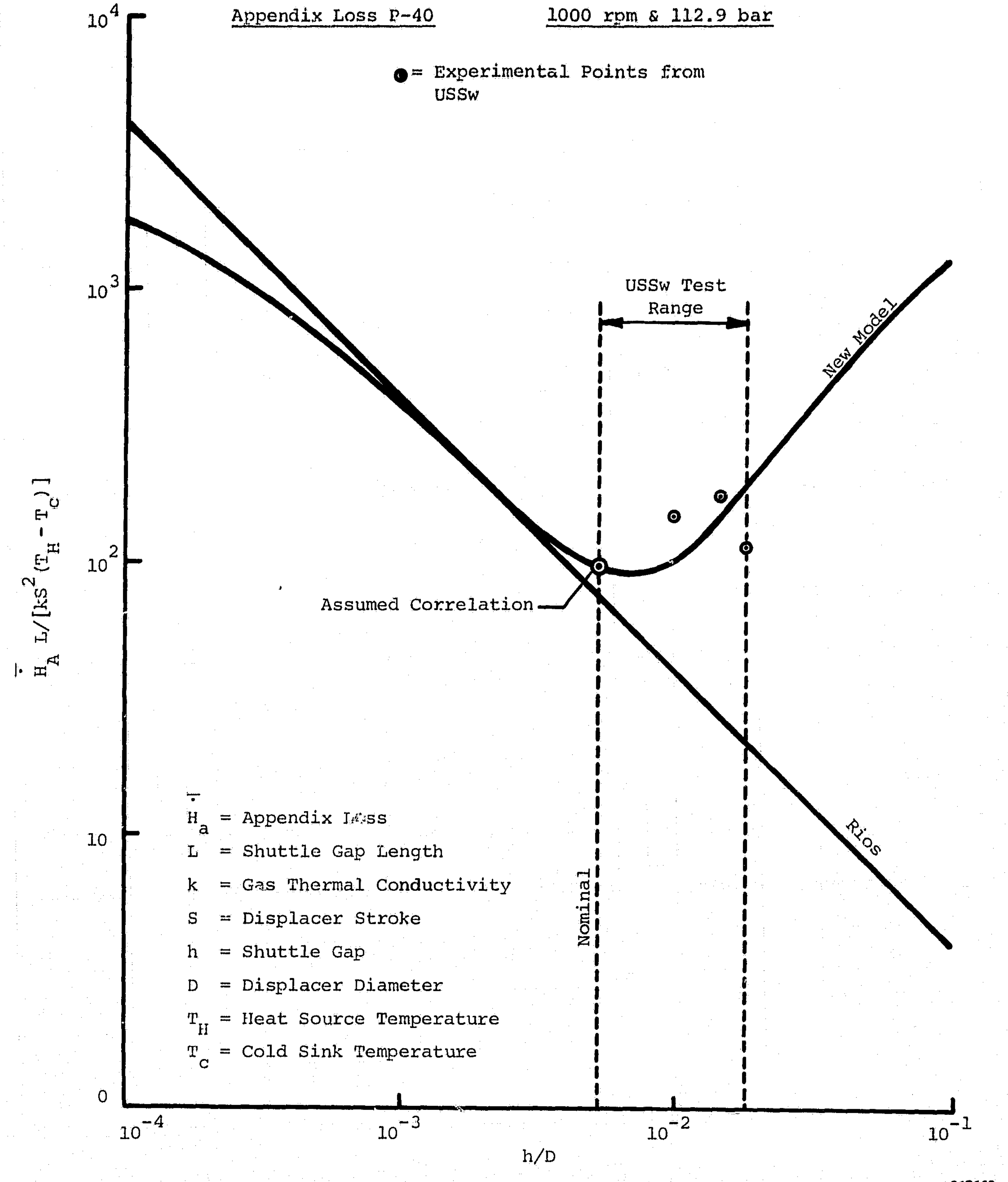


Figure 7.0-4 P-40 Appendix Loss — Gap Variation at 1000 rpm

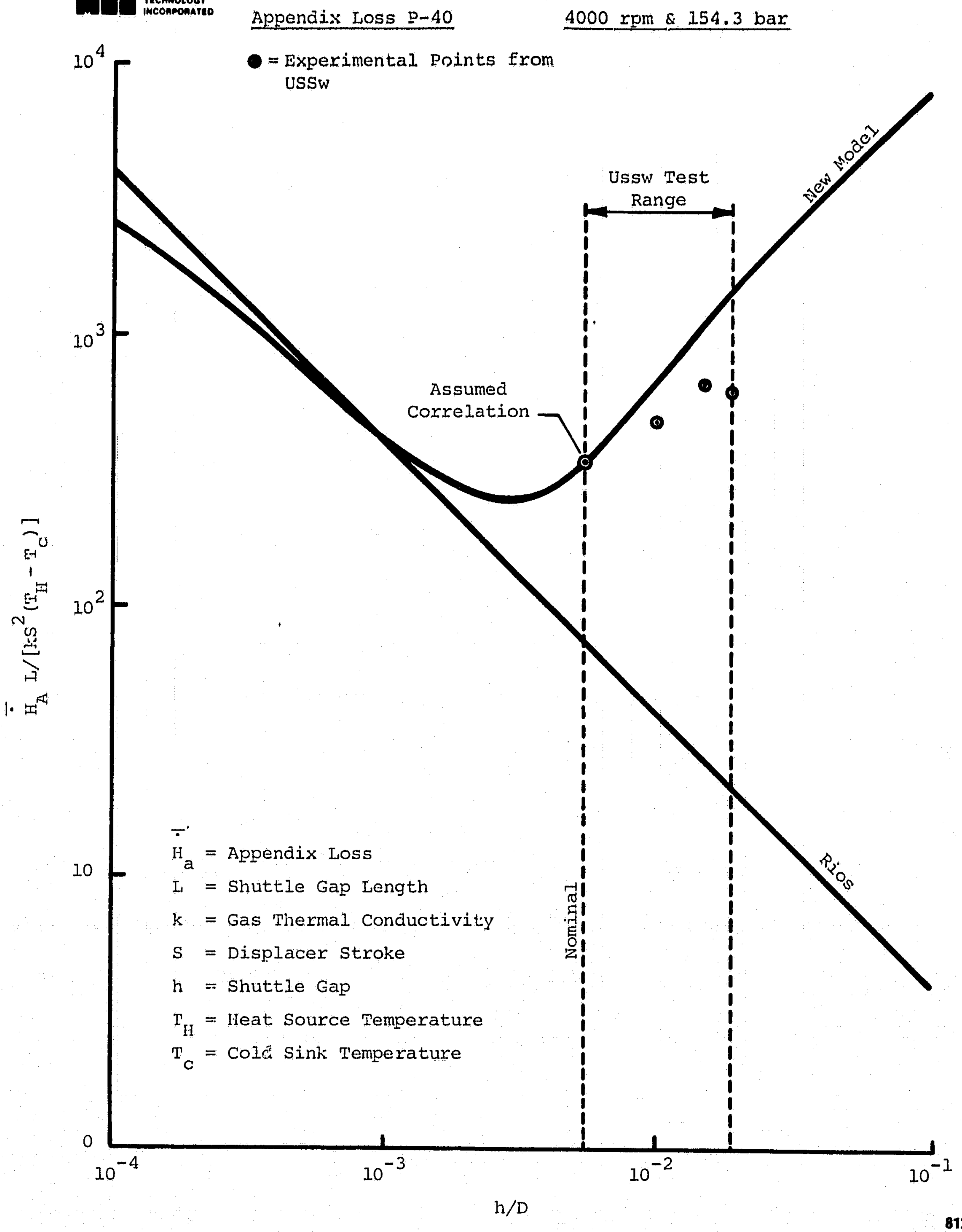


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Figure 7.0-5 P-40 Appendix Loss — Gap Variation at 4000 rpm

	Output Power Change	
	Test Point	Test Point
Gap	1	2
0.30 mm	0	0
0.55 mm	- 0.30 kW	- 0.84 kW
0.80 mm	- 0.42 kW	- 1.86 kW
1.05 mm	- 0.06 kW	- 1.50 kW

Test Point 1: N = 100 rpm

P(mean) = 11 MPa

Power = 10.75 kW

Test Point 2: N = 4000 rpm

P(mean) = 15 MPa

Power = 44.18 kW

Table 7.0-1 Changes in P-40 Power Output with Appendix Gap

MAJOR TASK 8 - TECHNICAL ASSISTANCE

During the quarter the following shows were attended and demonstrations held:

- In October, the ASE Program display was exhibited at the World Energy Engineering Congress in Atlanta. The P-40 Concord and an ASE Program display were exhibited at the Flint and Grand Rapids Energy Expo's.

