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DEVELOPMENT AND DEMONSTRATION OF A STIRLING/RANKINE HEAT ACTIVATED HEAT PUMP

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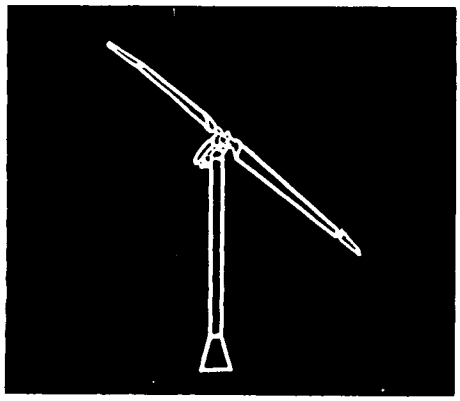
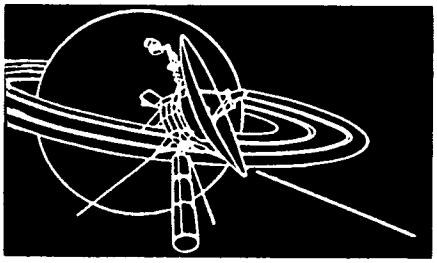
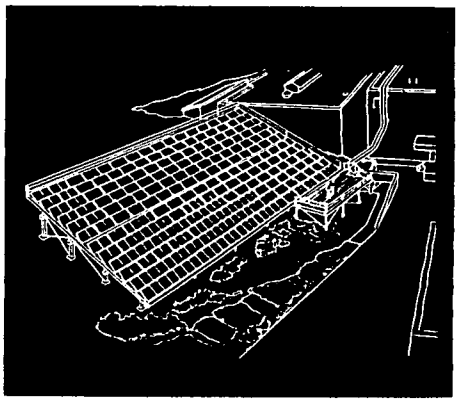
JULY THROUGH DECEMBER 1980

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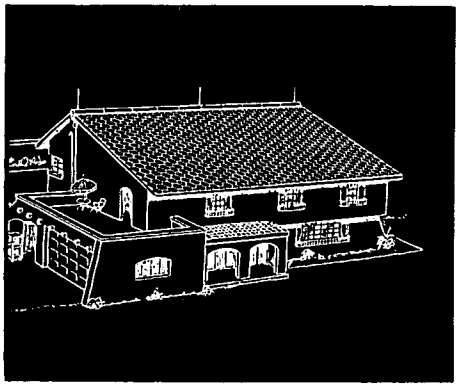
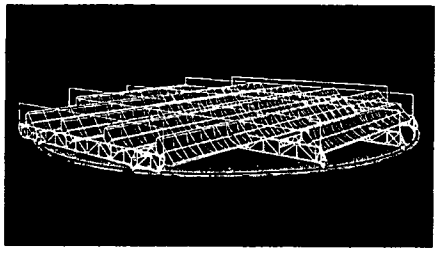
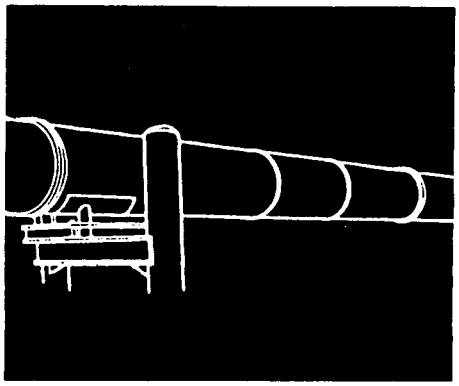

ENERGY SYSTEMS AND TECHNOLOGY DIVISION

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Valley Forge Space Center
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SECTION 1

INTRODUCTION

SECTION 1
INTRODUCTION

1.1 BACKGROUND

In January 1975, General Electric embarked with the American Gas Association (AGA) on a multi-phase program for the development of a Gas Heat Pump (GHP). The Department of Energy (DOE) joined General Electric and the AGA in the program in 1976, and the Gas Research Institute assumed responsibility for the program from the AGA in 1977.

Phase 1 of the program was completed in December 1975, and Phase II was initiated in February 1976.

1.1.1 PHASE I RESULTS

The Phase I program followed the task structure shown in Figure 1-1. The primary objective of this program was to evaluate, through an analytical effort, if a gas heat pump product could evolve into a viable business venture. A comparison of the possible heat activated heat pump concepts showed that the Stirling/ Rankine and the absorption cycle were the most promising from the standpoint of performance (potential operating savings), region of application, and market application (residential and/ or commercial); the product design was performed in sufficient detail to show that the variation in manufacturing costs of the candidate systems would not significantly affect payback considerations. Further refinement of the design, cost, and performance for the two candidate systems and a definition of product introduction scenario and pricing strategies allowed a business evaluation of the two concepts to be completed. The results of these efforts were reviewed within GE and with AGA and gas industry representatives. As a result of this Phase I effort, the Stirling/ Rankine concept was selected as the most promising approach and a decision was made to proceed with the Prototype Development, Phase II.

1.1.2 PHASE II OBJECTIVES

The objective of the Phase II program is to demonstrate that the performance estimates generated during Phase I can be realized and that cost goals are realistic. The Phase II task flow diagram is shown in Figure 1-2.

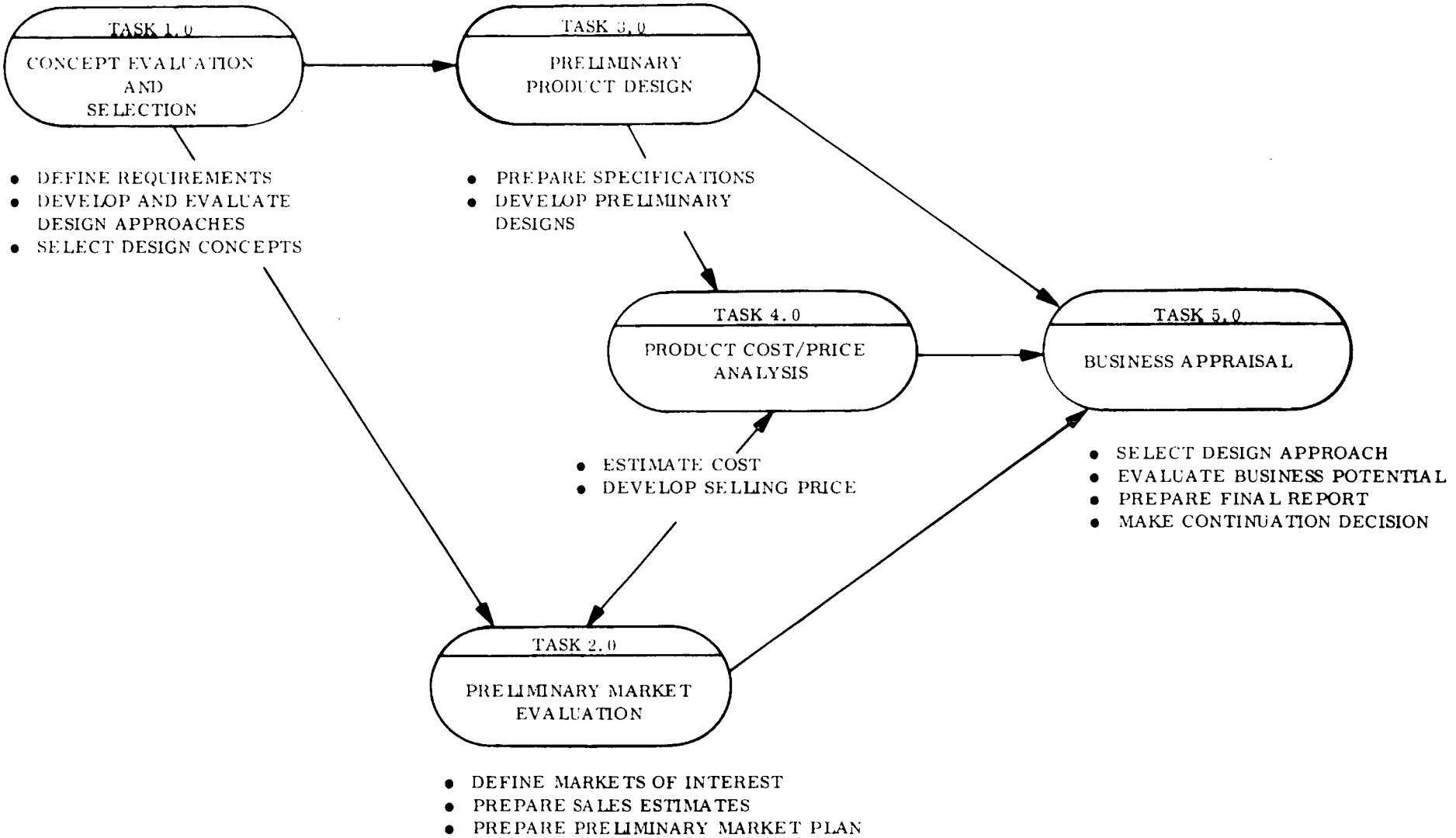


Figure 1-1. Phase I Flow Diagram

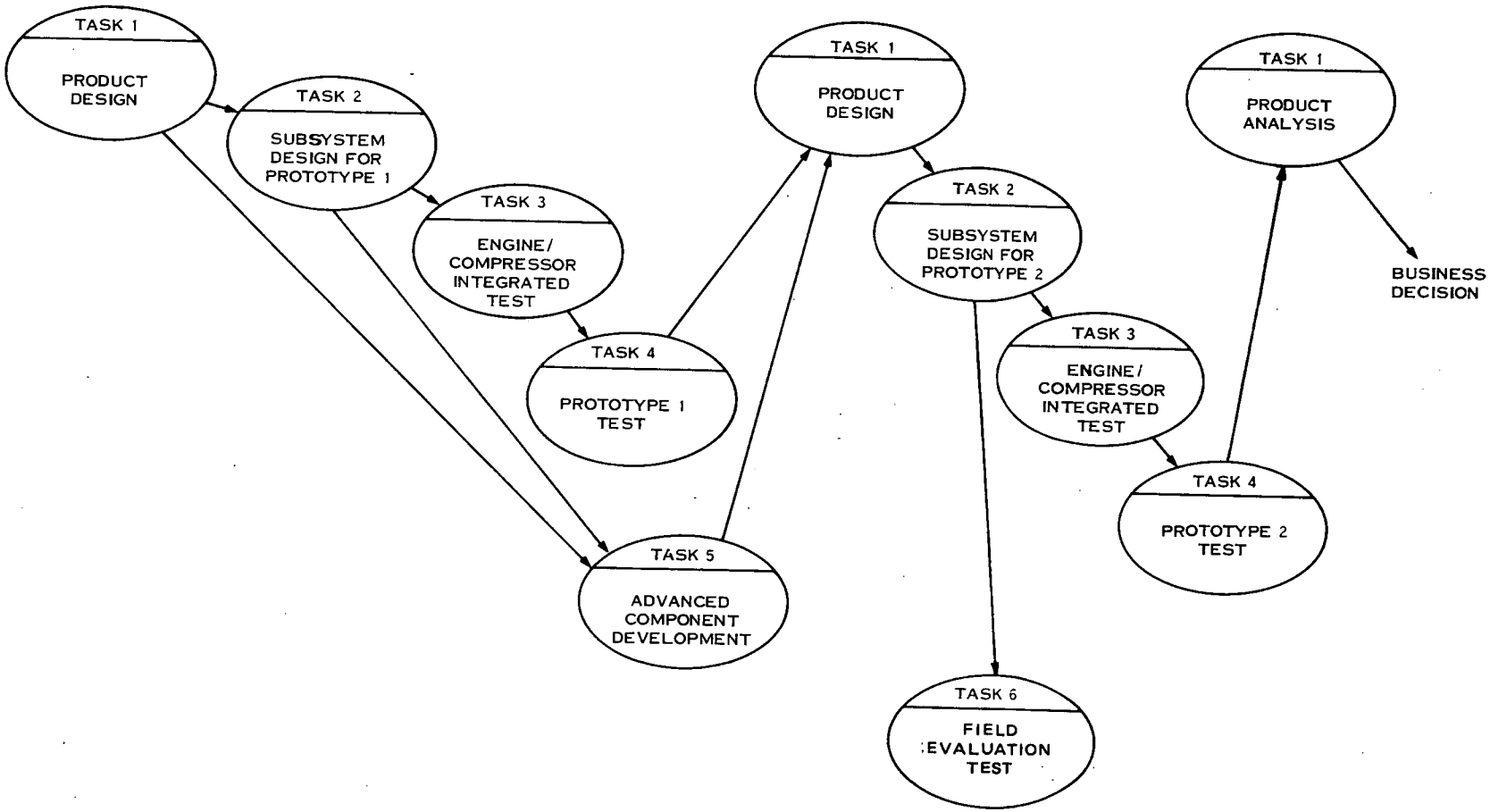


Figure 1-2. Phase II Flow Diagram

Two generations of prototype hardware will be developed during the Phase II program. Prototype 1 was the first attempt to test a complete Stirling/Rankine HAHP system with heat pump configured hardware. This system was comprised of hardware of a primarily developmental nature. The Prototype 2 system consists of more mature hardware, incorporating the results of the Prototype 1 development effort and designed toward the eventual product.

1.1.3 PHASE II PRIOR ACCOMPLISHMENTS

The gas heat pump can be described in terms of five major subsystems: the natural gas combustor, the Stirling engine, the refrigerant compressor, the controls, and the air-handlers.

The Prototype 2 HAHP consists of a 3-ton, split system targeted to provide approximately 64,000 BTU/hr in the heating mode at a nominal 47°F ambient temperature. The overall product performance goal was to achieve Coefficients of Performance (COP) of 1.6 and 0.85 for the heating and cooling modes, respectively (excluding the electrical power requirements). The electrical power required for fans, blowers and pumps was set at less than 1100 watts. Based on these performance levels, the following performance goals were assigned to the major subsystems:

- Stirling Engine Efficiency: 32%
- Refrigeration Loop COP: 3.5
- Overall Combustion Efficiency: 80%

The following sections summarize the accomplishments during the previous Phase II reporting period for the system design and each of the subsystem areas.

<u>Product Design</u>

During the previous reporting period the Product Design effort focused on: analytical tasks related to the data reduction and interpretation of integrated test results for the Proto-2 engine/compressor; parametric studies of engine modifications to improve performance;

and, refinements and improvements to the analytical design tools used for engine design and performance prediction.

Analysis of Proto-2 integrated test results indicated that mid-stroke venting of the engine second gas spring, required to permit successful starting of the engine, caused excessive power dissipation under full operational conditions. Tests with a variety of engine second gas spring volumes, and two different sized second gas spring piston areas, yielded minor improvements in second gas spring power dissipation. Analysis indicated that slight modifications to the second gas spring piston could reduce losses. The modifications tested were successful, resulting in a 30% reduction in power dissipation. Performance and parametric tests were run on a variety of engine configurations, and the engine configuration was tuned to the maximum power and efficiency configuration within the current Proto-2 design constraints.

Further improvements in performance were predicted through design of the engine heat exchangers and dynamic components. Analysis indicated that a higher effectiveness heater heads and cooler, and higher regenerator effectiveness with lower flow losses, could improve engine power output and efficiency substantially.

Work on engine/compressor simulation tools continued in the previous reporting period, utilizing the results of initial Proto-2 tests to correlate test data to the analytical models. In addition, supplementary tools were developed and/or modified to assist in the engine redesign. The thermodynamics simulation program (TDP) was correlated to test data from the Proto-2 engine/compressor, and refined and improved to reduce predictive errors. A faster convergence routine for the TDP was developed, and overall predictive accuracy of the model was improved in all major parameters except phase angle, which consistently overpredicted the actual test results.

A supplementary dynamics code, based on the Schmidt analysis, was developed and used for the parametric studies leading to the recommended modifications to the Proto-2 engine. This analysis was augmented by a second order model, adopted from the Martini design

analysis. Results of these analyses agreed with both the TDP and test data closely enough to permit their use in the parametric study leading to the recommended design changes.

Combustor

During the previous reporting period, combustor testing was initiated in conjunction with the engine/compressor integrated test. Due to engine performance deficiencies, the combustor experienced excessive thermal soaking, leading to higher than design operating temperatures and subsequent flash-back. This condition, caused by the engine, was temporarily alleviated for testing purposes by removing a portion of the insulation normally used on the outer casing of the combustor, to reduce the level of air preheat. The flash-back phenomenon was relieved, with the addition of minor internal changes to the combustor flow pattern, and testing proceeded without further incident.

Continued testing of the combustor burner element with the Nextel ceramic sock indicated that this type of protective ceramic woven material was not suitable for use with the HAHP combustor element. Due to the physical configuration of the combustor element, the operating conditions, and material incompatabilities between the Nextel and the Hoskins 875 element material, unsatisfactory performance of the burner element/Nextel sock was experienced. Alternative materials and application techniques were, however, identified, and examined in the current reporting period.

Engine

A number of engine configurations were assembled and tested as part of the integrated engine/compressor tests in the previous reporting period. Engines with different regenerator porosities, displacer rod areas, displacer gas spring volumes, displacer weights, second gas spring sizes, second gas spring volumes, and second gas spring vent patterns were tested to examine their effect on overall engine performance.

The magnetic starter, developed in a previous reporting period, was further modified in an attempt to increase its motion during start-up under a variety of initial conditions. The modifications were of limited success, and the effort in this area was concluded. The alter-

native starting approaches (engine housing impulse, and external pulse generator) were also examined. The engine housing impulse technique was attempted under a variety of conditions and successfully started the engine, but an evaluation of this technique indicated that it would be difficult to control and to implement with the current configuration. However, development of the external pulse generator technique was very successful, providing engine starts under a wide variety of initial conditions. This technique was therefore selected for further development and testing in the unit at the test site.

Bearing and seal development also continued in the previous reporting period. All seal and bearing hardware sample fabrication work was completed, and modifications to the wear test stand were successfully concluded. At the end of the reporting period, testing was ready to be initiated.

Development of the transient load acceptance subsystem for the engine continued in the previous reporting period. The position sensors selected for this subsystem were received and tested successfully. The control circuit for the position sensors/ valve of the transient load acceptance unit was also tested successfully.

Compressor

Tests of the Proto-2 compressor on the new test stand at the design conditions were initiated in the previous reporting period. Testing was conducted in excess of the 3 Kw design input. Off center operation of the compressor piston was detected, and changes to the suction/ discharge flow passages and plumbing arrangements were successful in correcting this condition. In addition, excess gas spring leakage in the compressor was determined to be causing significant reduction in refrigerant flow, and was corrected.

Compressor tests in the previous reporting period included tests with the compressor utilizing two of the three compressor regulators developed for this application. Testing of the Singer regulator (adapted for test stand use) provided suitable regulation for component tests. The Carlton regulator was received during the previous reporting period, checked out and mounted on the compressor in the test stand and successfully tested. The Ausco regulator was received near the end of the reporting period and was not tested.

Testing of the compressor in conjunction with the engine in the integrated test stand was limited due to engine power deficiencies. The compressor was utilized primarily as a load device for initial engine testing and for diagnostic tests of the effects of varying engine parameters. Full performance engine/compressor testing was planned for the current reporting period.

Air Handlers and Controls

Testing of the first air handler unit was completed in the previous reporting period. A calibrated electric compressor was used in conjunction with the three ton environmental simulation chamber to: optimize the refrigerant charge; determine rating point capacities; test the unit at off design conditions; determine overall unit parasitic power; and, conduct defrost tests. The number 1 air handler unit met all of its requirements in the tests, and required only 875 watts of parasitic power, vs. the 1100 watt goal. The time to defrost was determined to be 3 minutes, well within normal heat pump requirements.

Approximately 90% of the second outdoor unit assembly, field evaluation test unit, was also completed in the previous reporting period.

Integrated Tests

During the previous reporting period, integrated testing of the Proto-2 engine/compressor unit was initiated. The unit started on the first attempt, a significant achievement for a new engine design. During the course of the reporting period, over 40 different engine/compressor configurations were tested, yielding a significant body of test data for analytical tool correlation and diagnostic interpretation of performance and operation. Results of the tests were used to guide the redesign of the Proto-2 engine, aimed at increasing its operating efficiency and output power to the point where the unit would be ready to be installed in the field test unit.