- The Concord with a mockup engine and the ASE Program display were shown at the Los Angeles, San Francisco and Fresno Energy Fairs.

- The P-40 Concord and the ASE Program display were also exhibited at the New Hampshire Energy Show.

- In November, the P-40 Concord was exhibited at the DOE Automotive Technology Development Contractor Coordination Meeting (CCM) in Dearborn, Michigan. In addition to display material and the synchronized slide/tape show "The Stirling Engine - Helping to Solve our Nation's Energy Problem," the ASE Mod I model engine was shown and demonstration rides were given to meeting participants. While the CCM was in session, the last of the Michigan Energy Expo's was being held at Cobo Hall in downtown Detroit. Both the Concord with the mockup engine and the P-40 Concord were utilized for a static-display at this show which drew an estimated attendance of 100,000 people.

- In December, the P-40 Concord and ASE Program Exhibits were shown at the Hudson Valley Community College Automotive Mechanics Society Show in Troy, N.Y. Attendance was approximately 2000 people and a great deal of interest was shown by the automotive mechanics students.

In addition to shows and demonstrations, the following other activities occurred during the quarter:

- Preparations were made and exhibits designed for the SAE show in Detroit in February.

- Preparations were made to attend the 8th Energy Technology Conference in Washington D.C. in March.

- Work was initiated on the rebuild of the P-40 Concord at MTI. Initial baseline vehicle performance data were taken on a chassis dynamometer and during highway operation. A very detailed disassembly of the engine, with integral parts inspection, is now underway. Following the reassembly of the engine, the vehicle dynamometer and highway runs will be repeated for comparison with the baseline data. The vehicle is expected to be operational the end of January.

MAJOR TASK 9 - PROGRAM MANAGEMENT

- **MTI Product Assurance Program**

  - The "MTI Product Assurance Manual", MTI Report No. 81ASE176PA8, was issued on December 12, 1980.

  - A Swedish version of the approved United Stirling Product Assurance Manual was received and translated into English by MTI. A review of the Product Assurance Manual and program was scheduled for the week of January 26, 1981 at United Stirling.

  - The preliminary list of the critical parts of the ASE Mod I was reviewed at the monthly meeting at United Stirling. The list was finalized and issued in January, 1981.

- **Safety Study**

  - The MGA Safety Study Report, MTI Report No. 81ASE175ER13, was issued on December 12, 1981.

- **Operating Time/Failure Report**

  - A summary of accumulated operating time versus failures per month as of December 31, 1980 are as follows:

Engine	Operation Time	Mean Operating Time to Failure (Hours)
ASE 40-1 (NASA)	204.2	5.67
ASE 40-7 (MTI)	202.4	10.12
ASE 40-8 (SPIRIT)	245.4	3.36
ASE 40-12 (CONCORD)	67.3	9.6
ASE 40-4 (USSw-HIGH-TEMP)	5,620.7	85.16
ASE 40-13 (USSw-ANNULAR)	140.6	10.82

  - A summary of the Failure Notices and Discrepancy Notices as of December 31, 1980 are as follows:

Open Failure Notices	84
New Failure Notices since November 29, 1980	17
Failure Notices closed since November 29, 1980	0
Closed Failure Notices (total to date)	126
Total Failure Notices in System	210
P-40 Failure Notices	188
Mod I Failure Notices	22

Open Discrepancy Notices	84
New Discrepancy Notices since November 29, 1980	15
Discrepancy Notices closed since November 29, 1980	0
Closed Discrepancy Notices (total to date)	67
Total Discrepancy Notices in System	151
P-40 Discrepancy Notices	99
Mod I Discrepancy Notices	52

4.0 REFERENCES

1. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report". Period: January 1 - March 31, 1979. MTI Report No. 79ASE67QT4 (available as Government Report: DOE/NASA/0032-79/2 or NASA CR-159606 from NTIS).

2. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report". Period: April 1 - June 30, 1979. MTI Report No. 79ASE88QT5 (available as Government Report: DOE/NASA/0032-79/3 or NASA CR-159610 from NTIS).

3. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report". Period: July 1 - September 30, 1979. MTI Report No. 79ASE101QT6 (available as Government Report: DOE/NASA/0032-79/5 or NASA CR-159744 from NTIS).

4. "Stirling Demonstration Vehicle, Genesis I". MTI Report No. 78ASE26PR3a, October 1978.

5. "Pre-Developmental Demonstration of a Stirling-Powered Vehicle, Genesis I MTI Report No. 79ASE33T01, January 15, 1979.

6. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report". Period: October 1 - December 31, 1979. MTI Report No. 80ASE116QT7 (available as Government Report: DOE/NASA/0032-80/6 or NASA CR-159827 from NTIS).

7. "Assessment of the State of Technology of Automotive Stirling Engines". MTI Report No. 79ASE77RE2 (available as Government Report: DOE/NASA/0032-79/4 or NASA CR-159631 from NTIS).

8. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report". Period: January 1 - March 29, 1980. MTI Report No. 80ASE129QT8 (available as Government Report: DOE/NASA/32-80/7 or NASA CR-159851 from NTIS).

9. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report". Period: March 30 - June 29, 1980. MTI Report No. 80ASE130QT9 (avalable as Government Report: DOE/NASA/32-80/8 or NASA CR-165134 from NTIS).

10. "Automotive Stirling Engine Development Program Quarterly Technical Progress Report." Period: June 30 - October 3, 1980, MTI Report No. 81ASE163QT10 (available as Government Report No. NASA CR-165194, DOE/NASA/0032-81/9 from NTIS).

11. "Material for Reciprocating Seals on an Automotive Stirling Engine" - MTI Report No. 81ASE158PR11.

APPENDIX A

PISTON RING LEAKAGE

If there is no leakage across the piston rings the engine consists essentially of four separate, identical gas cycles. Ignoring frictional losses etc. during each revolution of the engine, each cycle produces a specific amount of work W1. Ideally, $W1 \propto M1$ where M1 is the mass of gas in one cycle. The total power from the engine, H = 4 W1N where N is the speed in revs/sec.

The mean cycle pressure $\overline{P}$ is also proportional to M1, therefore, $H \propto \overline{P}N$.

If the mass of gas leaking from each cycle across the piston rings during each revolution of the engine is small compared with the total mass in each cycle, there will be a negligible change in the mean operating conditions but the amplitude of the pressure variations in each cycle will be reduced with a consequent reduction in the engine power.

A simple approach is to consider that the mass leaking from a cycle during one revolution plays no part in the cycle i.e., the effective mass in the cycle is reduced by the leakage mass.

Let $\Delta M1$ be the mass leaking from one cycle/revolution.

Effective mass/cycle = $M1 - \Delta M1$

Work done/cycle/rev = $W1^1 = \left(\frac{M1 - \Delta M1}{M1}\right) W1 = \left(1 - \frac{\Delta M1}{M1}\right) W1$

Total power with leakage = $4W1^1N = \left(1 - \frac{\Delta M1}{M1}\right) H = H^1$

i.e. the power decreases in direct proportion to the leakage.

This is obviously a very simple model of the effect of leakage but it is representative of the system if $\frac{\Delta M1}{M1} << 1$

Each gas cycle is bounded by two pistons and each piston has two rings. Between the two piston rings the pressure is maintained at the minimum cycle pressure, P min as shown in Figure 1.

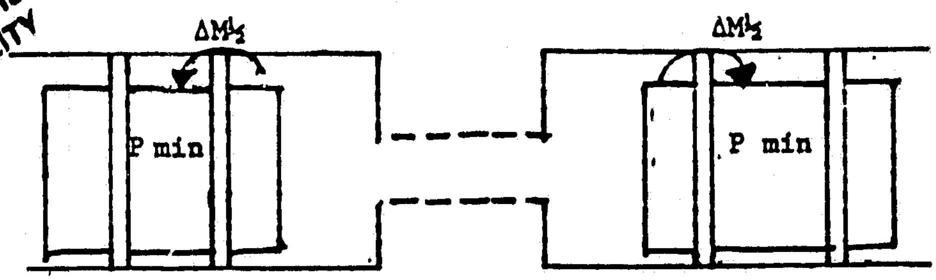

Figure 1

In the simpliest case, the leakage will be the same across each piston ring, or during each revolution the mass leaking across each piston ring is $\frac{\Delta M1}{2}$. The average mass leak rate per ring is $\frac{\Delta M1}{2} \times N$.

For the P40 engine with hydrogen $M1 = 1.474 \times 10^{-10}\ \overline{P}$ kg.