Integrated testing was the primary tool used to diagnose operation of the engine second gas spring and its effect on engine start-up and performance. Through a series of progressive tests, various mid-stroke venting configurations for the second gas spring were utilized to

permit free motion of the engine power piston at its mid-stroke position (during start-up). The high dissipation power incurred by the second gas spring was also measured and diagnosed through integrated tests, leading to recommendations for modifications to the second gas spring design to reduce these to acceptable levels.

Field Evaluation Tests

During the previous reporting period the field evaluation test activity included selecting, ordering, and receipt of the major components for the test site instrumentation rack. Preparation of fabrication and assembly layout and sketches continued, and sub-metering of the gas and electric energy consumption by the existing test site equipment was also accomplished. In April, work on the field evaluation test activity was rescheduled to accommodate the modification of the Proto-2 engine. Continuity of effort was maintained by completion of the initial instrumentation rack components and by establishing periodic coordination meetings with the site's sponsor, Philadelphia Electric, for review of submeter data and program schedule adjustments.

1.2 PROGRESS SUMMARY

Program effort during the 10th semi-annual period of Phase II (the current reporting period) continued to concentrate on integrated testing of the engine/compressor assembly in the integrated test stand. Design fabrication and assembly of the modified engine was completed, and initial check-out testing of the new configuration was initiated. Development work on the combustor burner element, the engine bearings and seals, the engine overstroke protection subsystem, and the auxillary heating unit design were continued. In addition, improvements and refinements to the TDP simulation tool were continued during this reporting period. Table 1-1 summarizes the major accomplishments during the June to December 1980 reporting period. Detailed descriptions of the program progress on each subsystem, the overall product design, and related program areas are presented in the following sections of this report.

Table 1-1. Prototype 2 HAHP System Development Status

System/ Substem	Status
System	• Baseline Engine Integrated Test Data Analyzed • Simulation Tool Accuracy Improved
Combustor	• Off-Design Operation Sensitivity Defined • Burner Element Coatings Tested
Engine	• Modified Proto 2 Engine Fabricated • Modified Engine Integrated Tests Initiated • Bearing and Seal Endurance Tests Continued
Compressor	• Unit 3 Compressor Tests Completed • Performance Goals Essentially Met
Air Handlers & Controls	• Auxiliary Heater Concept Selected • Preliminary Design Prepared
Field Test	• Test Activities Rescheduled • Submetering Continuing

SECTION 2

PRODUCT DESIGN

SECTION 2

PRODUCT DESIGN

2.1 INTRODUCTION

Integrated testing of the combined engine/compressor/combustor assembly for the baseline Proto 2 HAHP was continued this reporting period. The main objective of these tests was to improve power output and to lower the second gas spring power dissipation. A significant effort was devoted to analyzing and reducing the test data, from these and past tests. This included optimization of, and improvements to, the TDP computer code.

2.2 ENGINE/COMPRESSOR/COMBUSTOR PERFORMANCE

2.2.1 PERFORMANCE CHARACTERISTICS

Performance tests run in the July-December 1980 period are shown in Figure 2-1, in terms of indicated power and efficiency.

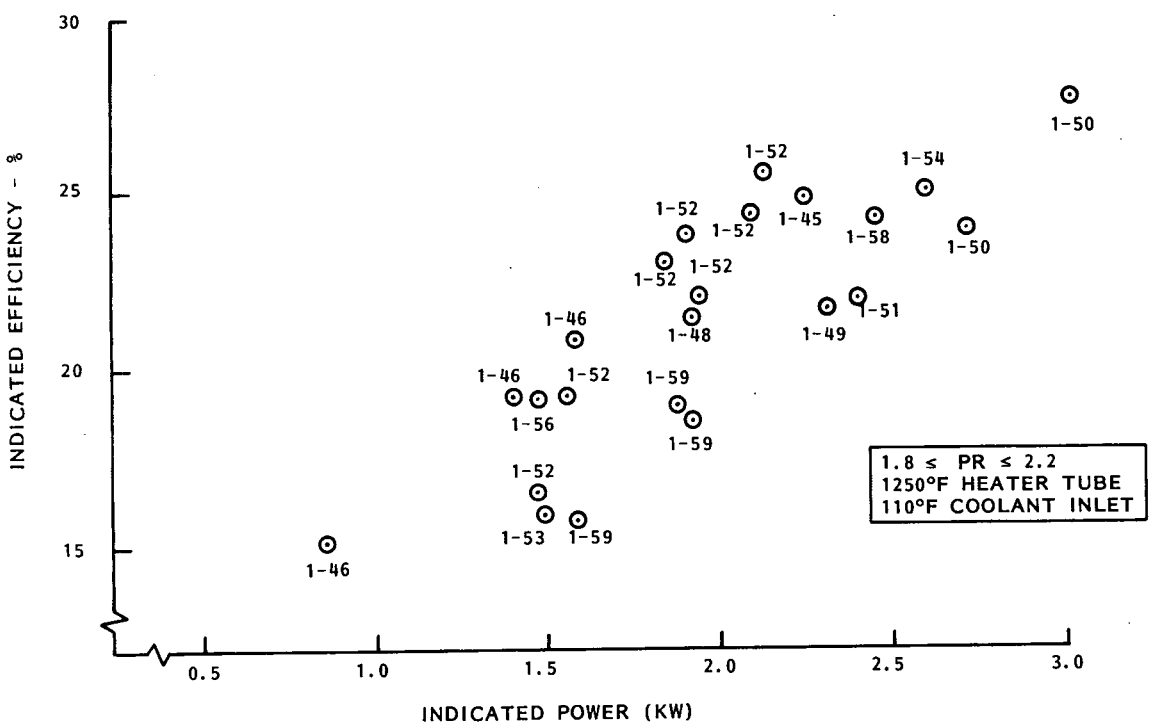


Figure 2-1. Proto 2 Engine Performance Tests, 7/80-12/80

Performance of principal Proto 2 configurations to date is shown in Figure 2-2. Indicated performance differs from shaft output by the second gas spring power dissipation and by other, minor, losses. The data in Figures 2-1 and 2-2 have been normalized to design-point heater tube and coolant inlet temperatures. Details of the configuration, and measured performance values (prior to temperature correction), are shown on Table 2-1.

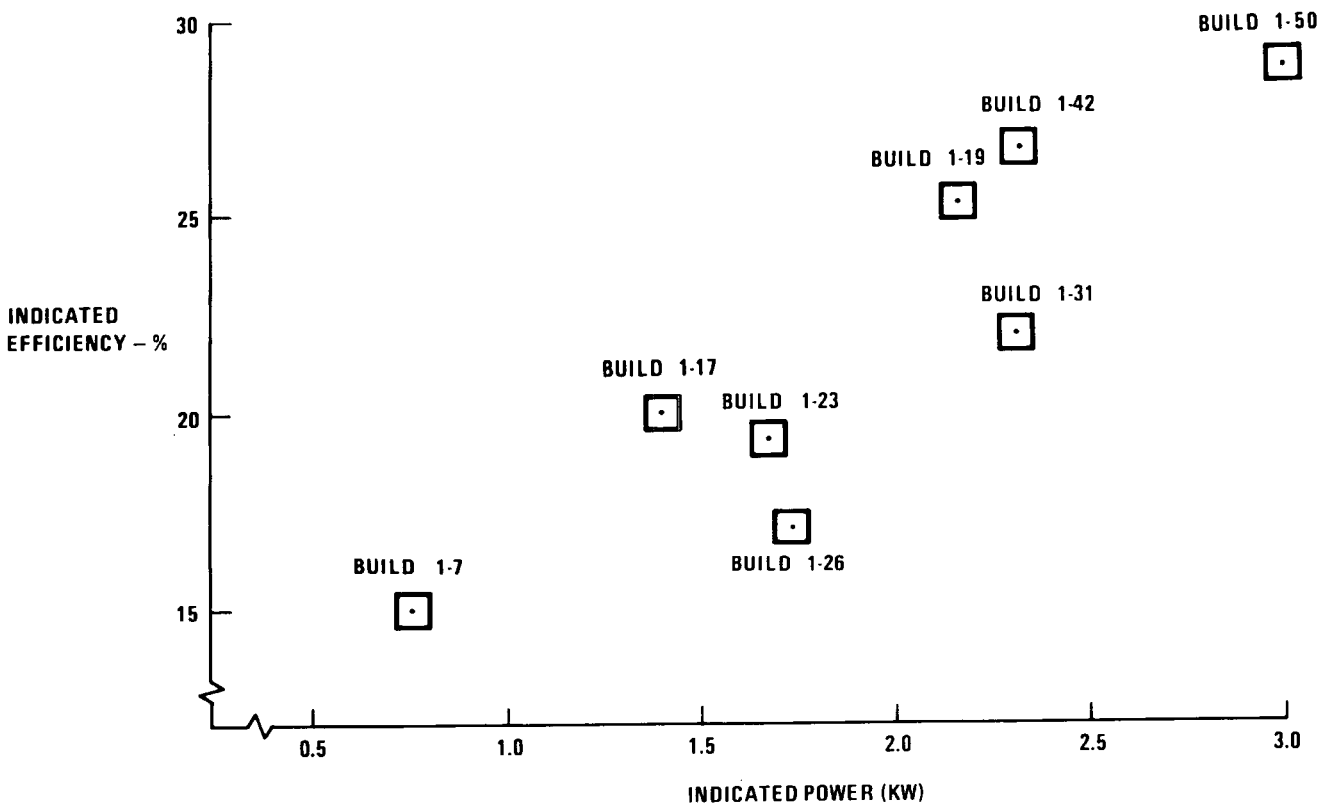


Figure 2-2. Performance of Principal Proto 2 Engine Configurations

The test results in Figures 2-1 and 2-2 show that an indicated power near 3 Kw and an indicated efficiency over 27 percent were achieved on one configuration, and a number of tuned configurations achieved high levels of indicated power and efficiency. However, the test results indicate that inherent limitations of the engine design will require the engine modification selected to achieve the Proto 2 performance goals.

Table 2-1. Test Details for Selected Configurations

Build No.:	1-45	1-50	1-52	1-54	1-56	1-58	1-59
Date:	7-8-80	7-17-80	7-22-80	7-29-80	8-1-80	8-13-80	8-14-80
Time:	15:14	9:32	14:49	11:09	9:42	9:08	10:00
Data Point:	#1	#3	#7	#1	#2	#1	#1
Description:							
Regen. Porosity (%)	80	80	80	80	80	80	80
Second Gas Spring Area (in.^{2})	11.6	11.6	11.6	11.6	11.6	11.6	11.6
Second Gas Spring Vent Pattern	2 x 54 @ ± 1/8" & 2 x 18 @ 1/4"	2 x 54 @ ± 1/8" & 2 x 18 @ 0	Same	Same	2 x 18 @ ± 1/8" & 2 x 18 @ 0	2 x 36 @ ± 1/8" & 2 x 18 @ 0	
Displacer Weight (lb.)	1.76	1.76	1.83	1.83	1.83	1.83	1.83
Displacer Gas Spring Vol (in.^{3})	122.5	122.5	81	81	81	81	81
Rod Group	G-3	G-4	G-4	G-4	G-4	G-4	G-4
Compressor Unit No.	2	2	2	2	2	2	2
Q Into Engine (kBtu/hr)	31.5	37.3	~29.0	35.0	27.1	34.7	35.4
Heater Tube Temp. (^{o}F)	1144	1147	1218	1157	1240	1227	1260
Charge Pressure (psia)	880	1005	885	880	930	885	885
Compressor Suction (psia)	108	99	99	90	114	90	91
Compressor Discharge (psia)	201	214	193	196	270	193	195
Pressure Ratio	1.86	2.16	1.95	2.18	2.37	2.14	2.14
Operating Frequency (Hz)	26.0	28.6	27.3	27.7	29.6	27.8	27.7
Piston Stroke/Housing (in.)	1.27	1.12	1.14	1.27	1.26	1.20	1.20
Piston Stroke/Ground (in.)	-	0.99	-	1.12	-	1.05	1.05
Displacer Stroke/Housing (in.)	-	1.62	1.62	1.75	1.60	1.60	1.54
Phase Angle (degrees)	-	53.1	46.0	46.2	-	47	42
Refrigerant Flow (lb/hr)	324	424	328	314	193	377	325
Indicated Power (kW) Fourier Analysis	-	2.23	-	2.08	-	1.72	-
Energy Balance Analysis	2.27	3.02	2.17	2.60	1.54	2.74	2.33
Indicated Efficiency Fourier Analysis	-	20.4	-	20.3	-	-	-
Energy Balance Analysis	24.6	27.6	25.5	25.3	19.4	26.9	22.5
Compressor Power (kW) $\dot{m}\Delta h$ at Pressure Vessel	1.70	2.36	1.64	1.98	0.92	2.15	1.75
Second Gas Spring Power (kW) - Fourier Anal.	-	0.77	-	0.61	-	0.64	-
Avg. Coolant Temp. (^{o}F)	71.7	76.0	~72	76.0	73.5	75.1	72.1

Testing during the January-June 1980 reporting period also indicated that the second gas spring dissipated much more power than had been anticipated. A thorough analysis of the second gas spring configuration in this period yielded three promising approaches to reducing dissipation:

1. Chamfering of piston corners as an aerodynamic enhancement to reduce dissipation during venting.
2. Slotting of the liner between vent holes.
3. Increased piston ring overlap, to decrease leakage.

Due to the constraints imposed by the current hardware design (Figure 2-3), only the first approach was implemented. The results are encouraging. As shown in Figure 2-4, dissipation was decreased up to 18%. However, power dissipation is still high (approaching 1 kW at design stroke and frequency), and additional improvement is planned for this area.

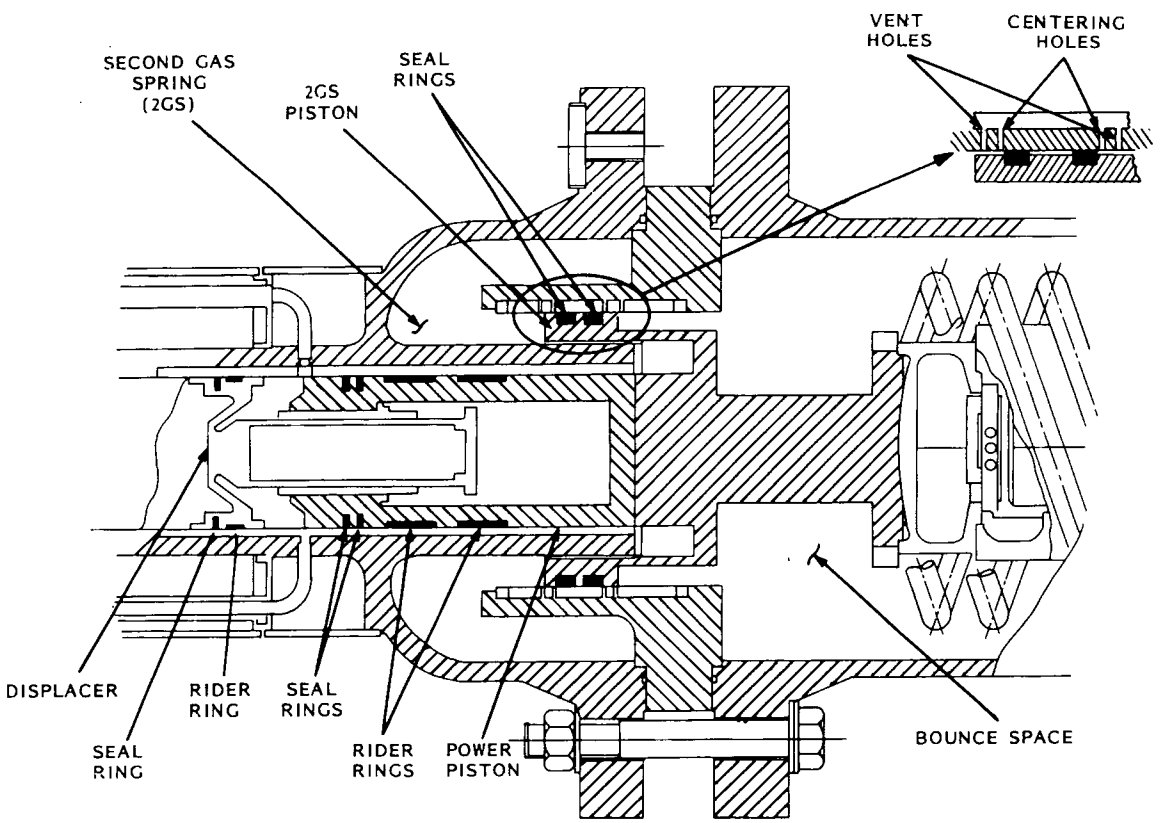


Figure 2-3. Proto 2 Second Gas Spring Design

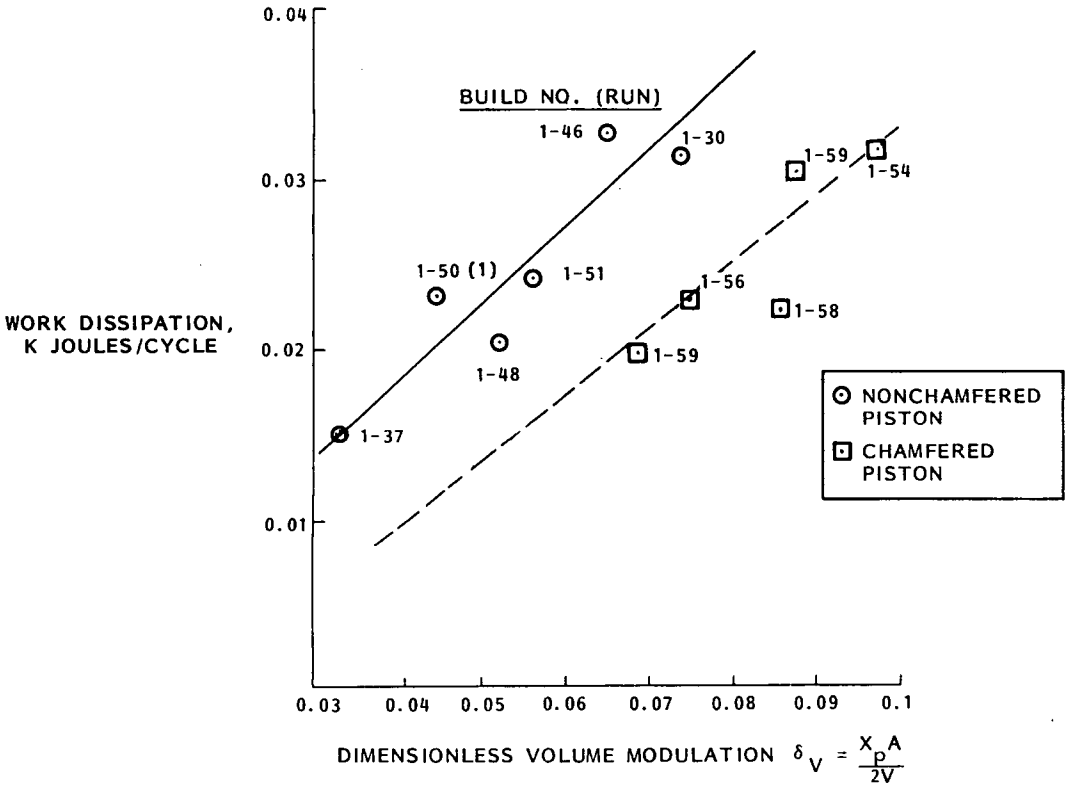


Figure 2-4. Second Gas Spring Work Dissipation

Data Reduction

As discussed in the previous semi-annual report, some unresolved differences were evident in the two main approaches used to calculate indicated power, the Energy Balance technique and the Fourier Analysis technique. Development of a third method was initiated in this period, utilizing compressor power based on oscilloscope traces of compressor pressures versus power piston motion (position). Integrating and summing the areas yields the power transmitted to the compressor. Preliminary results from this approach are in good agreement with those from the energy balance method (discussed in the last semi-annual). This technique appears to give consistent results, and allows isolation of engine performance from compressor performance. Also, fewer assumptions (regarding heat transfer and frictional losses) are required in this technique. However, its accuracy is very sensitive to the accuracy of dynamic data measurements, with signal phasing of critical importance. A complete evaluation of the data reduction techniques will be conducted in the next reporting period.