If $\Delta M1 = 0.01\ M1$ (1% leakage).

$$\Delta M1 = 1.474 \times 10^{-12}\ \overline{P}\ \text{kg.}$$

If $\dot{m}_L$ is the mass leakage rate across each piston ring

$$\dot{m}_L = \frac{\Delta M1}{2} N = 0.737 \times 10^{-12}\ \overline{P}\ N\ \text{kg/s.} \qquad \text{———} \quad 1$$

Based on a 1% reduction in power the 'acceptable' leakage rate will obviously vary with the operating conditions. For the modest conditions of $\overline{P} = 5 \times 10^6\ \text{N/m}^2$ and $N = 15$ rev/s (900 rpm), $\dot{m}_L = 55.3 \times 10^{-6}$ kg/s.

'Equivalent' Test Rig Conditions

In the engine the pressure difference across the piston rings varies during the cycle but in the test rig the pressure difference will be constant.

If the leakage is dependent only on the pressure difference, an 'equivalent' test rig condition would be one in which the constant rig pressure difference is equal to the mean pressure difference in the engine cycle.

For the P40 engine the cycle pressure, P is given by

$$P = \overline{P}\,[1 + 0.25\ \sin\theta]$$

$$P\ min = 0.75\ \overline{P}$$

$$\Delta P = P - P\ min = 0.25\ \overline{P}\ [\ 1 + \sin\theta]$$

$$\Delta Pav = 0.25\ \overline{P}$$

If the gas in the test rig discharges to atmosphere and the equivalent gas supply pressure is P_G then, from above:

$$P_G = 0.25\ \overline{P}$$

Some examples of engine/rig conditions are given in Table I assuming that 1% of the gas in a cycle is lost by leakage every revolution.

A very simple model has been used to assess the effects of leakage but nevertheless it should be valid if the mass of gas lost by leakage is a small proportion of the total mass of gas. The data of Table I is for a 1% leakage, corresponding to a 1% reduction in power. At this level the effects of leakage should be represented to a reasonable accuracy.

The most significant feature of the data is that while the leakage per revolution may be small, the leakage <u>rate</u> will be quite large and particularly when represented in terms of ambient volume. For the examples given, the smallest ambient leakage rate for one ring is of the order of 1000 cm^3/s. At this level, a flow meter would be more suitable for measuring leak rate than a mass spectrometer type leak detector.

Table 1

	ENGINE CONDITION			
	Speed = 1000 RPM		Speed = 3000 RPM	
	$\overline{P}$ = 5 MPa	$\overline{P}$ = 15 MPa	$\overline{P}$ = 5 MPa	$\overline{P}$ = 15 MPa
'Equivalent Rig Pressure' P_G (MPa)	1.25	3.75	1.25	3.75
Mass leakage rate of hydrogen across one piston ring (kg/s)	6.14×10^{-6}	0.184×10^{-3}	0.184×10^{-3}	0.553×10^{-3}
*Ambient volumetric leakage rate of hydrogen across one piston ring (cm^3/s)	748	2244	2243	6729

*Ambient conditions taken as 0.1 MPa, 20°C

APPENDIX B

DUTY CYCLE FOR SEALS TEST RIGS

According to the RESD vehicle specification, the engine design must be able to meet either of the following conditions:

- 55,000 miles of driving over the Urban Driving Cycle and 45,000 miles of driving over the Highway Driving cycle;

- 100 hours at maximum power.

It is desirable to devise simplified duty cycles which can be used for rig testing of components such as new seal designs.

USSw has proposed a simplified engine test cycle for use in evaluating alternative heater head materials. The cycle is based on mean pressure variations computed for the RESD in a vehicle driven over a combined driving cycle (55 percent urban driving and 45 percent highway driving). The mean pressure variations were analyzed from a low cycle fatigue point-of-view and linear creep damage theory was used to establish an equivalent creep damage for the simplified duty cycle. The resulting duty cycle is shown in Figure B-1. The duty cycle would be repeated to accumulate the test hours required.

Although the USSw duty cycle could also be used for testing new seal designs, MTI has proposed a possible alternative duty cycle based on equivalent seal wear. This model is based on the following assumptions:

- Wear is proportional to the product of seal load and sliding distance.
- Seal load is proportional to mean engine pressure.
- Mean engine pressure is proportional to torque.

These assumptions lead to the following relationship for seal wear.

Wear   torque x time x speed

Computer predictions of torque-speed relationships for urban and highway driving cycles were used to establish equivalent engine operating conditions (speed and mean working gas pressure), as shown in Tables B-1 and B-2. This analysis established the following equivalent engine conditions for the driving cycles.