2.3 TDP SIMULATION

Continued use has been made of the Stirling engine/linear compressor computer simulation program. The Thermodynamic Program (TDP) is the key tool for simulating engine operation. It is a detailed code that accounts for both the dynamic and thermodynamic behavior of the free piston Stirling engine and compressor. A cleaned-up and streamlined version of the TDP code (NEWTDP) has been created, and is fully operational. It reduces run costs by 28%, and facilitates future code improvements and changes.

In the past, the correlation of test data with the TDP simulation has been relatively good, but exhibited three major deficiencies:

1. Working gas leakage and gas-spring operation simulations were oversimplified.
2. A number of assumptions and/or configuration-specific correction factors were required to achieve reasonable correlation.
3. Some unexplained correlation anomalies were obtained.

A number of significant code changes have been made in this regard. The more important improvements are:

1. The second gas spring model has been updated and incorporated into TDP.
2. A refined compressor model was developed and inserted into the TDP as a subroutine. The flow rate predicted compares favorably with test results:
3. The displacer gas spring model has been modified and adjusted to match test data.
4. All significant leakage paths have been modelled.

The baseline engine configuration selected for code verification is Build 1-58, which provided good performance and relatively centered, stable operation. Performance predictions of TDP, and the newer NEWTDP, are correlated against test data in Figures 2-5 and 2-6. The accuracy of the new correlation (for Build 1-58, as well as 1-19) is within the range of measurement accuracies. Close matching of the dynamic parameters was achieved. The

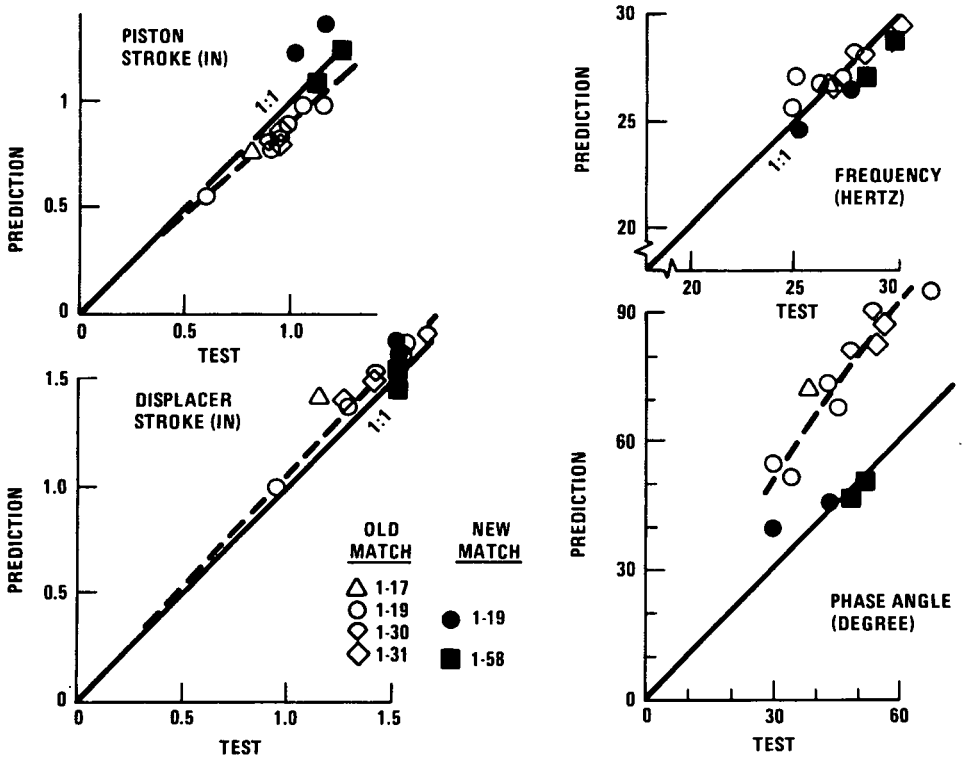


Figure 2-5. TDP Model Correlation with Dynamic Test Data

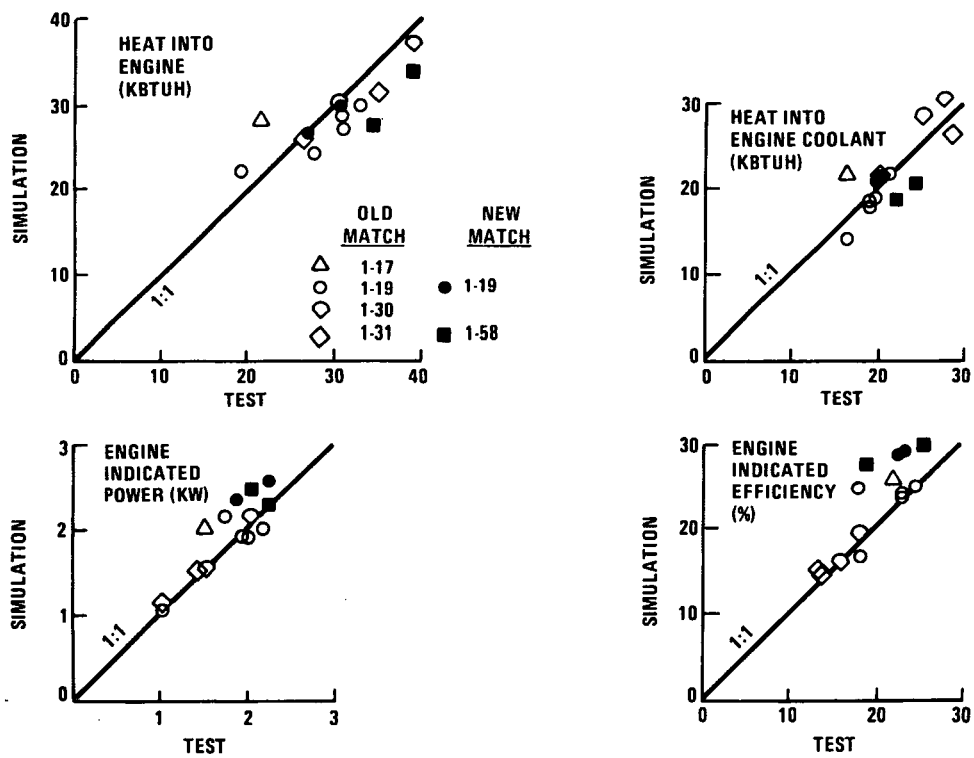


Figure 2-6. TDP Model Correlation with Thermal Test Data

excellent agreement in the displacer/piston phase angle correlation is a major improvement. But thermal parameter matching was off more than is desirable. Several further improvements and refinements to the TDP simulation were identified for future development.

2.4 PLANNED ACTIVITY FOR NEXT PERIOD

Activity in the next reporting period will concentrate on:

1. Reduction and interpretation of tests on the modified Proto 2 engine design.
2. Refinements in data reduction methods.
3. Removal of all empirical adjustment factors and improvement of individual component models within the TDP code.
4. A comparison of TDP results with those of another code, from Sunpower, Inc., is planned to verify the overall validity of TDP.

SECTION 3

PRODUCT DEVELOPMENT

SECTION 3

PRODUCT DEVELOPMENT

3.1 INTRODUCTION

During this reporting period, principal product development activities were directed at component tests supporting Proto 2 engine/compressor integrated tests. An evaluation of combustor off design performance and its sensitivity to flashback was conducted to determine what temporary modifications to the combustor would result in satisfactory operation with the engine operating off-design. In addition, refractory and silicate coatings for the burner element were identified and tested to reduce excessive oxidation rates on the combustor surface. The modifications selected to improve engine performance in the last reporting period were executed this reporting period, with check-out tests started by the end of the period. Bearing and seal endurance improvements were also continued this period. Tests of the Unit 3 compressor were completed this period. This unit met essentially all functional and performance requirements for the compressor.

In addition to component development in support of Proto 2 Integrated Testing, an auxiliary subsystem providing back-up heating capability at low ambient conditions was selected and designed. The requirements of the subsystem were defined, a number of optional approaches evaluated, and a preliminary design of the selected concept was prepared.

3.2 COMBUSTOR

3.2.1 SUMMARY

Combustor performance verification during this reporting period continued in support of the integrated engine/compressor test. The pre-ignition phenomenon observed and reported earlier was further analyzed during this period. Efforts to enhance the life of the burner element also continued, and initial results obtained with a new ceramic coating development technique were encouraging.

3.2.2 COMBUSTOR PERFORMANCE

In order to minimize the heat losses to the ambient, the outer surface of the combustor is surrounded by fiberous ceramic insulation. The insulation improves combustor efficiency and minimizes the outer surface temperature. During integrated tests higher than design values of preheater air temperature was being experienced due to engine performance deficiencies. When the preheater air temperature exceeded approximately 900°F, pre-ignition occurred inside the burner element. This condition worsens when the engine is running at off-design condition.

In order to expedite integrated testing, it became necessary to remove most of the insulation from around the preheater section such that the preheat air temperature would remain within the design value of 900°F. There is an optimum insulation level where the temperature of the air/fuel mixture does not approach ignition level while at the same time keeping the combustor efficiency high. Therefore, an analysis was conducted to determine the sensitivity of preheater air temperature to engine performance.

Sensitivity of Pre-Heat Temperature to Engine Performance

To prevent pre-ignition, the preheat temperature of air-fuel mixture should not exceed approximately 900°F (as measured at the location of distributor vanes). A correlation of the experimental data between the preheat temperature and the exhaust gas temperature is shown in Figure 3-1. For a 900°F preheat temperature level, the corresponding exhaust gas temperature is approximately 675°F.

If the engine absorbs less than the design condition combustor heat, the exhaust gas temperature and preheat temperature will increase. Under these circumstances, although the surface heat losses will increase slightly most of the additional heat (80%) must be rejected in the exhaust gases. Figure 3-2 shows the effect of change in the amount of energy removed by the engine on the preheat temperature. As shown, preheat temperature is sensitive to changes in engine heat input.

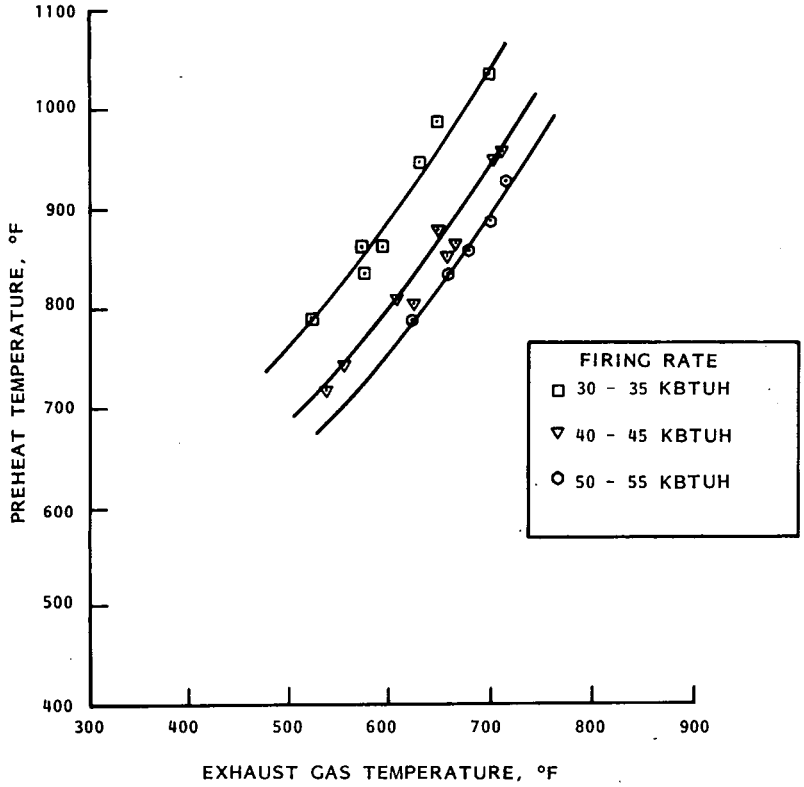


Figure 3-1. Correlation of Proto 2 Combustor Pre-Heat Temperature and Exhaust Gas Temperature

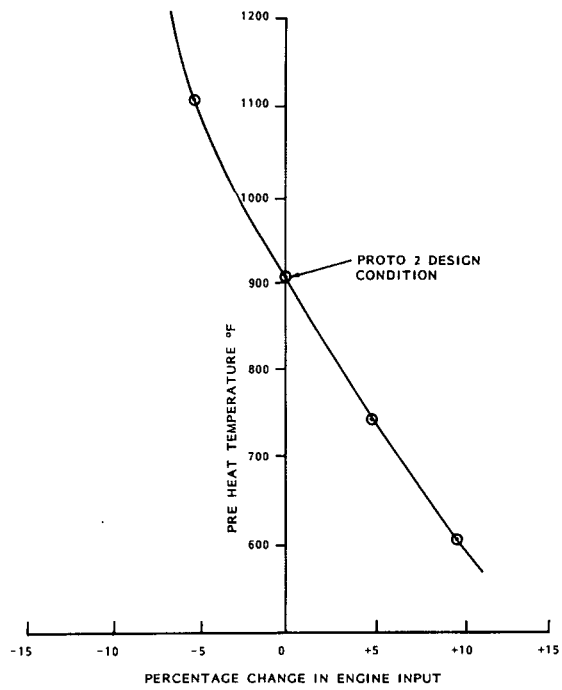


Figure 3-2. Sensitivity of Preheat Temperature to Engine Heat Input

Methods to reduce the sensitivity of preheat temperature to the amount of engine heat removed have been identified. One technique would be to closely identify the possible deviation in engine heat input level for a given design condition and then optimize the insulation level taking into account this deviation. This would assure that the preheat temperature will remain within limits when engine operates as prescribed.

3.2.3 BURNER ELEMENT COATING DEVELOPMENT

The Proto 2 combustor burner element is made of chopped, pressed and sintered Hoskins 875 wire with an 80% porous wall structure. The outside temperature of the burner element during operation is in the range of 2200-2600°F. This leads to deterioration of the element due to oxidation.

As reported previously, component tests conducted with a Nextel ceramic sleeve surrounding the burner element indicated a reduction in the operating temperature of the burner. However, when the burner element was fired on the integrated test stand, several incidences of flash back occurred and visual observation indicated embrittlement of the Nextel sleeve and consequent flaking. Since the material is rated up to 2500°F, it was apparent that in the combustor application the temperature of the sleeve would exceed 2500°F, although the felt metal temperature may still be lower than when operated without the ceramic sleeve. An additional problem is that it would be difficult to accomplish a tight fit between the Nextel sleeve and the burner element due to surface irregularities and thermal distortions. Gaps could jeopardize stable burner operation.

The Nextel test results led to consideration of alternate possibilities of enhancing the burner life. One such possibility would be direct application of a ceramic coating to the burner element material.

Material Selection and Evaluation

The ceramic coatings examined consist of powdered refractory glass, clay and water ("refractory") or a mixture of water and an aqueous alkali silicate ("silicate").

The coatings were first applied to the following materials:

1. Felt metal mat approximately 0.125" thick consisting of 0.0056" diameter Hoskins 875 material fibers.

2. 40 mesh wire cloth, constructed of 0.010" diameter Incoloy 800 material.

3. Braided Nextel ceramic fiber cloth.

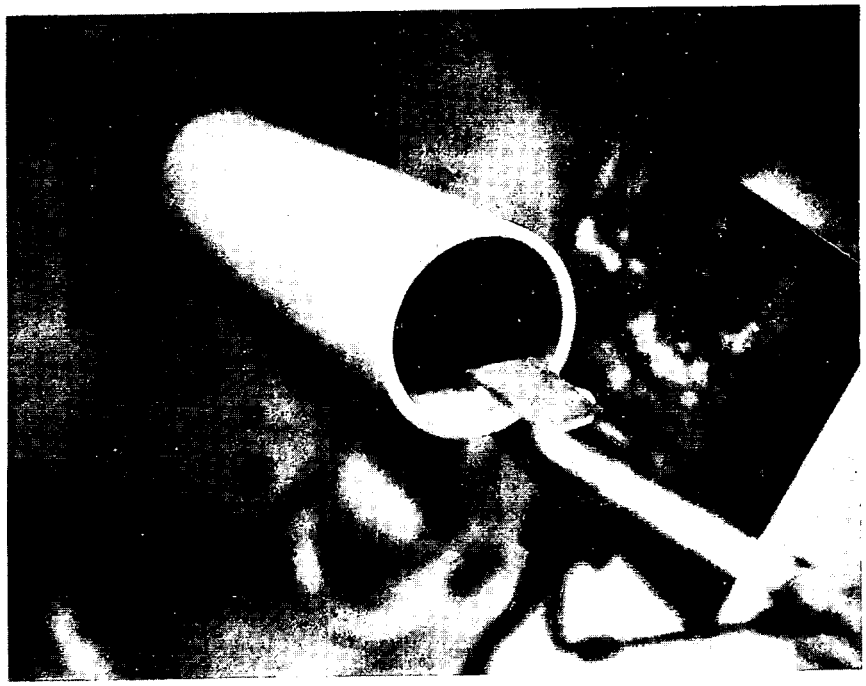
Another material, woven Irish Refrasil ceramic fiber cloth, a product of the H.I. Thompson Company, was also evaluated but not coated since the basic material contains chromic oxide.

Coupons approximately 1" x 2" in dimension of the various materials were spray coated, dried, and fired at temperatures ranging from 1850°F to 2150°F, depending on coating composition. Coated specimens were examined under a microscope to determine coating continuity, coating thickness, and, in the case of felt metal specimen, the depth of coating penetration into the fiber mat.

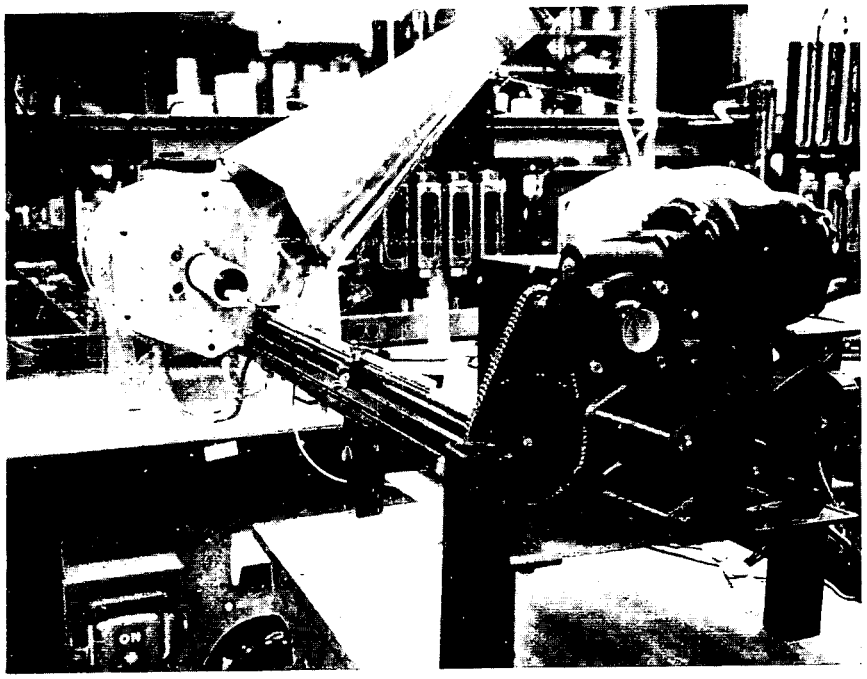
Thermal cycling tests were accomplished by inserting a specimen into a 2300°F furnace for five (5) minutes and then retracting the specimen to cool at room temperature. Two and one-half (2.5) minutes each were required for insertion and retraction, and all specimens were subjected to 35 cycles. Figure 3-3 shows an overall view of the cycling rig and a close-up of the specimen in place.

Coupon Results

Consistently good coatings, uniformly green in color, were produced on the Hoskins 875 alloy with both the refractory and silicate coating materials. Although the Nextel ceramic cloth accepted both coatings very well, the ceramic cloth subsequently became severely embrittled. The Incoloy 800 wire cloth appeared to react with both coating types to form areas which were dark brown to black in color and totally devoid of coating.



(a)



(b)

Figure 3-3. Thermal cycling rig overall view (a) and close-up (b) showing specimen about to enter furnace

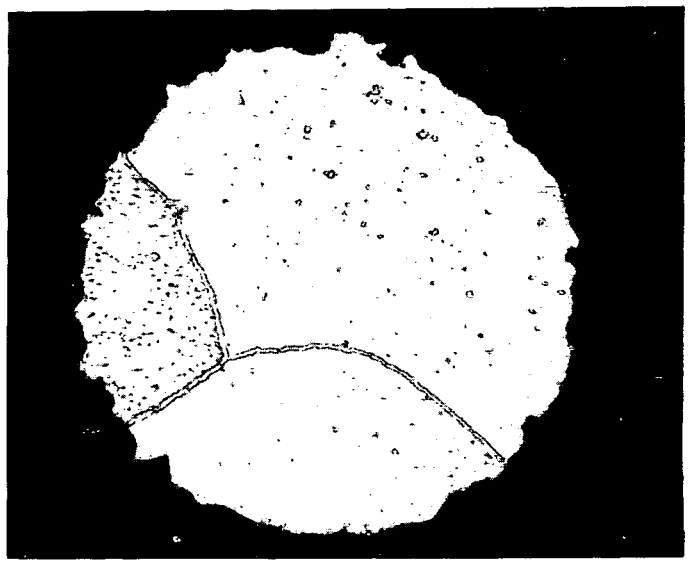
Metallographic examination of coated and uncoated Hoskins 875 after 35 thermal cycles revealed that the coated specimen contains more grains per fiber cross-section than the fibers in the uncoated specimen: evidence that more grain growth occurred in the uncoated fibers. These results are shown in the photomicrograph of Figure 3-4. The encouraging results obtained by coating small pieces of Hoskins 875 material prompted a decision to coat eight 3.5" diameter disks of this material:

Sample No.	Coating
1	Uncoated (control)
2,3	Light (single pass) refractory coating
4,5	Heavy (double pass) refractory coating
6,7	Light (single pass) silicate coating
8,9	Heavy (double pass) silicate coating

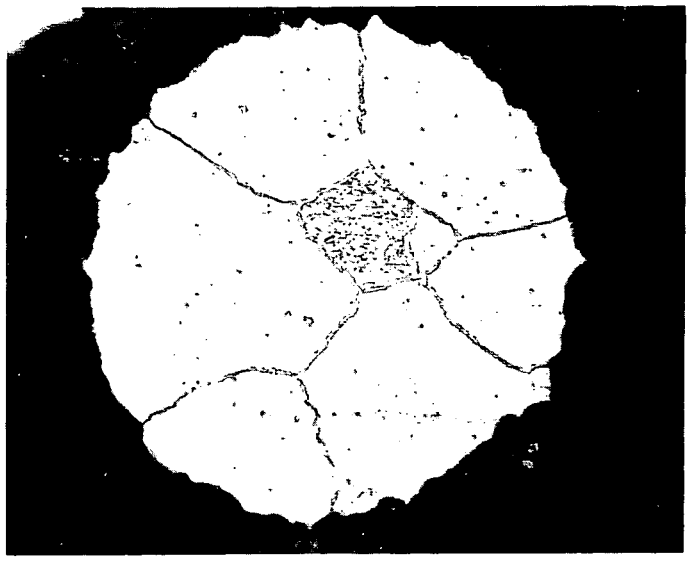
The same procedure was used to coat the disks as was used on the small screening specimens except that a larger furnace was used to fire the coatings. All but two of the coated disks (heavy silicate coating) exhibited some spalling of the ceramic coating. This behavior was not observed before. The probable explanation was that the shelf life of the coating may have expired and/or different levels of pre-oxidation existed for the disks. The effect of the spalling will be determined from longer duration cycle testing.

Pressure Drop Measurement

In order to ensure that the application of the protective coating does not result in excessive pressure drop across the burner element, pressure drop across the coupon was measured before and after coating. Tests were for a range of air flow rates at 85°F, 300°F and 600°F preheat temperatures. The coated coupons were also fired for half an hour each and pressure drop was measured for firing rates from 10 KBTUH to 50 KBTUH and 15% excess air.



(a)



(b)

Figure 3-4. Cross-section of uncoated (a) and coated (b) Hoskins 875 fibers after 35 thermal cycles

Results of the pressure drop test on the coupons were correlated versus predictions. Since the results for all the disks were extremely close, data for the control disk can be considered to represent the entire sample. Results for the uncoated control disk are very well correlated for the entire range of preheat temperatures and flow rates, as shown in Figure 3-5. For an equivalent firing rate of 60 KBTUH with 15% excess air for a full size burner element, the pressure drop would be less than 0.4 in of H_2O for preheat temperatures up to 800°F, as shown in Figure 3-6.

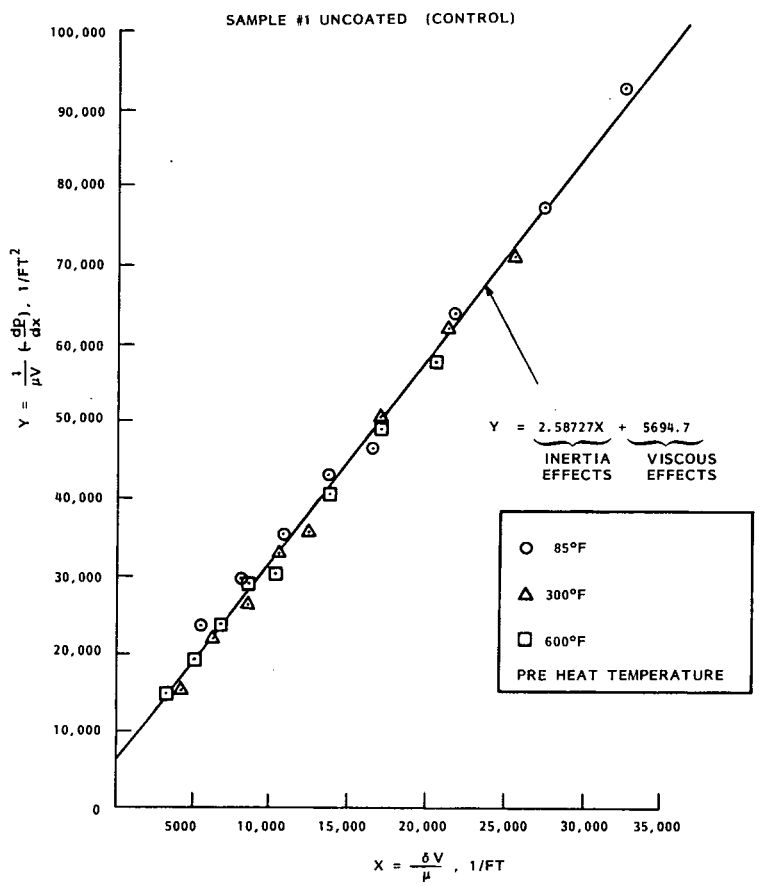


Figure 3-5. Pressure Drop Correlation on Control Disc

After the application of the coatings, pressure drop data were again correlated, as before. Since the results for all the disks were very close, only the correlation for sample disk #9 (coated with heavy silicate) is shown in Figure 3-7. A comparison of Figures 3-5 and 3-7 reveals that after coating, the contribution to pressure drop from viscous effects has increased, whereas that from inertia effects has remained the same. Further comparison of results before and after coating suggests that the increase in pressure drop after coating is relatively insignificant.

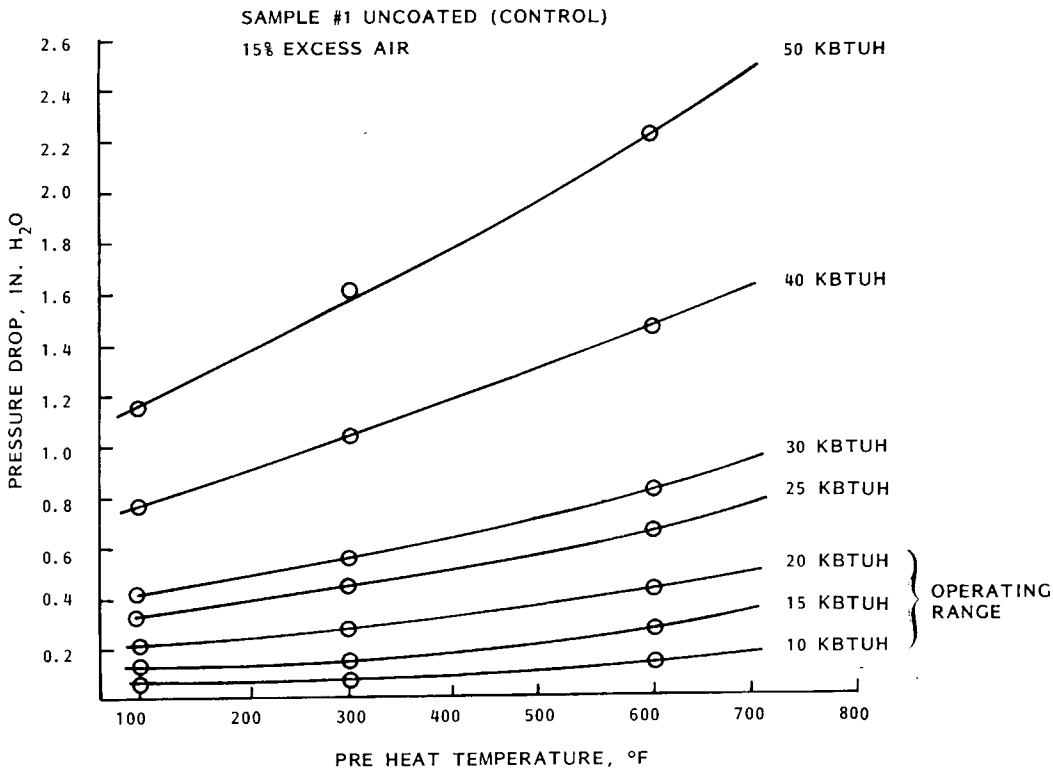


Figure 3-6. Pressure Drop as a Function of Pre-Heat Temperature for Different Firing Rate

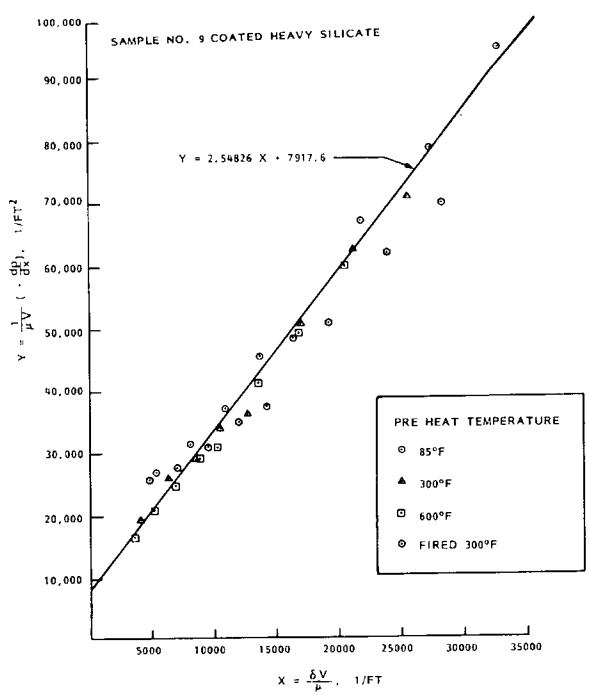


Figure 3-7. Pressure Drop Correlation After Coating

3.2.4 FULL SIZE BURNER ELEMENT

Upon successful completion of the testing on coupons, a full size burner element was coated with the refractory coating utilizing the same procedures as before. The burner element was subjected to a pressure drop test in the component test facility. Figure 3-8 shows that excellent correlation of the test data is obtained over the entire range of firing rates and pre-heat temperatures. Again, no difference in the pressure drop across the burner element could be distinguished before and after the coating. Hence, only the results for the uncoated burner are shown.

When the coated burner element was fired, some spalling of the coating was observed. This was confirmed after firing when the accumulation of small specks of coating material was observed underneath the burner element. A preliminary evaluation of the coating process revealed that when the coating material is cold sprayed, its adherence to the substrate material is very sensitive to the surface preparation technique since only a mechanical bond is obtained during the coating process.

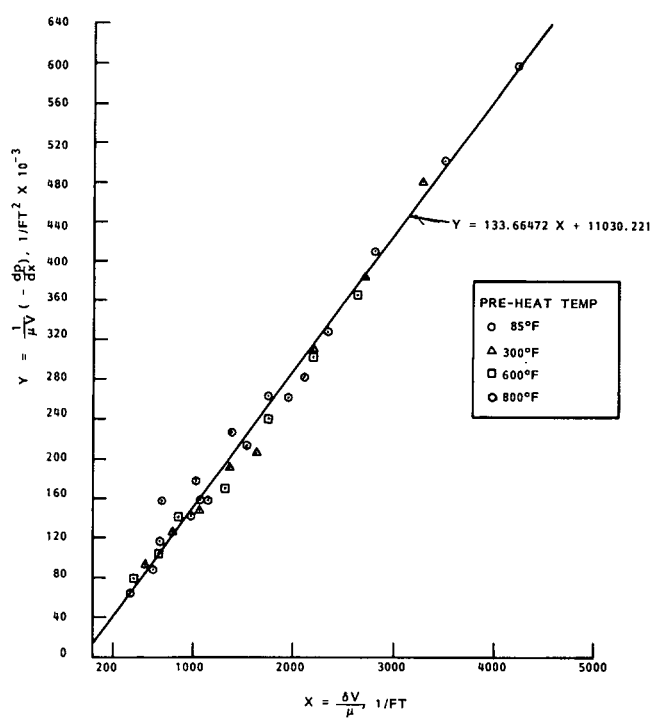


Figure 3-8. Pressure Drop Characteristics of Full Size Uncoated Burner Element

3.2.5 PLANNED ACTIVITIES FOR NEXT PERIOD

Based upon the results obtained during this reporting period, good coating adherence requires a chemical bond between the coating and the substrate, in addition to mechanical bond. Plasma spraying of the coating on the substrate appears promising. Techniques for plasma spraying a ceramic coating on the porous metal will be considered during next reporting period. In addition, other possible means of enhancing the burner life are also being evaluated.

3.3 ENGINE

3.3.1 SUMMARY

The engine development effort during this reporting period consisted of continued support of the Proto-2 Integrated Test program, plus completion of the engine housing redesign and fabrication. The modified engine housing was designed to improve engine power output and efficiency through a modification of the engine heater head, regenerator and cooler assemblies. This redesign effort entailed the completion of detail and assembly drawings and the subsequent retrofit of a Proto-2 engine housing.

3.3.2 PROTO 2 ENGINE DEVELOPMENT

Proto 2 engine development effort during the early portion of this reporting period included assembly of hardware configurations for performance-trend testing.

- A variety of second gas spring vent port areas and hole patterns to reduce mid-stroke pumping losses.
- A second gas spring piston having both ends chamfered to eliminate flow restrictions during the vent portion of the piston stroke.
- A new set of second gas spring seal rings incorporating an improved step joint overlap to reduce seal leakage.
- Second gas spring mid-stroke volumes of 81 and 122 in.^{3} to vary the spring stiffness and change operating frequency.
- Displacer rod areas of 1.48 in.^{2} (G-3) and 1.58 in.^{2} (G-4) to vary the displacer gas spring characteristics.

- Displacer gas spring volumes of approximately 10.5, 11.5, and 12.5 in.^{3} to vary the displacer gas spring characteristics.

- Displacer rod centering port areas and/or locations varied to center the displacer with reference to the power piston.

- The engine housing weight was doubled to effectively reduce the engine housing motion by 50 percent. The added weight of 276 lbs. was assembled to the engine around the pressure vessel, and the increased load was supported by the addition of housing to ground isolation springs. A photograph of this assembly is shown in Figure 3-9.

- The power piston seal ring grooves were modified and expander rings were installed under the seal rings.

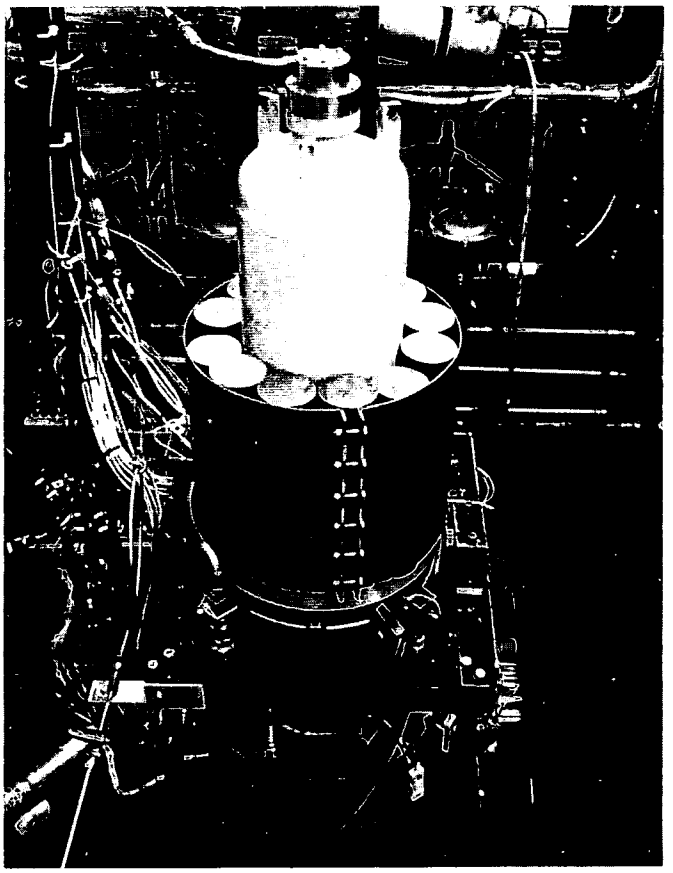


Figure 3-9. Proto 2 Engine Assembly with Housing Weight Doubled by Adding External Weights

The above modifications were primarily directed toward reducing the second gas spring power losses and/or tuning engine dynamic components for improved performance. One change was made to evaluate the effects of reducing the engine housing vibration amplitude,

and another was an attempt to reduce the power piston seal leakage. Figures 3-10 and 3-11 are cross-sectional views identifying the components described above.

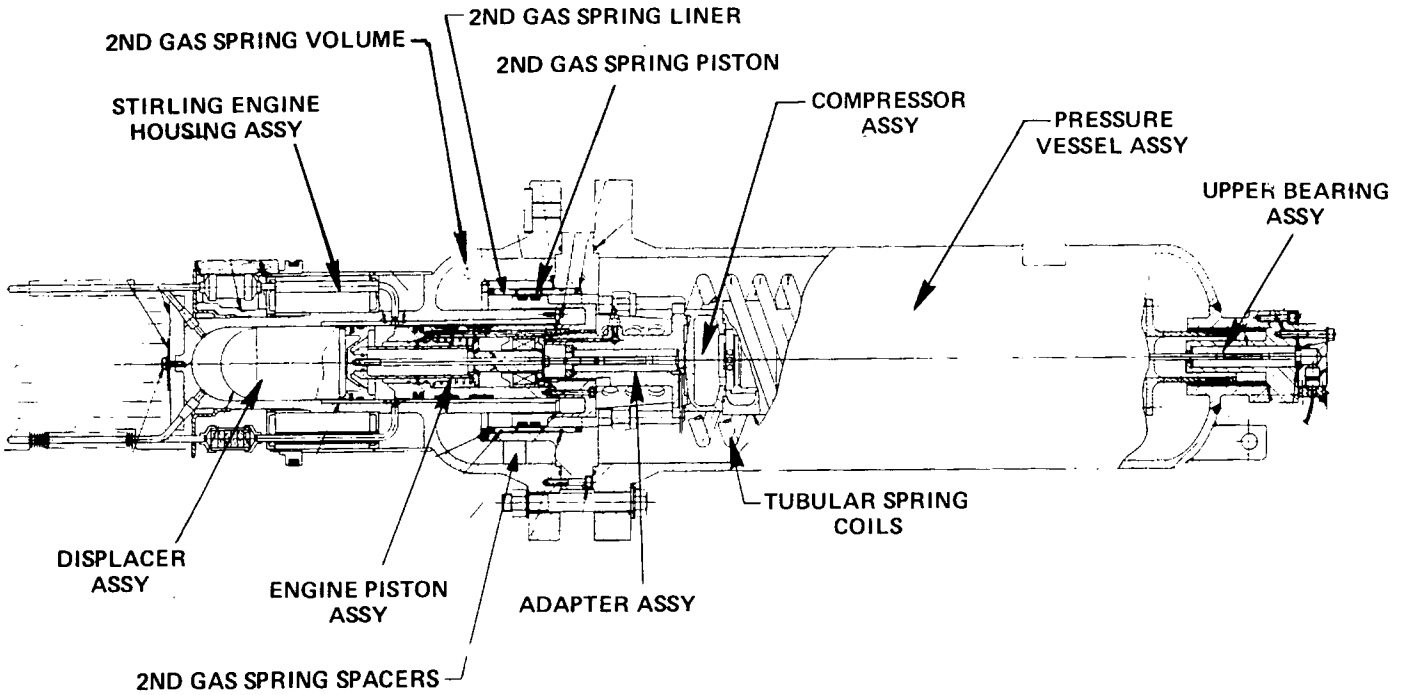


Figure 3-10. Proto 2 Engine and Compressor Assembly

Results of these tests are summarized in Section 2 and Section 4 of this report. Further testing of the engine was prevented by the development of hairline cracks in two of the engine heater head tubes. Both cracks were located at the braze fillet of the first fin on the regeneerator side of the heat exchanger (Figure 3-12 (a)). A failure analysis was completed, the results of which showed that in the Inconel 625 tubing, the wall areas in which the crack occurred had been undercut approximately 1/3 of the wall thickness during the original brazing process. Approximately another 1/3 of the wall thickness had reacted with the braze material and formed alloys. The balance of the 0.028 inch wall thickness was good base metal, although the grain size had increased as a result of the multiple brazing cycles. The combination of thermal and mechanical stresses during engine operation produced small cracks in the "notched" section of the tube. The problem is unique to the outboard fin region adjacent to the very rigid fin and tube assembly.