- Urban driving cycle   841 rpm, 4.68 MPa
- Highway driving cycle   1136 rpm, 6.98 MPa

The proposed duty cycle for seals test rigs is shown in Figure B-2. In addition to the regions of the duty cycle which represent urban and highway driving cycles, a 20-second interval at maximum working gas pressure is included, followed by a cool-off period where seals are allowed to return to near ambient temperature. This cool-off period is important since it is known to affect the ability of a seal system to function properly.

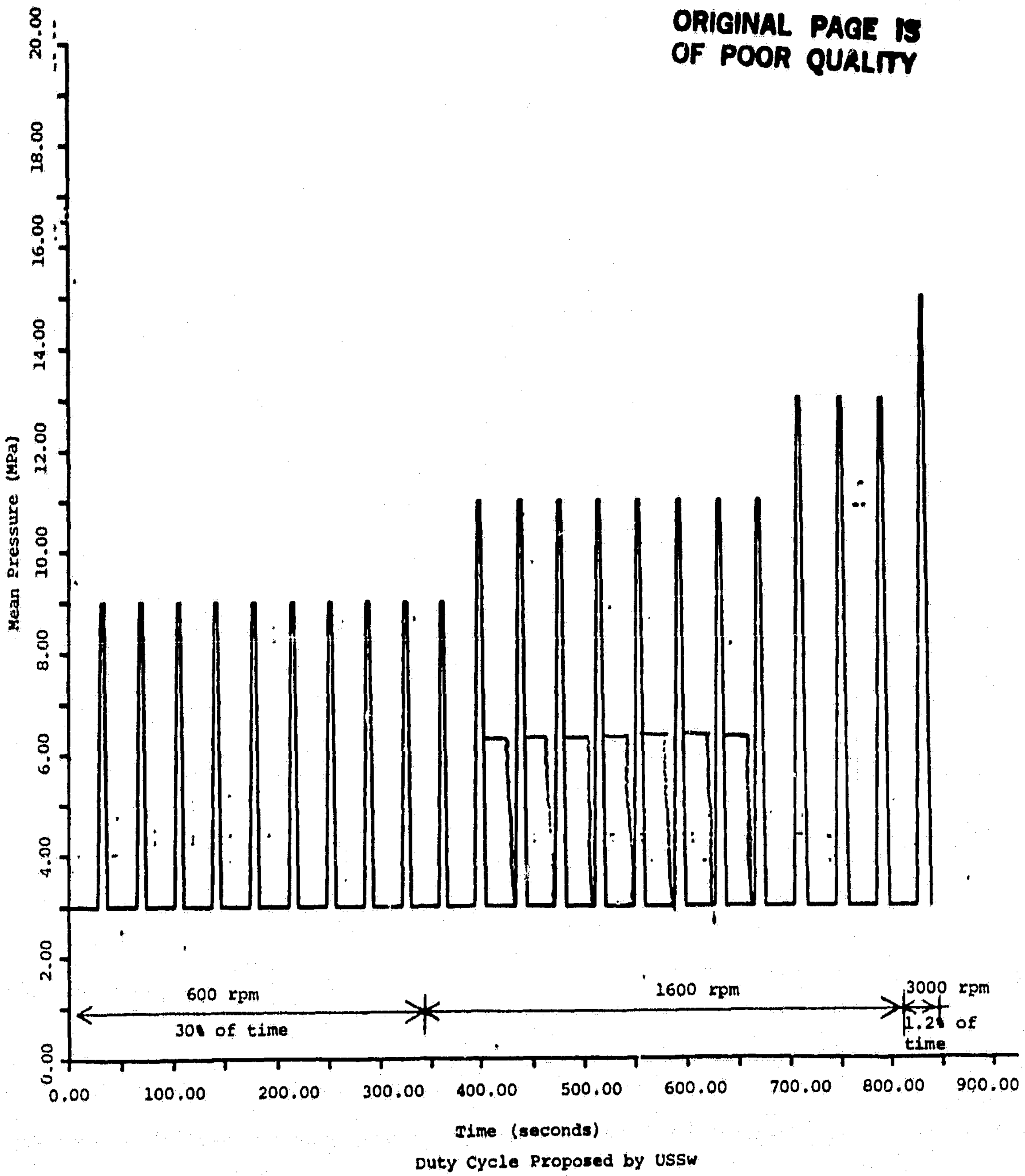

Figure B-1 Duty Cycle Proposed by USSw

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<u>Constant Mean Pressure/Constant Speed Cycles "Equivalent" to Driving Cycles</u>

Base "equivalence" on wear

Assume:

a) Wear α load on seal x sliding distance
b) Load on seal α mean engine pressure
c) Mean engine pressure α torque

Wear α torque x sliding distance

Wear rate α torque x speed

$\therefore$ Wear = wear rate x time α (torque x speed) x time

or <u>Wear α (torque x time) x speed</u>

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Table B-1. Mod II Computer Predictions - Urban Driving Cycle

Total Cycle Time = 1370 seconds

Percent of Cycle Time

Torque Range lb ft	Mean Torque lb ft	400-600	600-800	800-1000	1000-1200	Speed Range rpm
		500	700	900	1100	Mean Speed rpm (N)
0-10	5	0.00	0.00	0.53	0.01	
10-20	15	0.00	3.75	1.28	0.02	
20-30	25	5.45	32.02	1.38	0.30	
30-40	35	9.09	3.79	1.34	0.58	
40-50	45	1.83	2.83	1.34	0.67	
50-60	55	1.50	1.04	2.19	0.69	
60-70	65	0.66	0.81	3.31	0.64	
70-80	75	1.41	0.84	2.36	1.63	
80-90	85	1.34	0.36	0.12	2.50	
90-100	95	0.12	1.30	0.05	2.50	
100-110	105	0.00	0.74	0.02	1.17	
110-120	115	0.00	0.37	0.01	0.30	
120-130	125	0.00	0.00	0.00	0.26	
130-140	135	0.00	0.00	0.01	0.07	
140-150	145	0.00	0.00	0.00	0.22	
150-160	155	0.00	0.00	0.00	0.03	
Total % Driving Cycle Time		21.42	47.83	13.93	11.79	Σ = 94.97%
Σ Torque x Time lb ft sec x rpm		1.22×10^4	2.14×10^4	0.95×10^4	1.30×10^4	
N Σ Torque x Time lb ft sec x rpm		6.12×10^6	15×10^6	8.56×10^6	14.35×10^6	$\Sigma = 44.03\times10^6$

$$\text{Weighted Mean Speed} = \frac{[6.12\times10^6\times500+15\times10^6\times700+8.56\times10^6\times900+14.35\times\times10^6\times1100]}{44.03\times10^6} = 841 \text{ rpm}$$

For "Equivalent" wear with constant torque, $\bar{T}$, constant speed, N and time period = 1370 seconds

$$\bar{T} = \frac{44.03\times10^6}{1370\text{x}N}$$

For N = 850 rpm $\bar{T}$ = 37.8 lb ft

Under these conditions corresponding mean pressure = 700 psi (4.68 MPa)

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Table B-2 **Mod II Computer Predictions - Highway Driving Cycle**

Total Cycle Time = 766 seconds

Percent of Cycle Time

Torque Range lb ft	Mean Torque lb ft	600-800	800-1000	1000-1200	1200-1400	Speed Range rpm
		700	900	1100	1300	Mean Speed rpm (N)
0-10	5	0.00	2.73	1.27	0.00	
10-20	15	1.14	0.57	1.32	0.15	
20-30	25	2.03	0.93	2.40	0.22	
30-40	35	0.39	0.98	2.28	0.65	
40-50	45	0.13	1.57	4.70	2.34	
50-60	55	0.27	1.03	7.43	3.58	
60-70	65	0.65	1.79	6.80	8.88	
70-80	75	0.13	1.27	2.98	7.18	
80-90	85	0.25	1.48	4.20	3.05	
90-100	95	0.81	1.12	2.72	2.25	
100-110	105	1.18	0.52	1.52	1.32	
110-120	115	0.59	0.95	1.37	0.63	
120-130	125	0.13	0.69	1.09	0.39	
130-140	135	0.00	0.66	0.72	0.00	
140-150	145	0.00	0.29	0.39	0.00	
150-160	155	0.00	0.00	0.16	0.02	
Total % Driving Cycle Time		7.69	16.57	41.33	30.65	Σ = 94.24%
Σ Torque x Time lb ft sec x rpm		3.53×10^3	7.91×10^3	20.77×10^3	16.75×10^3	
N Σ Torque x Time lb ft sec x rpm		2.47×10^6	7.12×10^6	22.85×10^6	21.77×10^6	$\Sigma = 54.21\times10^6$

$$\text{Weighted Mean Speed} = \frac{[2.47\times10^6\times700+7.12\times10^6\times900+22.85\times10^6\times1100+21.77\times10^6\times1300]}{54.21\times10^6} = 1136 \text{ rpm}$$