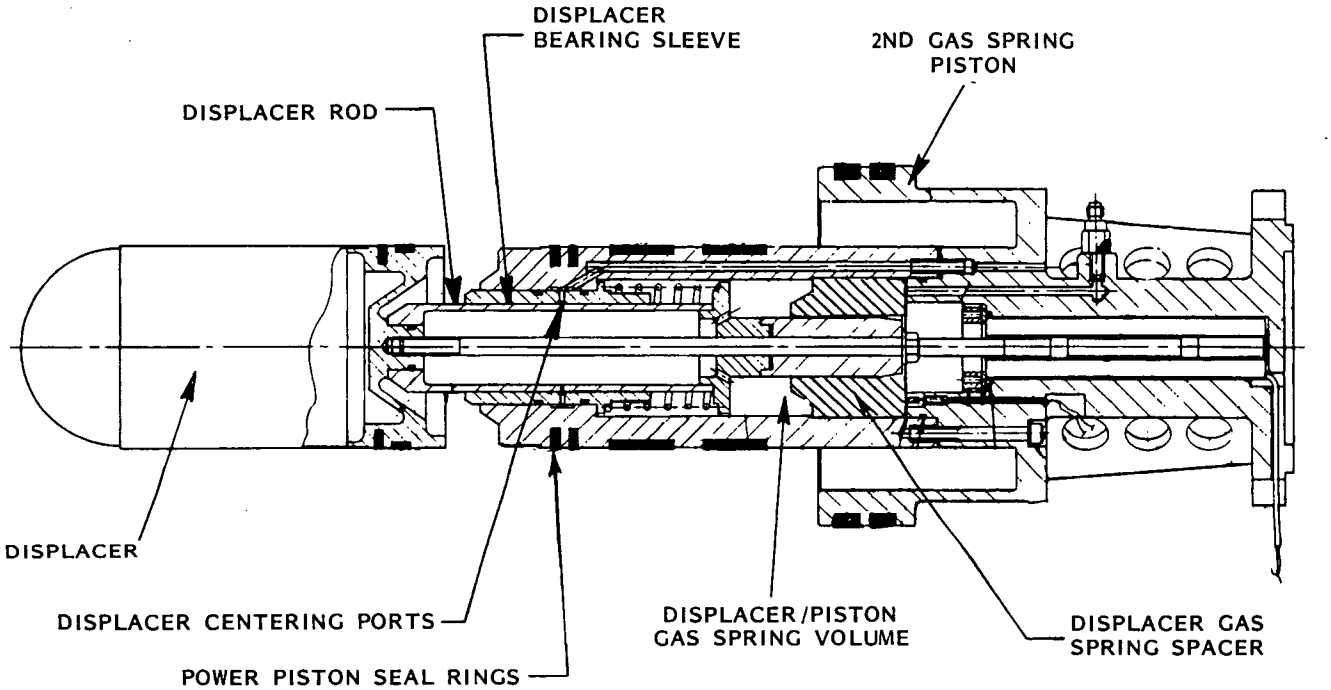
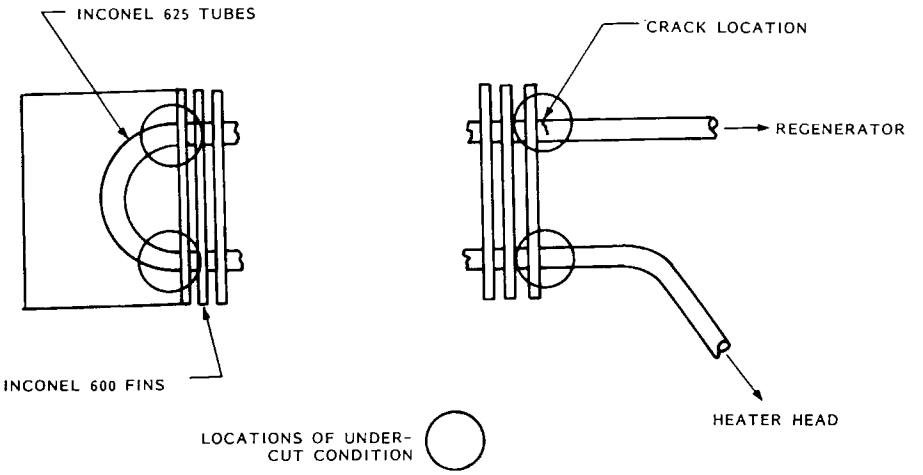


Figure 3-11. Proto 2 Displacer, Power Piston, and 2nd Gas Spring Piston Assembly

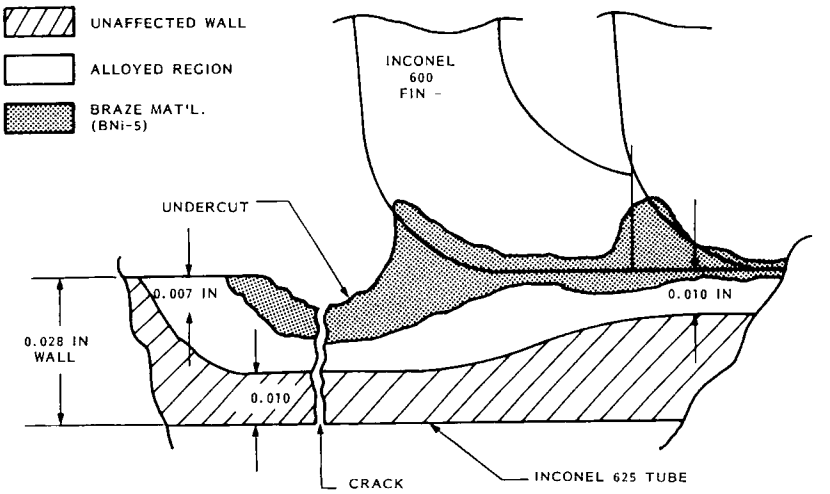
An alternate assembly/braze procedure was developed as a result of this analysis to prevent a re-occurrence of the problem in the engine modification effort discussed later in this report (modified Proto 2 engine). Fin and tube assembly braze specimens were subsequently fabricated and processed through this alternate braze cycle. The results, shown in Figure 3-12 (c), indicate that good braze joints were obtained, permitting release of the prime hardware for the braze operation.

3.3.3 MODIFIED PROTO 2 ENGINE DEVELOPMENT

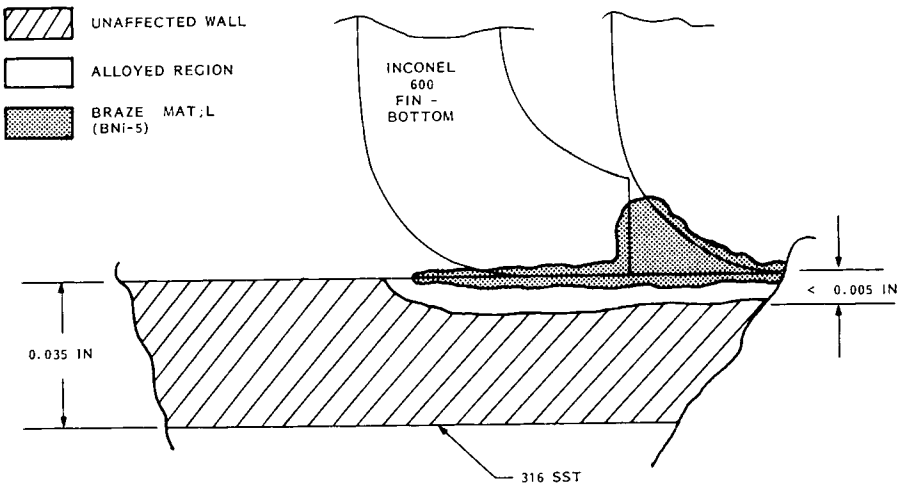
Engine design modification/performance improvement studies initiated during the last reporting period were completed during this reporting period. These studies, included parallel analytical optimization and hardware layout efforts resulting in the definition and selection of new heater head, regenerator, and cooler assembly designs that were compatible with the basic Proto 2 engine. The hardware layout studies and subsequent design, fabrication, and assembly efforts are discussed in the following paragraphs.



(a) Location of Failure on Heater Head Fin/Tube Assembly



(b) Failed Tube Section



(c)Revised Process Braze Sample

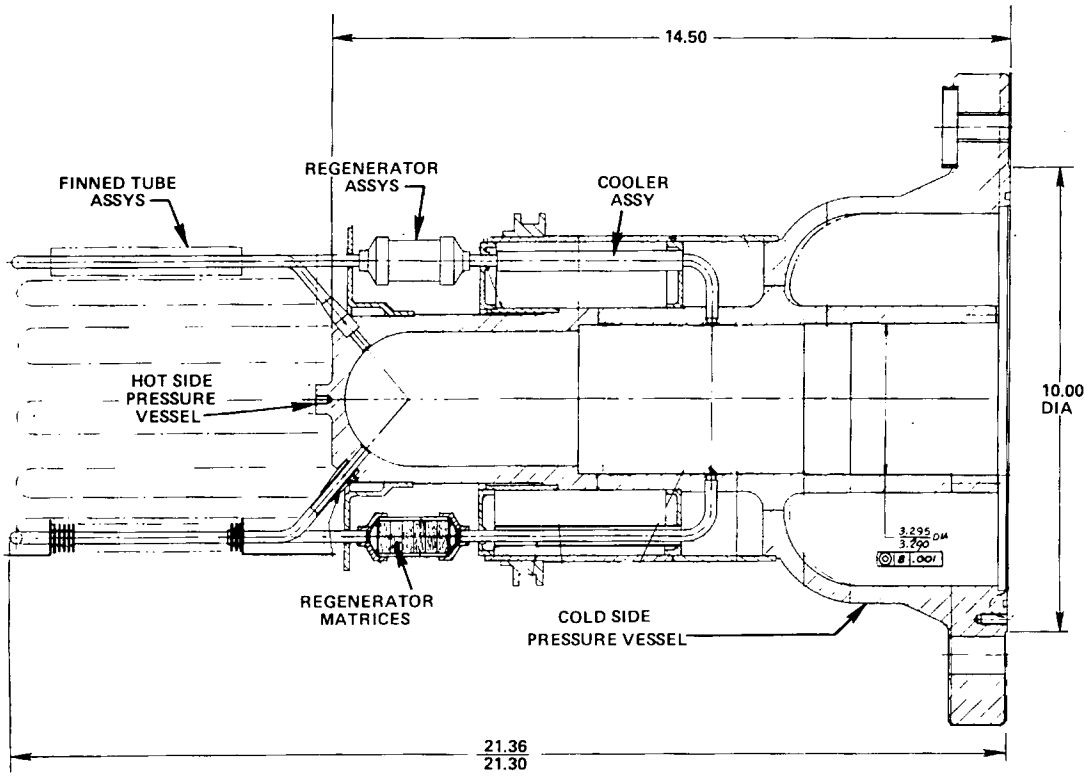
Figure 3-12. Fin/ Tube Assembly Failure

In concert with the analytical studies identifying potential performance-improving configurations, hardware layout studies were conducted to evaluate the impact of these changes on the basic engine housing assembly. Heater head, regenerator, and cooler assembly modifications were evaluated, with the common restraint that all changes had to be compatible with inter-relating Proto 2 hardware, including the rework/modification of an existing engine housing assembly.

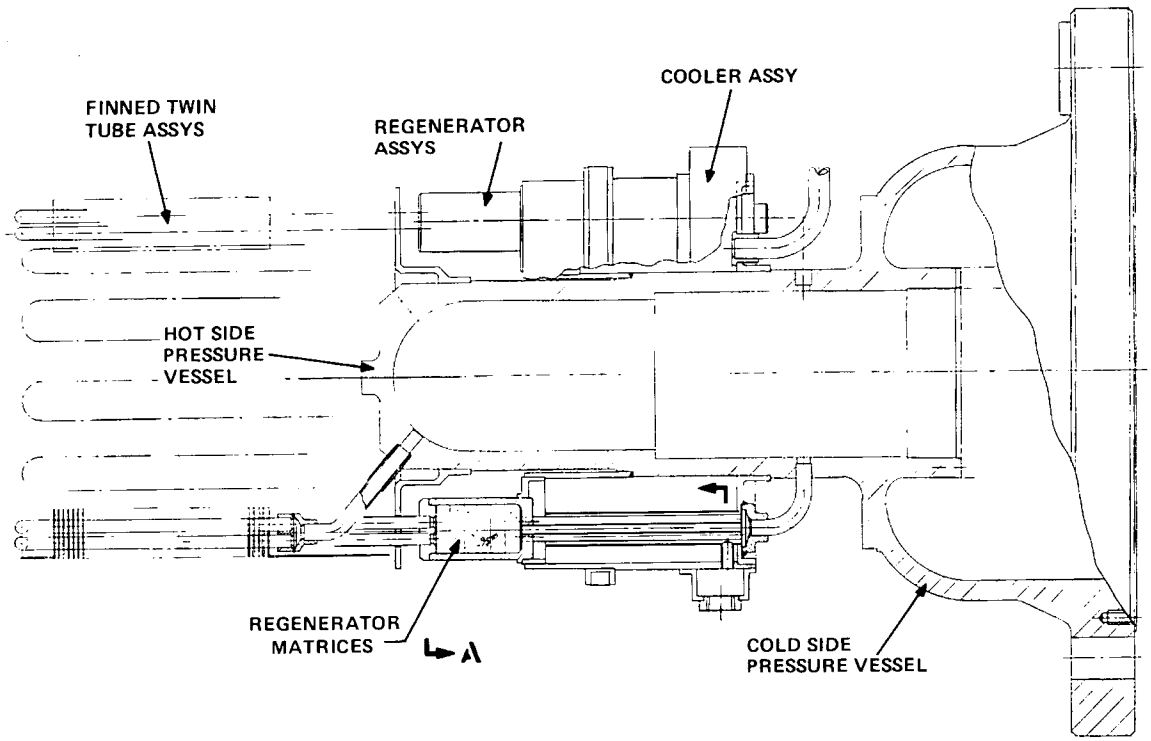
Based on these studies, a modified engine configuration was selected and detail and assembly drawings were completed. Figure 3-13 (a) shows a baseline Proto 2 engine housing and Figure 3-13 (b) illustrates the modified housing design.

Material selections for the modified Proto 2 engine housing were made to be compatible with all inter-relating Proto 2 hardware, available within a reasonable time-frame, and suitable for engine performance/operating conditions. These goals were met with the material selections listed in Table 3-1. The baseline Proto 2 engine housing materials are also presented for comparison. The differences are seen to be in the substitution of CRES, Type-316L, for the Inconel 625 regenerator housing and heat exchanger tubing. This material substitution will not limit engine operating temperatures and pressures, but does reduce the unit's design life to approximately 1000 hours. This design life is considered to be more than adequate for the performance testing effort envisioned for this engine.

The Proto 2 engine housing retrofit effort was completed on schedule and included the installation of the cylinder liner and the hot side instrumentation thermocouple. Figures 3-14 and 3-15 show photographs of the modified engine housing and engine heater head, respectively. A flow versus pressure drop test was conducted on the modified Proto 2 engine housing heater head to establish the steady state pressure frop characteristics of the new configuration. A schematic diagram of the pressure drop test set-up is presented in Figure 3-16. The test results show that the measured pressure drop of the retrofit engine is less than that of the original Proto 2 unit, and in good agreement with predicted values. Figure 3-17 presents the heat exchanger pressure drop characteristics for both original and retrofit engine configurations.



(a) Baseline Proto 2 Housing



(b) Modified Housing

Figure 3-13. Proto 2 Baseline and Modified Engine Housing Designs

Table 3-1. Material Comparison of Proto 2 Engine Housings

Item	Material	
	Baseline	Modified
Housing Weldment	Inconel 625 and 316L SS	Inconel 625* and 316L SS
Cooler Tubes	Inconel 625	CRES, Type 316L
Cooler Housing	CRES, Type 316L	CRES, Type 316L
Regenerator Housing	Inconel 625	CRES, Type 316L
Regenerator Matrix	CRES, Type 309	CRES, Type 309
Heater Head Tubes	Inconel 625	CRES, Type 316L
Heater Head Fins	Inconel 600	Inconel 600

*The same engine housing weldment

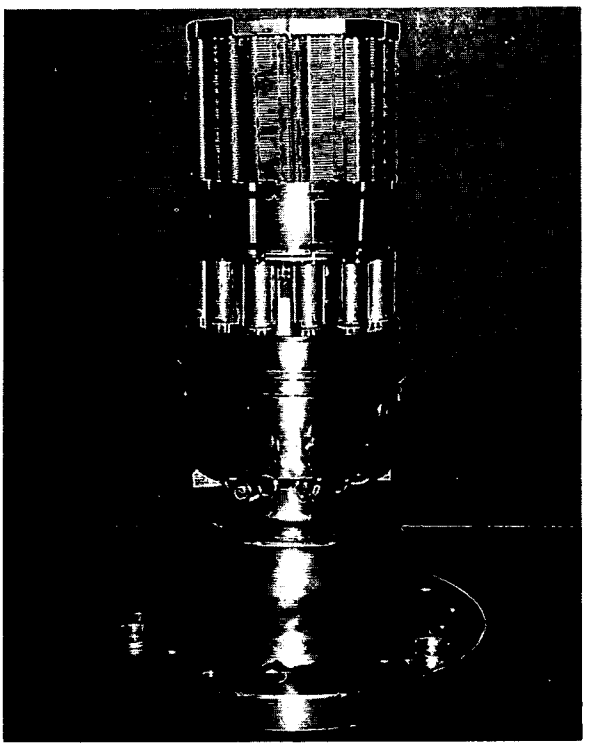


Figure 3-14. Modified Proto 2 Engine Housing

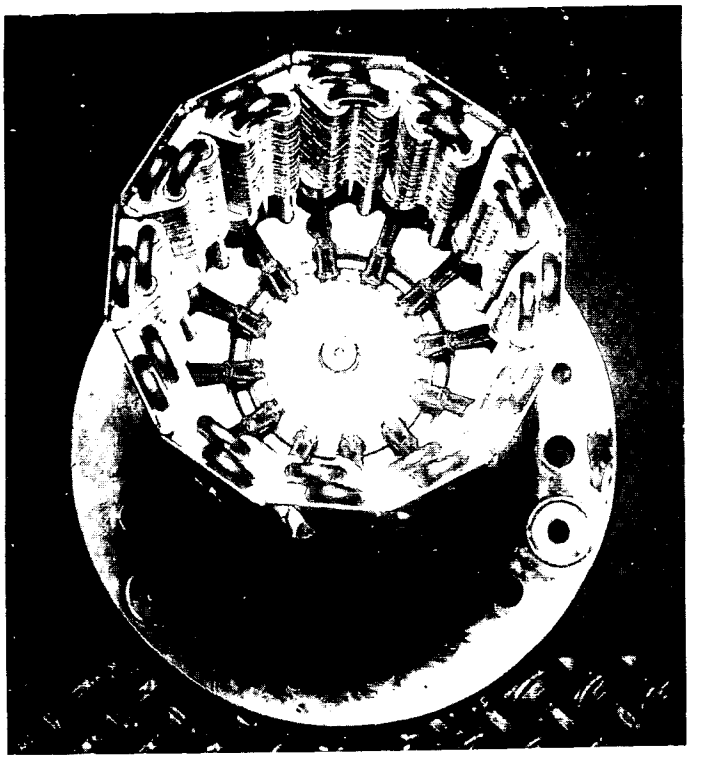


Figure 3-15. Modified Proto 2 Engine Heater Head

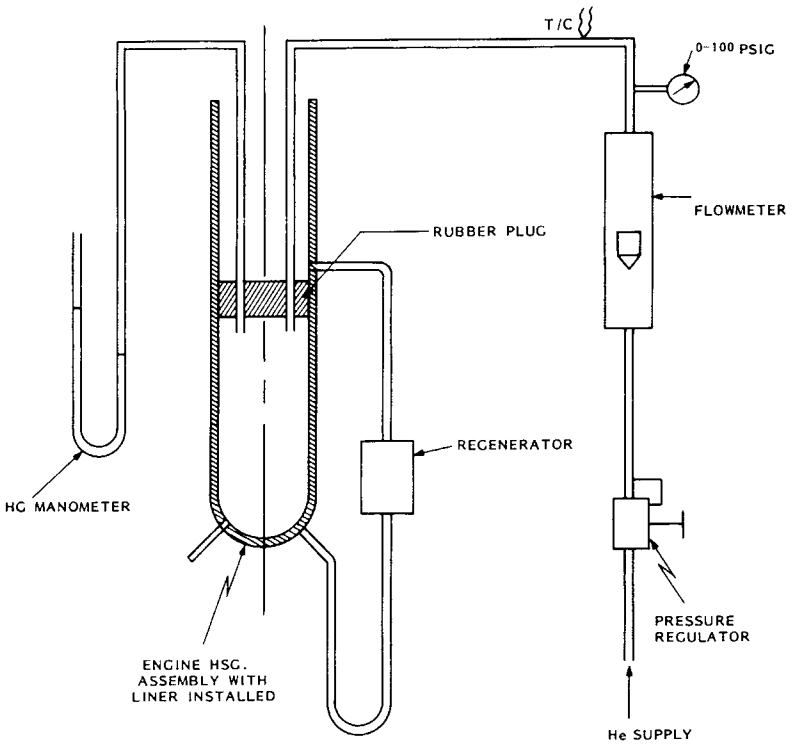


Figure 3-16. Schematic Diagram of Heater Head Pressure Drop Test

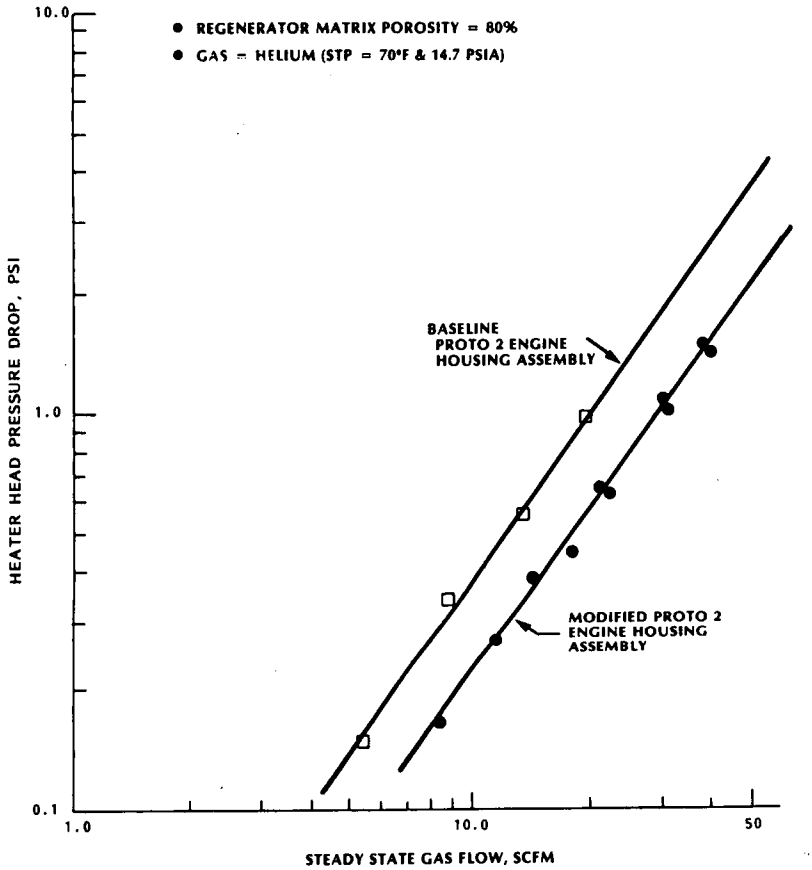


Figure 3-17. FPSE Modified Proto 2 Engine Heat Exchanger Pressure Drop

The Proto 2 modified combustor/ engine/ compressor units were assembled and subsequently installed in the Integrated Test Stand near the end of this reporting period. Leakage tests were also completed on both the helium and refrigerant loops, and good tight systems were verified. Integrated Tests were subsequently initiated. (See Section 4)

3.3.4 SEAL AND BEARING DEVELOPMENT

The seal and bearing wear test rig was de-bugged early in this reporting period with the installation of structural stiffeners, an oil seal on the drive rod, and a gas-to-water heat exchanger in the helium compression loop. Wear tests were subsequently started, and eight sets of rod/ sleeve and two set of ring/ liner wear test candidates were subjected to 72 hour screening tests. These ring/ liner screening tests will continue into the next reporting period.

Rod/ Sleeve Bearing Tests

Eight rod and sleeve matched sets were subjected to the endurance wear screening tests, with a 72 hour operational goal. Table 3-2 summarizes the materials and coatings of these test sets.

The first two sets failed by seizure in 1.8 and 4.3 hours, respectively. Seizure occurred when the rod Tufram coating wore through and exposed the aluminum base material. An examination of the test stand reciprocating drive assembly and its supporting structure indicated that the relatively early failure of Group 1 and 2 might be due to excessive flexibility of the structure and/ or an excessive misalignment of the drive assembly. The structure was stiffened by the addition of braces, and an alignment tool was fabricated to verify good assembly alignment. A new Group 1 matched-set was also ordered for subsequent re-test of this material combination.

Groups 3,4,7,8,9 and 10 matched-sets were subjected to and completed their 72 hour screening tests. The relative wear of these test candidates, as indicated by maximum diametrical wear and weight loss, is shown in Figure 3-18. It is shown that the minimum wear was experienced on the Group 9 test specimens, closely followed by Groups 7 and 8. All of these test groups incorporated rod coatings of RULON LD.

The test rig was next changed over to accomodate the testing of seal and rider rings.

Ring/ Liner Wear Test

Initial testing of the seal/ rider rings and liner wear test specimen combinations identified a temperature gradient problem in the test fixture/ test hardware assembly. The piston and test rings ran hotter than predicted with reference to the mating liner, and the piston growth was subsequently greater than predicted. The rider rings, trapped between the piston and liner, were subjected to high loading which resulted in increased friction, and, therefore, higher operating temperatures. The wear test rig was therefore modified to incorporate a water to helium heat exchanger into the helium compression loop, and the high temperature problem was solved.

Table 3-2. Matched - Set Matrix for Rod and Sleeve Wear Test

Test Sequence	Group No.	Material Combinations			
		Rod		Sleeve	
		Mat'l	Coating	Mat'l	Coating
1	1	6061-T6 Alum	Tufram + Mo S2	4340 Steel	Chrome + Mo S2
2	2	6061-T6 Alum	Tufram + Mo S2	440C Cres	
3	4	6061-T6 Alum	Rulon-Ld	440C Cres	-
4	7	440C Cres	Rulon-Ld	440C Cres	Xylan
5	10	440C Cres	Rulon-J	Nitronic 60	-
6	8	440C Cres	Rulon-Ld	Nitronic 60	-
7	9	440C Cres	Rulon-Ld + Mo S2	Cast Iron	Mo S2
8	3	6061-T6 Alum	Koplon + Mo S2	4340 Steel	Chrome + Mo S2
9	1a	6061-T6 Alum	Tufram + Mo S2	4340 Steel	Chrome + Mo S2

Two sets of ring/liner test specimens completed their 72 hour screening tests, and a third set was being installed at the end of this reporting period.

3.3.5 TRANSIENT LOAD ACCEPTANCE/OVERSTROKE PROTECTION

The purpose of the transient load acceptance/overstroke protection system is to control the engine piston stroke over the system operating load range. Characteristically the Free Piston Stirling Engine piston stroke is a function of the load applied. As the load is decreased the piston stroke increases. Maximum piston stroke levels must be controlled to prevent damage to internal engine components. This is to be accomplished in the Proto 2 engine via a piston mounted position sensor which modulates a solenoid valve that in turn by-passes working fluid around the piston when the piston exceeds its nominal stroke.

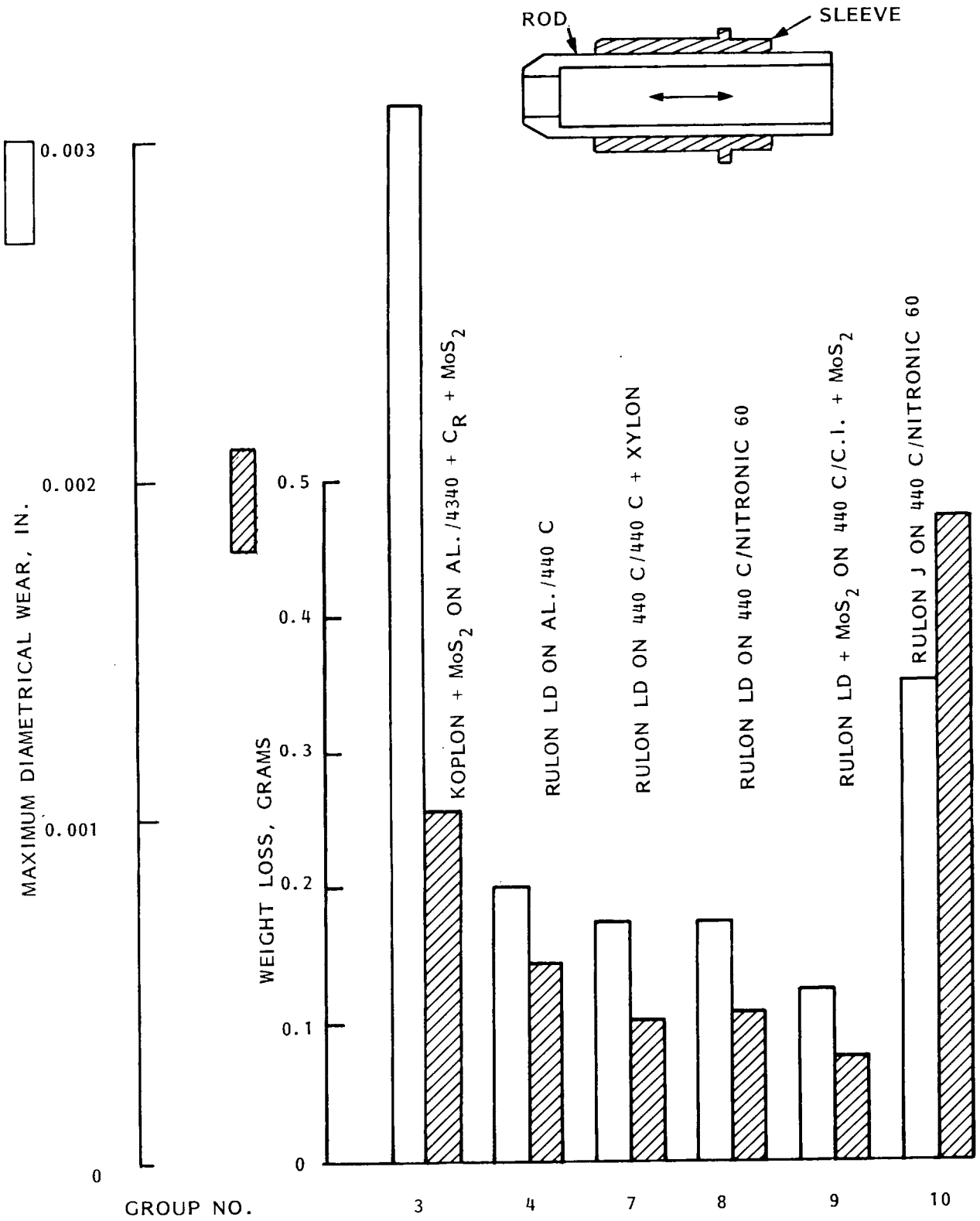


Figure 3-18. 72 Hour Rod/Sleeve Wear Test Groups

System Design

Engine requirements dictate that the response time of the system shall be such that the engine piston will contact the mechanical stops a minimum number of times during periods when the engine load is changing. The engine operating frequency is basically 30 cycles/sec. A modulated solenoid valve with a response time of 10 milliseconds was selected (full open to full closed). In addition, an allowance of 5 milliseconds was alloted for the control electronics. Therefore, the maximum total number of piston cycles or collisions for the maximum regulation (full open to full closed) time period is one half cycle.

Figure 3-19 depicts the magnet and sensor mounting configurations. As illustrated, the magnets are mounted on the moving part (engine piston) while the sensor is mounted to the stationary housing. The system will initiate the bypass made at a predetermined percentage of the total available piston travel between the mechanical stops.

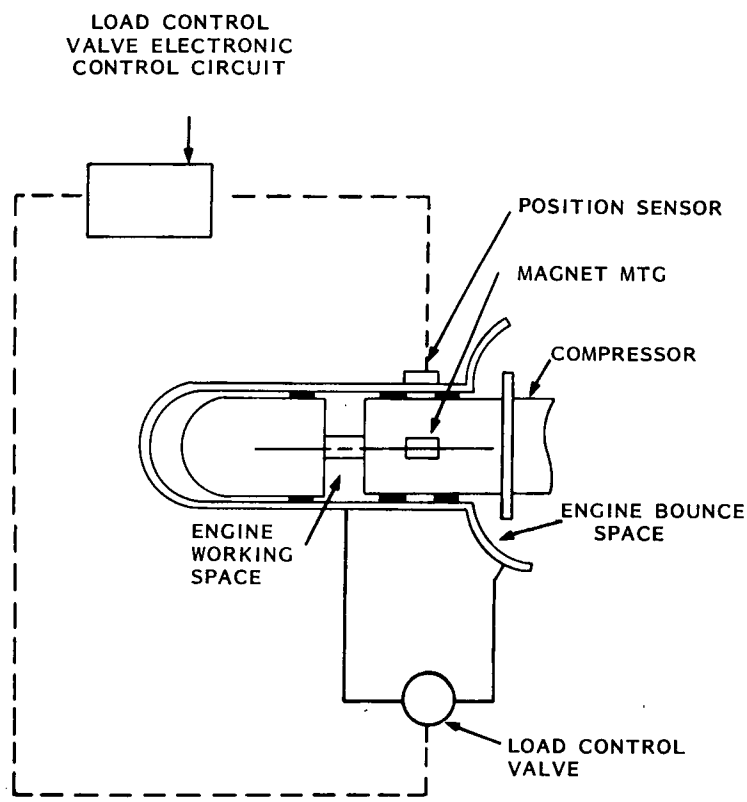


Figure 3-19. Transient load acceptance/overstroke protection system

As previously reported, several configurations (placement of magnets on power piston) were tested and the optimum selected for the hardware design. In addition, the electronic control circuit was designed, fabricated and tested. The circuit exceeded the function requirements. A load control valve was selected, procured and tested in conjunction with the control circuit and position sensor. The system met the requirements as previously described. Figure 3-20 illustrates the test set-up. All hardware associated with the sensor and magnets have been fabricated and trial assembly to an engine has been successfully completed.

The next step in the transient load acceptance/ overstroke protection system development is to install the components into the integrated test set-up, and test the total subsystem. This effort has been deferred to permit higher priority tests.

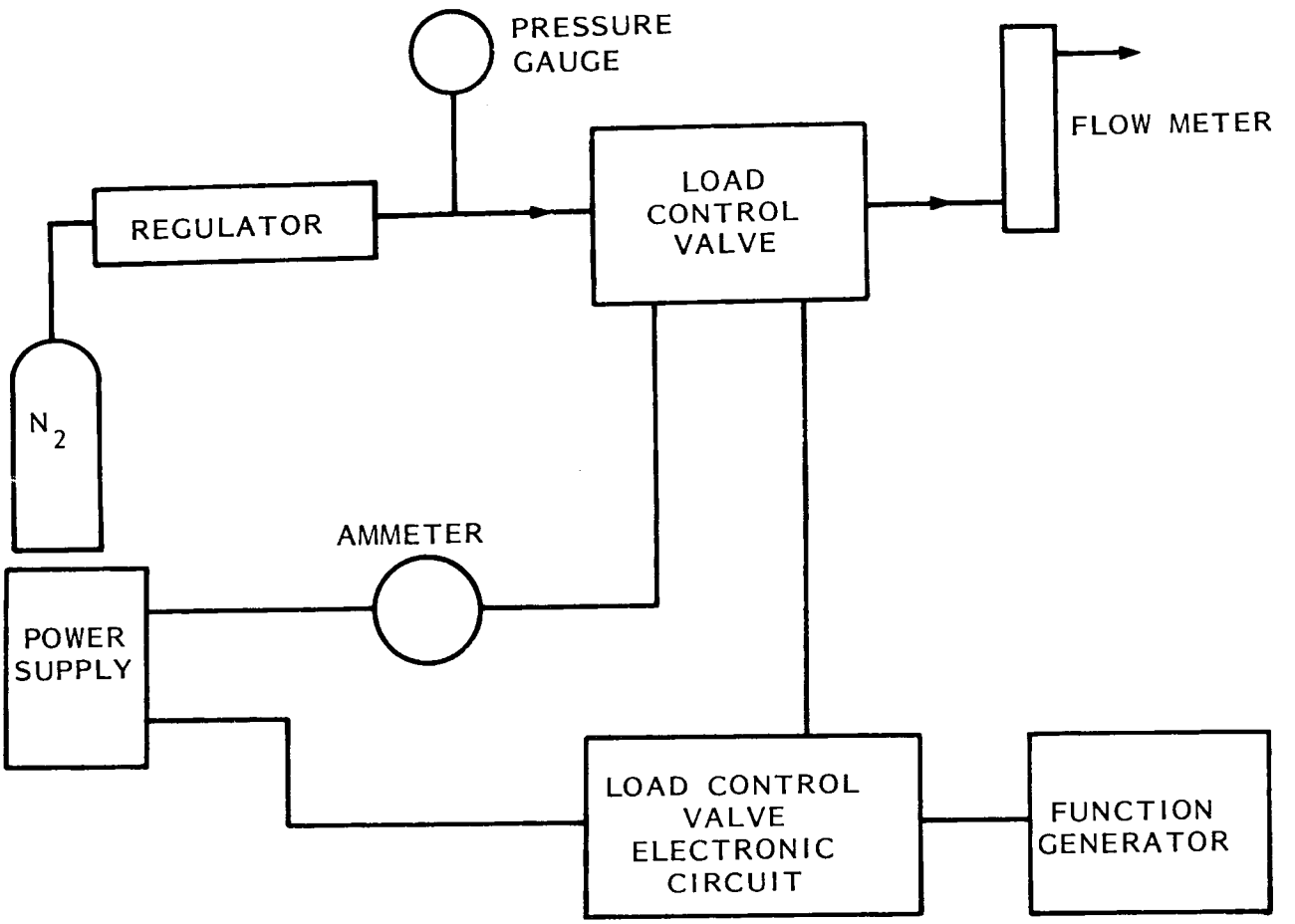


Figure 3-20. Transient load/ overstroke protection load control valve test set-up

3.3.6 PLANNED ACTIVITY FOR NEXT PERIOD

Next period activities will be the support of the modified Proto 2 engine Integrated Test effort. Various component configuration changes (gas spring area and/or volume, vent port area and/or location, etc.) will be implemented for both timing and performance evaluations, similar to those performed for the baseline Proto 2 engine.