For "Equivalent" wear with constant torque, $\bar{T}$, constant speed, N and time period = 766 seconds

$$\bar{T} = \frac{54.21\times10^6}{766\ N}$$

For N = 1100 rpm $\quad \bar{T}$ = 64.3 lb ft

Under these conditions corresponding mean pressure = 1050 psi (6.98 MPa)

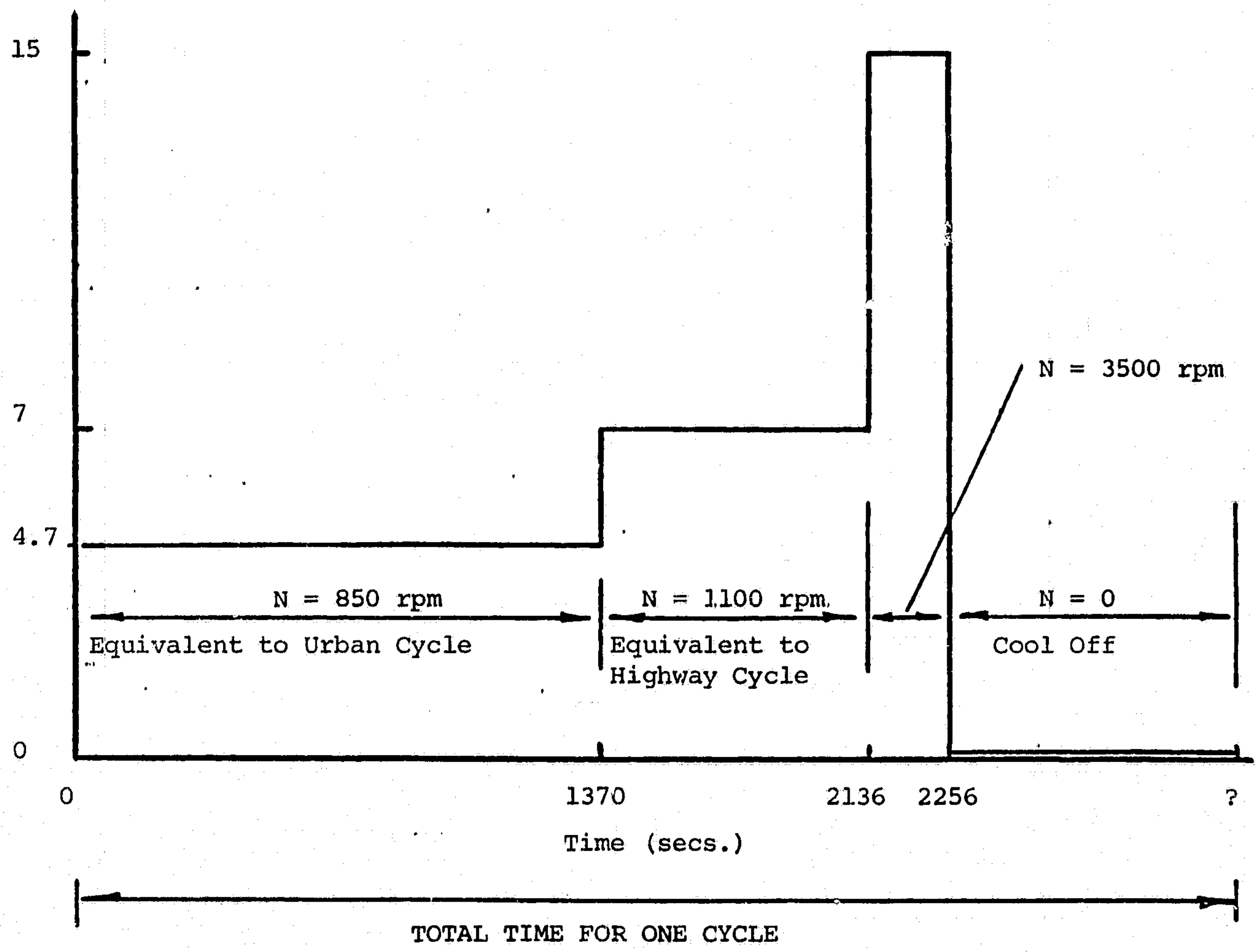

Figure B-2 Possible Duty Cycle for Seal Test Rigs Based on Analysis of Computer Predictions for Driving Cycles (Mod II)