In addition, it is scheduled to complete the 72 hour screening tests on all rod/sleeve and ring/liner test candidates, select the two most promising wear-life candidate sets from each combination, and start the 144 hour extended wear tests on these selected candidates.

3.4 COMPRESSOR

3.4.1 SUMMARY

Compressor activities this period consisted of the assembly and test of a Proto 2, Unit 3 compressor and the correlation and matching of the compressor computer program to the test results. Compressor operation at design conditions and near design performance were the significant accomplishments this period. The improvement in performance of the Unit 3 assembly was attributed to the addition of a third piston seal ring which reduced gas spring leakages and power losses while producing centered piston operation. The piston weight was also increased to 10.72 from 9.62 lbs. to enable compressor/engine power matching at lower strokes and frequencies. The unit was tested at the 95°F ambient degree day design point, producing approximately 560 lbs/hr flow, a 3.2 kW input power, and nearly 70% efficiency. The following sections describe the hardware changes, test results and analysis activity this period.

3.4.2 HARDWARE MODIFICATIONS

The Proto-2 Unit 3 compressor was assembled this period with several modifications to improve performance. The primary modification was the addition of a third piston ring on each piston to more effectively seal the gas spring volume, thus reducing leakages and power losses encountered in the previous two units. This third ring, located at the gas spring end of the piston, prevents leakage along the piston/cylinder wall interface to the reference ports. The 2nd piston ring, located about 1.0 inch from the same end of the piston, now provides backup sealing for the third ring and also seals whenever piston travel is sufficient to move the third ring past the external volume ports. In this position the 2nd ring prevents leakage from the external volume to the reference port while the 3rd ring provides sealing for the "hard" gas spring effect produced by sealing off the external volume. The 1st piston ring primary function is sealing the compression spaces.

Two additional changes were tested with no improvement in compressor functional operation or performance. The first was the elimination of the suction manifold flow inserts originally added to balance flow between upper and lower cylinders. The second was the elimination of

the connecting rod center porting for reducing gas spring pumping losses. Rather than improve performance, both modifications resulted in a degradation of piston centering and were discarded.

The Unit 3 compressor test results presented in the following section are based on the configuration which included the third piston ring and heavy piston (10.72 lbs). The heavy piston configuration was obtained by substituting bronze for cast iron in the two piece cast iron/tungsten piston assembly. The heavy piston improved matching of the power absorbed by the compressor to that produced by the engine at lower frequencies and strokes. In addition, this unit included the following features of the Unit 2 build:

- Suction flow inserts
- Connecting rod with reduced center porting
- Carlton pressure regulator (110 psia)

3.4.3 TEST RESULTS

The previously described modifications improved the functional operation of the compressor and permitted performance testing at the 95°F and 80°F ambient degree day conditions. The heavy piston reduced the operating frequency and/or stroke of the compressor at design conditions to better match the predicted rebuilt engine operation. The unit was assembled with the engine and checkout testing begun at the end of this period.

Functional Tests

The functional operation of the compressor was improved to eliminate the off-center piston operation encountered in previous builds. The off-center operation was produced by two factors; leakage (possibly unequal) out of the gas springs, reducing their effectiveness; and, unequal suction flow pressure drop to the two compression cylinders. The suction flow inserts used to balance pressure drops reduced off-center operation, but it was not eliminated until the third piston ring was installed. This third ring reduced leakage and hence reduced power loss. The increased effectiveness of the gas springs contributed to the elimination of off center operation of the piston. It is also suspected that uneven leakage out of one gas

spring resulting in a softer spring, contributing to off center operation. The 3rd piston ring corrected both problems simultaneously.

The compressor functional operation has been satisfactory in all other respects. The switch to a heavier (10.72 versus 9.62 lbs) piston predictably reduced the necessary stroke and frequency relation to attain the required flow output. The significance of this change is minor with respect to compressor performance but will match the compressor to the engine at either lower frequency or lower stroke. This effect is exhibited in the performance measurement at design condition presented below. This factor permits generalization of the measured compressor performance to other units not having the heavy weight piston and allows comparison to original performance prediction.

The compressor was tested at cooling mode operating conditions of 95°F and 80°F ambient degree day design points. Functional operation in the heating mode at low ambient degree days with low suction pressure remains to be verified. A low suction pressure results in larger compressor piston strokes for similar loads on the engine. The lower suction pressure limit tolerated by the compressor without mechanical collision remains to be emperically verified.

Performance Tests

The Unit 3 compressor was tested at the 95°F ambient degree day design point conditions and the 80°F day conditions for a variety of operating strokes and frequencies. The regime of operation is graphically displayed on the compressor map in Figure 3-21. A summary of the ranges of component test conditions is presented in Table 3-3. The Unit 3 compressor was then assembled with the engine by the end of the reporting period and checkout testing initiated.

The component tests on the Unit 3 assembly achieved a major milestone with tests at the 95°F day design point. The test results in Table 3-4 indicates the unit meets almost all performance design requirements. Performance for two test points are given in the table with different dynamic inputs (strokes and frequencies) and near design operating pressures. The lower than original design operating frequency and stroke are a function of the heavy configuration selected. The results indicate that the unit will perform as predicted.

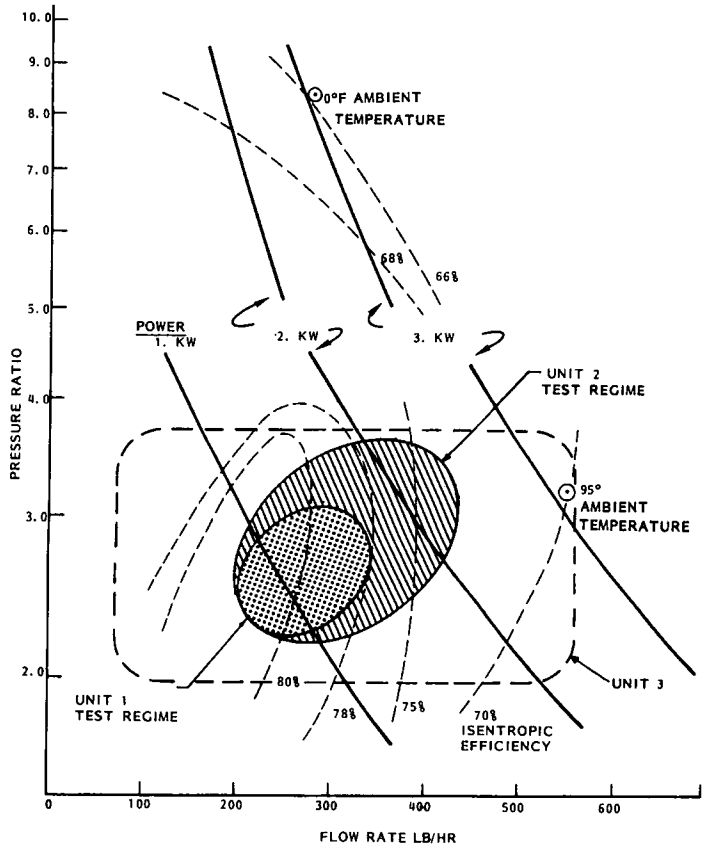


Figure 3-21. Proto 2 Compressor Test Results

Table 3-3. Ranges of Measured Compressor Performance in Component Tests

	95° Day	80° Day
Pressure Ratio	~ 3	~ 2
Power, kW	0.35 to 3.25	0.62 to 2.35
Flow, lb/hr	43 to 567	230 to 547
Gas Spring Loss, %	14 to 20	23.5 to 14.4
Efficiency, %	47 to 74	68.2 to 78.5
(W/O Gas Spring Loss), %	59 to 86	80.0 to 90
Housing Stroke, in	1.16 and 1.35	1.16 and 1.35
Freq., Hz	25.8 to 28.4	23.2 to 24.9

Table 3-4. Proto 2 Unit 3 Compressor 95° Day Performance

Parameter	Unit	Design Req'ts	Test a*	Test b*
Housing Stroke	inch	1.3	1.35	1.16
Frequency	Hz	30	27.5	28.4
Suction Pressure	psia	88	84	88.5
Discharge Pressure	psia	273	280	270
Suction Superheat, Upper	^{o}F	50	64	56
Lower	^{o}F	50	61	52
Pressure Ratio	--	3.10	3.36	3.06
Mass Flow,	lb/hr	530	522	567
Power, $\dot{M} \Delta h$ Total	kW	3.22	3.07	3.14
$\int Pdv$ Total	kW	--	3.47	3.25
Cylinders	kW	--	2.92	N/A
Gas Springs	kW	--	0.55 (16%)	N/A
Efficiency, Isentropic	%	70	71	69
Compressor Piston Stroke	inch	1.70	1.84	1.86

* a) 10/28/80, 15:53 Data Point
b) 11/14/80, 8:50 Data Point
N/A - Not Available

Slight performance degradation may occur during integrated engine-compressor operation due to the higher suction gas superheat temperature expected. The design superheat level of 50°F was reached in component test, and superheat is expected to be higher in integrated test due to heat transfer in the interface ring and tubular spring coils. The full impact of this heat transfer will be assessed in future integrated testing.

The Unit 3 compressor was also tested at the 80°F day operating condition. Test results are presented in Table 3-5. For this operating point, the heat pump system is designed to operate at half capacity to better match the required cooling load. The compressor thus operates at

1.6 kW, and a lower pressure ratio (2.0 versus 3.1). The two test conditions have negligible differences from prediction and have a measured efficiency 2 to 6 percent higher than predicted.

Table 3-5. Proto 2 Unit 3 Compressor 80° Day Performance

Parameter	Unit	Design Predictions	Test c	Test d
Housing	inch	1.0	1.35	1.16
Frequency	Hz	28.5	23.27	24.22
Suction Pressure	psia	95	89.5	96
Discharge Pressure	psia	195	197	195
Suction Superheat, Upper	°F	50	46	37
Lower	°F	50	45	40
Pressure Ratio	--	2.05	2.21	2.03
Mass Flow	lb/hr	400	422	415
Power, $\dot{M} \Delta h$ Total	kW	1.58	1.43	1.34
$\int Pdv$ Total	kW	--	1.55	1.59
Cylinders	kW	--	1.34	N/A
Gas Springs	kW	--	0.20 (13%)	N/A
Efficiency, Isentropic	%	72	78.5	74.5
Compressor Piston Stroke	inch	--	1.47	--

* c) 11/4/80, 9:22, Data Point
d) 11/14/80, 9:35, Data Point

N/A - Not Available

Tables 3-4 and 3-5 present several independent power measurements, in addition to the power measurement based on mass flow-enthalpy change measurement with a calculated heat transfer correction. The additional power measurements are derived by integration of pressure-volume dynamic oscilloscope traces of the four compressor gas volumes. There exists a discrepancy between the two measurements on the order of 3 to 18 percent for the test data presented. If this power measurement was used to determine operating efficiency, the 95°F day isentropic efficiency would be approximately 67% and the 85°F day value would be 72%.

Both methods of determining power are valid. Further testing is required to resolve the differences and reduce experimental error: use of the mass flow-enthadpy calculation will continue as the prime method until the discrepancies are resolved.

A factor minimizing the significance of the efficiency value discrepancies is the power actually consumed by the compressor, regardless of the efficiency. The test points closer in operation to the design prediction, 95°F point "b", and 80°F point "d", were at compressor power values within 1 percentof the predicted values. The power consumed, rather than the isentropic efficiency, is the more important of the two measurements in determining system performance (COP).

3.4.4 COMPRESSOR ANALYSIS

The analytic model and computer program used for predicting Proto-2 performance was modified and improved during this period. Several leakage paths out of the gas springs were added to the model to account for some of the gas spring power losses and produce a closer match to test data of compressor piston motion. The third piston ring was added to the model as was the leakage between gas springs through the connecting rod center porting and past the connecting rod seal. These latter changes are accounting for 50 to 75% of the leakage losses in the gas springs.

An integrated pressure-volume ($\int Pdv$) method to calculate work and subsequently power consumption by the compressor was added to the model. This technique permits closer correlation and cross checks between the dynamics and thermodynamics of the model and test results. The analysis and test data correlation is now conducted by comparing dynamic operation along with thermodynamic operation for a particular input stroke and frequency.

The present model prediction to test data correlation is shown in Figures 3-22 and 3-23 for the Proto-2 Unit 3 compressor tests. The mass flow versus operating frequency is shown in Figure 3-22 for 95°F and 80°F ambient day operation. The predicted mass flow is within 8 percent of measured value for all but one very low flow point. Design operation of the compressor for both the 95° and 80°F design is at a flow rate in excess of 400 lbs/hr, minimizing

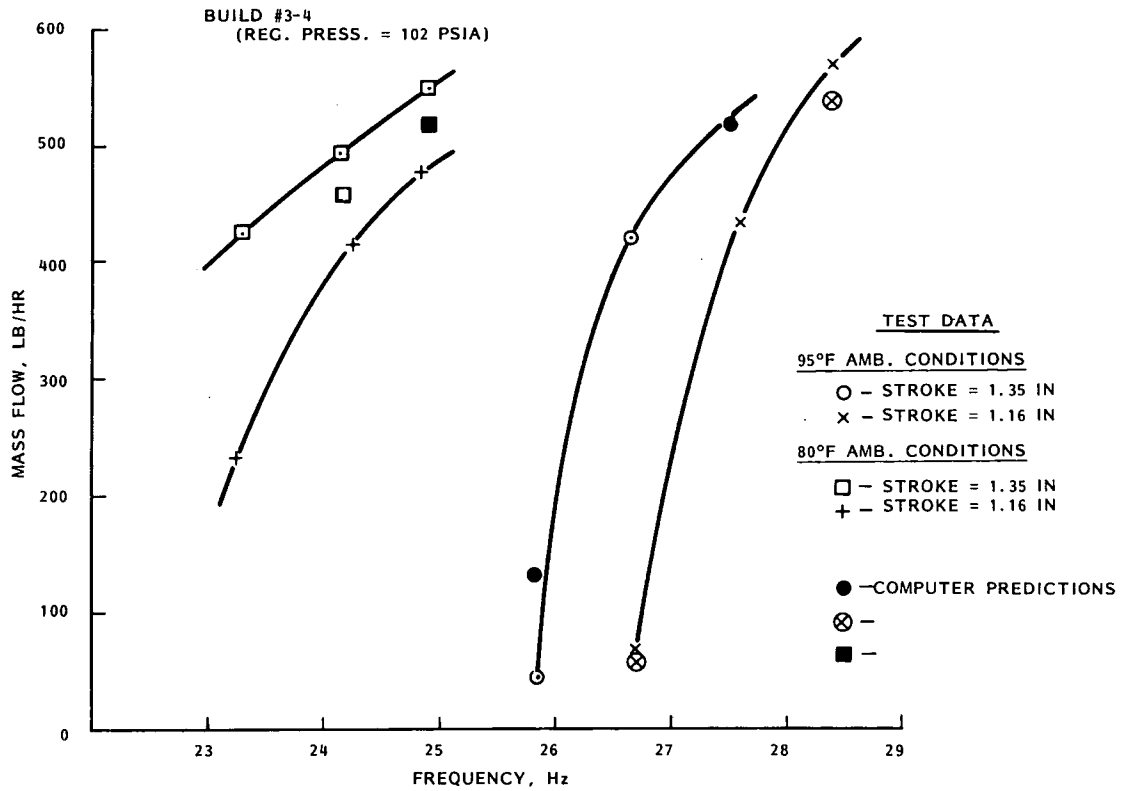


Figure 3-22. Compressor Model/Test Data Correlation for Flow Rate

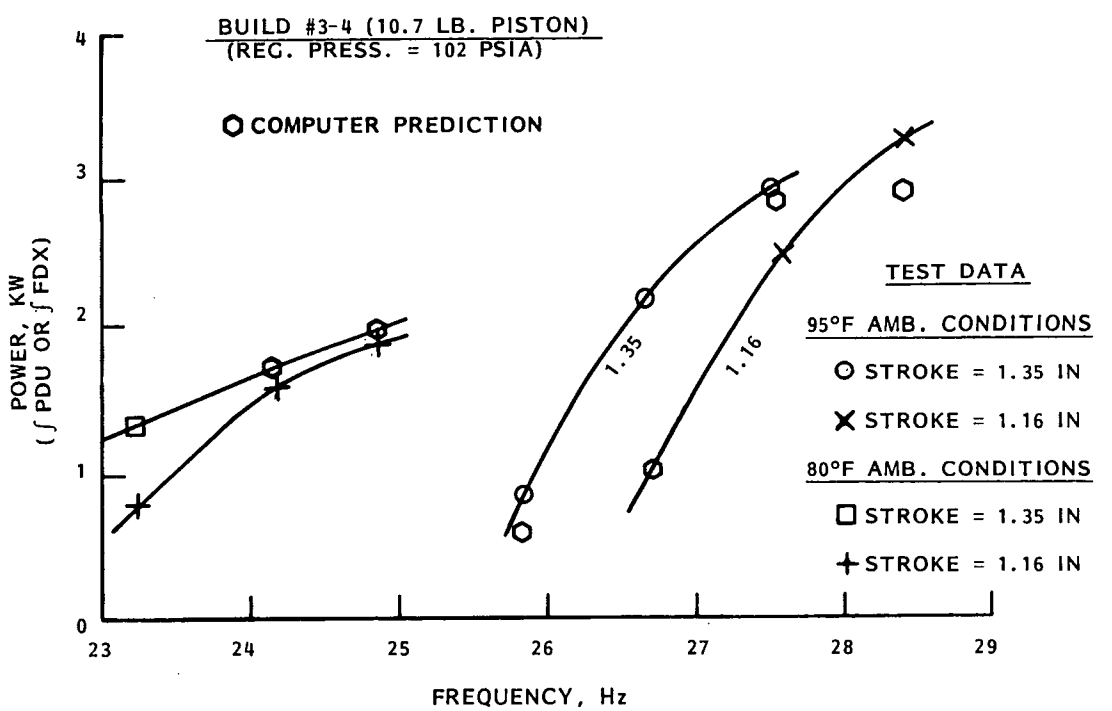


Figure 3-23. Compressor Model/Test Data Correlation for Power

the effect of the variation at lower flow rates. The power consumption of the compressor, and correlation to test data, is shown in Figure 3-23 for the same operating conditions. The analysis predictions are within 12 percent of test data with all but one significantly closer.

3.4.5 PLANNED ACTIVITY FOR NEXT PERIOD

Principal activity on the compressor in the next reporting period will be to support integrated engine/compressor testing of the modified Proto 2 engine, interpret compressor performance test data, and continue correlation of compressor models with test results. Tests with the modified engine will also be planned and results interpreted for various compressor pressure regulator settings. These tests will be used to determine dynamic response of the engine/compressor assembly to variations in compressor gas spring stiffness, varied by regulator pressure.

3.5 EXTENDED HEATING SUBSYSTEM

3.5.1 SUMMARY

The purpose of the extended heating capacity subsystem is to provide heat to the conditioned space during periods of time when it is not efficient to operate in the heat pump mode and/or supplemental heating is required to meet high load conditions.

During this reporting period an analysis was conducted to determine the most practical and cost effective means of providing both the type and level of supplemental heating. Subsequent to selecting a method, a preliminary hardware design and heat pump integration study was completed.

3.5.2 DESIGN REQUIREMENTS

The principal requirements of the extended heating subsystem is to provide adequate capacity to heat a structure utilizing a 3-ton HAHP at -20°F. Capacity studies of current typical homes and homes meeting ASHRAE 80 - 75 standards were conducted. The results are shown in Figure 3-24, for moderate (Philadelphia and Seattle), high (Boston) and severe (Bismark) heating climates. As shown, a capacity of 100 kBtu meets the most severe requirements.

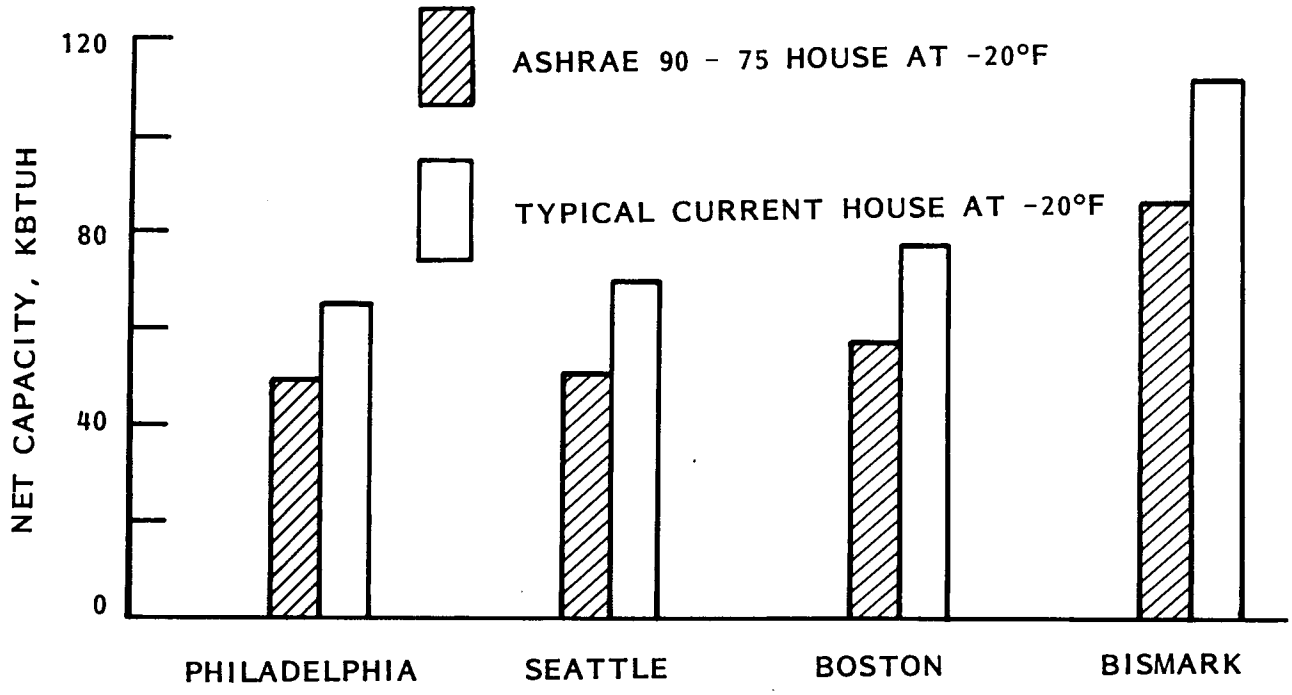


Figure 3-24. Heating Capacity Requirement for 3-ton HAHP

Since parallel assessments of preferred subsystem options indicated little volume or cost differences between units with 70 kBtu or 110 kBtu capacities, the 110 kBtu requirement was adopted. This will provide a product capacity capable of meeting most northern climate installations, a prime market for the HAHP.

3.5.3 SUBSYSTEM OPTIONS

A number of alternate concepts were evaluated for their potential cost, operating efficiency, compatibility with the HAHP product, and other design and operating features. The concepts evaluated are briefly described below, along with their principal advantages and disadvantages.

Modified GFHP Combustor

This concept is a compact natural gas fired boiler which uses HAHP type combustor. It consists of a porous felt metal burner along with a tube and fin heat exchanger. The unit would be located inside the outdoor air handler.

This concept would package well into outdoor cabinet (compact), have high efficiency, and could incorporate auxiliary defrost capability. However, it requires a separate water heat exchanger, uses a developmental combustor, and requires insulated water lines into the heated space.

Electric Resistance Unit

In this approach, an electrical resistance heater would be installed directly into the indoor air handler as is currently done in electric heat pumps. The advantages of this approach are that installation provisions exist in the indoor air handler and are a standard GE Co. option, and it has a low first cost. However, it is the least efficient of all the options and does not provide auxiliary defrost capability. It would not interface directly with the outdoor coil and hence could not defrost the coil.

Domestic Hot Water Heater

This system would utilize a fast recovery gas fired hot water heater that could supplement the space heating in addition to providing domestic hot water. One advantage of this system is that it could incorporate an auxiliary defrost circuit. However, it also requires a larger than normal hot water heater, a hydronic coil in air handler or duct, and another circulator pump.

Gas Duct Furnace

This system concept incorporates a gas fired duct furnace installed at a convenient location inside the living quarters. This approach does not require a hydronic coil, but can be cumbersome to install at an existing site. In addition, it requires separate combustion locations with required facilities, may need multiple sites because of duct differences, and has no auxiliary defrost capability.

Auxiliary Gas Hot Water Furnace

This concept would be a separate furance in the structure basement or service area. It would feature a simple installation and could provide auxiliary defrost capability. However, it also requires considerable space inside the living area, is less efficient than a HAHP combustor concept, and cannot be packaged inside the outdoor unit.

Gas Pulse Combustor

This system utilizes the pulse combustion technique to both improve combustion efficiency and reduce the furnace package size. The unit would be physically located inside the heated space in a typical furnace installation. The unit would have a low exhaust temperature, and could use plastic flue pipe, operate with high efficiency (over 90 percent), and will be available soon.

However, this high efficiency feature is not very important, due to the low number of operating hours per year. In addition, there would be a potential noise problem during operation and first cost would be relatively high.

3.5.4 SELECTED CONCEPT

The HAHP combustor concept was selected because of its high efficiency, small size and proven technology base. Several heat pump/auxiliary heater system configurations which incorporate the HAHP combustor heat source were analyzed in depth and a selection was made based upon performance and system cost and reliability.

Figure 3-25 schematically illustrates the selected configuration incorporating a dual speed coolant pump. The lower flowrate of 2.2 gpm is required for engine cooling during heat pump operation. The higher flowrate, 11 gpm, is employed when the extended heating system is required and includes the engine cooling function. As indicated, a solenoid operated shut-off valve is used to isolate coolant flow from the heater loop during normal heat pump operation.

During extended heating operation, the higher flowrate is appropriately divided between the heater and engine cooler by the pressure drop ratio of the two heat exchangers. The heated fluid from both sources are subsequently mixed and are available for space conditioning via a 3-way valve and a separate indoor unit heat exchanger. Flow exiting this heat exchanger is then collected in the combination reservoir and pump head tank.

The three-way valve is included in the loop to permit selective routing of the hot fluid to a defrost supply exchanger in the outdoor unit. This permits deficiency of the refrigerant coils without refrigerant flow reversal.

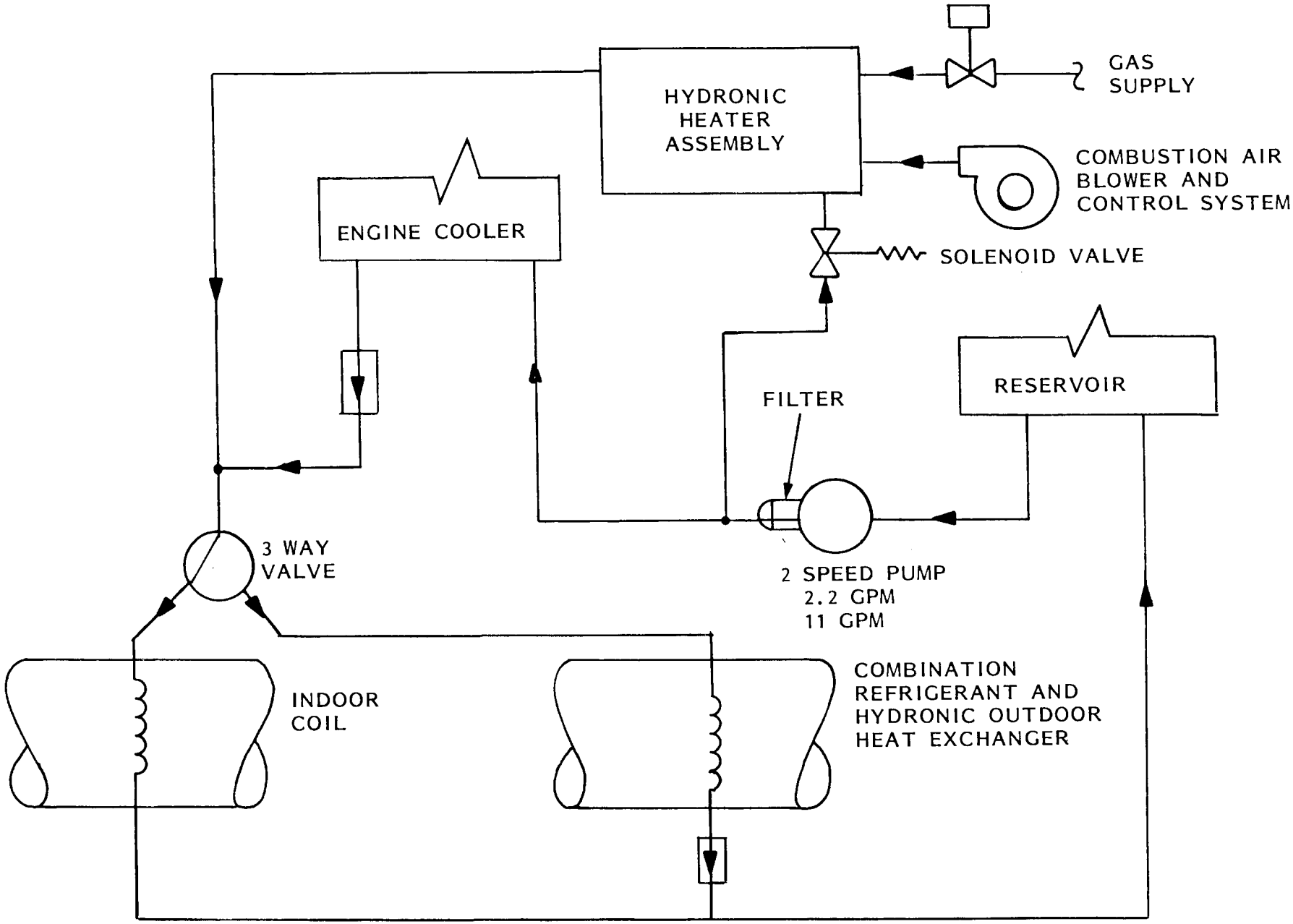


Figure 3-25. Selected Heat Pump/Auxiliary Heater System Configuration

The hydronic heater assembly is illustrated in Figure 3-26. This unit is envisioned as a separate subsystem that is packaged for installation within the HAHP outdoor units heat exchanger enclosure. It will contain an independent combustion air blower, gas supply system and the required operational and safety controls. Connections to the outdoor unit will be through the hydronic loop, gas manifold and operational controls only.

Status

A preliminary engineering design of the gas fired hydronic heater has been completed. Many of the detailed designs have been finished and detail drawings completed. Further development efforts were deferred after completion of this portion of the work to concentrate on higher priority engine developments.

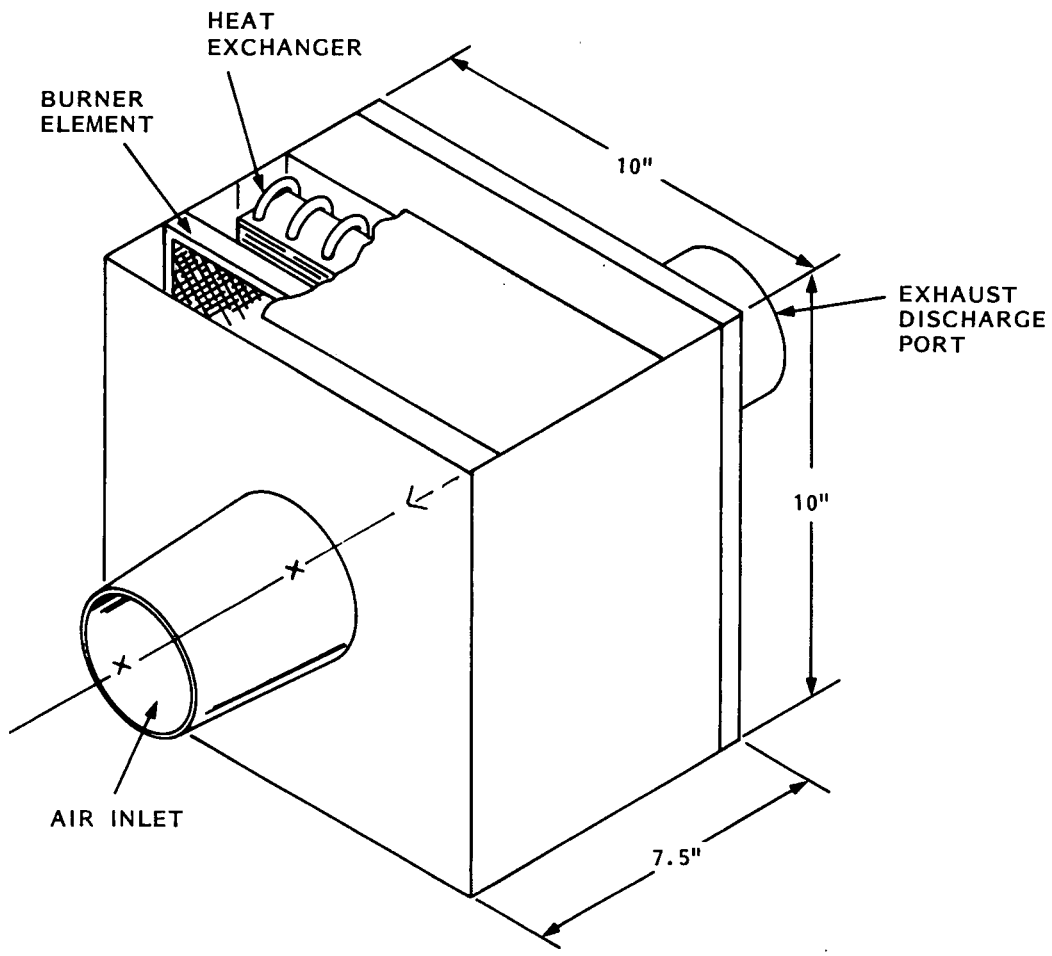


Figure 3-26. HAHP Extended Heating Capacity Subsystem Concept

SECTION 4

INTEGRATED TEST

SECTION 4

INTEGRATED TEST

4.1 SUMMARY

The Integrated Test effort during this reporting period continued with the assembly and testing of multiple engine hardware configurations to reduce second gas spring losses and tune dynamic components for optimum performance. In addition, special tests were conducted to evaluate the effects of reducing the engine housing vibration amplitude and to reduce power piston seal leakage. Section 3 of this report provides detailed descriptions of the hardware variations tested. The effects of different operating conditions were also investigated. Reference is made to Section 2 of this report for a discussion of the engine performance results versus build configurations/operating conditions.

4.2 BASELINE PROTO 2 ENGINE TESTS

Testing continued this period on the original Proto 2 combuster/engine/compressor assembly (Figure 4-1) to improve performance and investigate hardware changes. A number of areas were investigated in support of both tuning and diagnostic evaluations of the Proto 2 engine.

4.2.1 SECOND GAS SPRING MODIFICATIONS

Initial testing of the Proto 2 FPSE identified the need for increasing the second gas spring dynamic-centering vent port area and/or effective length to "unload" the gas spring at its mid-stroke position. While engine start-up and piston stroke characteristics were improved, excessive gas spring power dissipation was also evident. Second gas spring development was subsequently initiated to reduce power losses due to throttling during the vent portion of the piston stroke and/or leakage past the seals.

Specific areas of investigation included vent port area/hole pattern and piston leading and trailing edge chamfers regarding their effects on throttling losses, and piston seal rings and their installation as they affect leakage. Seal ring changes evaluated included an improved step joint overlap and an attempt to reduce the leakage around the back of the piston rings by the addition of O-rings.

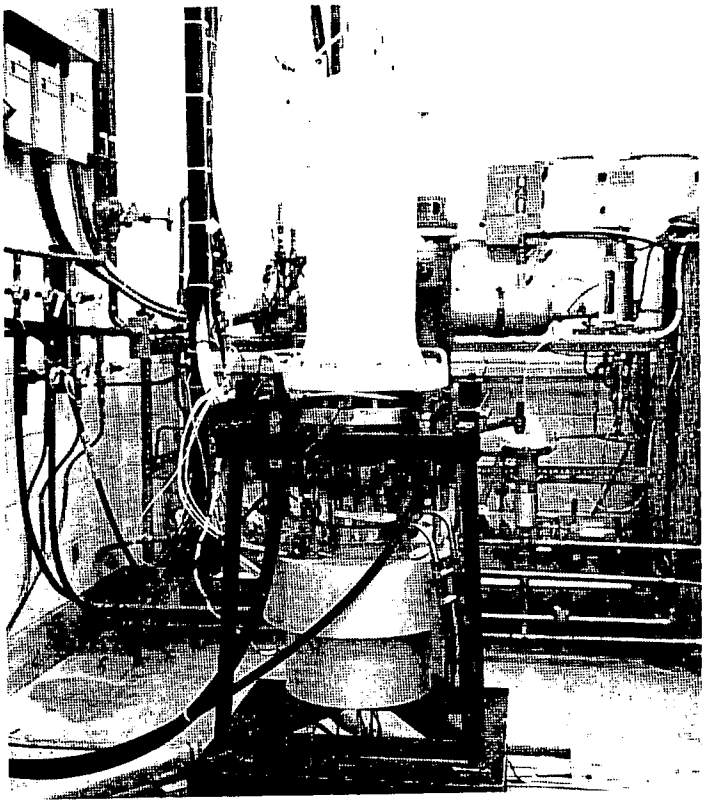


Figure 4-1. Proto 2 Combustor/ Engine/ Compressor Integrated Tests

Limited success was obtained in reducing the second gas spring power dissipation, especially with chamfering of the gas spring piston. This change resulted in losses reduced by up to 18 percent. Neither of the seal modifications made measurable changes in the second gas spring leakage characteristics. Section 2 of this report presents a discussion of these test results.

4.2.2 DISPLACER GAS SPRING TUNING

Dynamics tuning of the FPSE is significantly effected by changes in the displacer gas spring characteristics. Gas spring volume, displacer rod (piston) area, and centering port axial location are all variable parameters in the Proto 2 engine, and all were varied during this reporting period for tuning purposes. Displacer stroke, dynamic center, and displacer to power piston phase angle were all effectively tuned by selectively changing these parameters. Different gas spring configurations implemented included displacer rod areas of 1.48 and 1.58 in.^{2}, volumes of approximately 10.5, 11.5, and 12.5 in.^{3}, and several centering port locations. Although some performance improvements were noted, results of these tuning

tests confirmed that the basic tuned configuration of the baseline engine was delivering its maximum performance potential.

4.2.3 WEIGHTED HOUSING TEST

A weighted housing test was conducted to determine the effects of engine housing vibration amplitude on unit performance. The engine housing weight was doubled to reduce the engine housing to ground operating stroke by one-half. This was accomplished by attaching 276 pounds of steel bars around the periphery of the pressure vessel, and supporting the increased weight by doubling the number of housing to ground isolation springs. See Figure 3-9 for a photograph of this assembly. No significant performance changes were measured with this configuration, other than the engine housing amplitude being reduced by approximately 1/2, as planned.

4.2.4 HEATER HEAD TUBE FAILURE

Further testing of the Proto 2 engine incorporating the 80% porosity regenerator matricies was prevented by the development of hairline cracks in two of the heater head tubes. A failure analysis was conducted and the results are presented in Section 3.3 of this report.

4.3 RETROFIT PROTO 2 ENGINE ASSEMBLY

The retrofit Proto 2 engine assembly was installed in the Integrated Test Stand near the end of this reporting period, and the check out phase of the test effort was started. The initial engine start-up attempt was successful and tests were run at both 80°F and 95°F ambient day operating conditions. The untuned engine performance was low with limited power piston and displacer strokes of approximately 1.0 inch and a corresponding phase angle of 68 degrees. Dynamic tuning by means of displacer changes was found necessary to increase the piston and displacer strokes by 25% to 30%, and to reduce the phase angle.

Three displacer configurations were subsequently tested, and venting of the second gas spring was also tried. The third change provided a marked improvement. This configuration used the G-4 (largest) displacer rod, and had the displacer gas spring volume increased by approximately 16%. Improved engine operation was demonstrated by increased power piston

and displacer strokes, a reduced phase angle, and an improved heat absorption capability by the engine heater head. The data from this test is currently being reduced to determine the overall engine performance.

4.4 PLANNED ACTIVITY FOR NEXT PERIOD

The planned activity for next period is to tune the retrofit Proto 2 engine assembly for optimum performance and then performance map the final configuration. The combustor/engine/compressor assembly will then be installed in the previously tested outdoor unit air handler and controls package in the Systems Test Facility. System testing will be conducted over a wide range of ambient heating and cooling conditions, simulating operation at the Field Test site.