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

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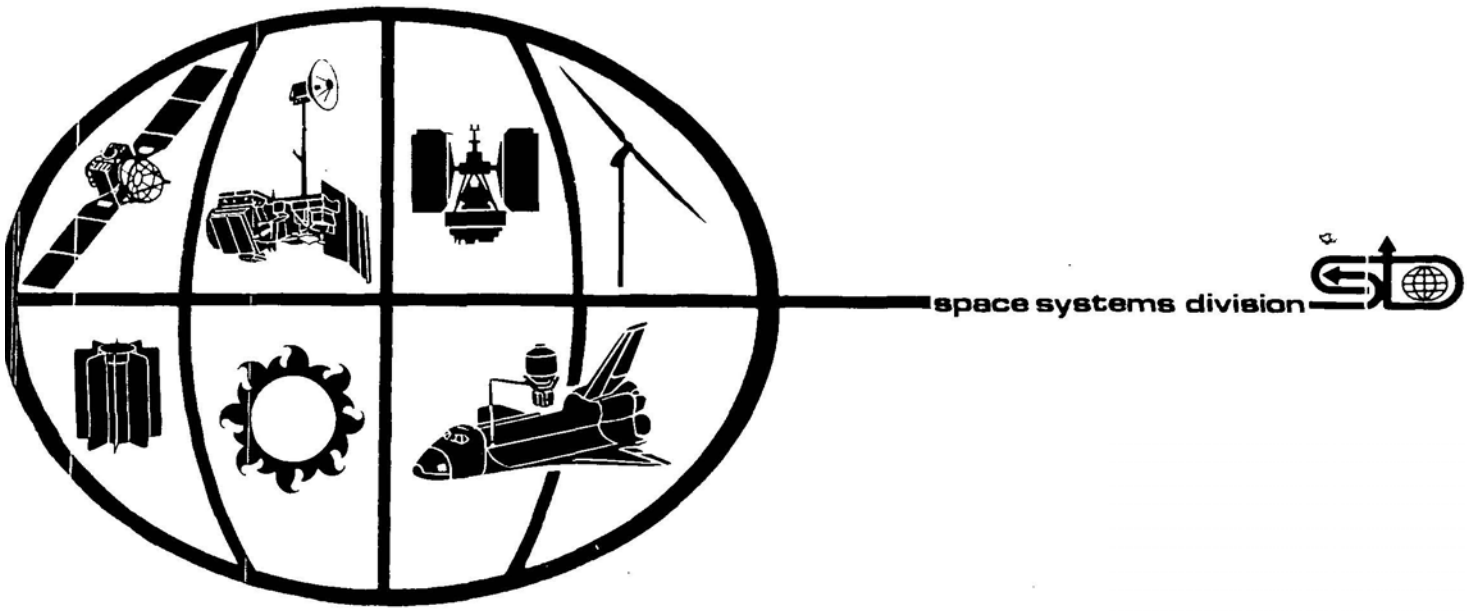
NO. C00/2911-7

JANUARY THROUGH JUNE 1980



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GENERAL ELECTRIC
SPACE DIVISION
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 I 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 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.

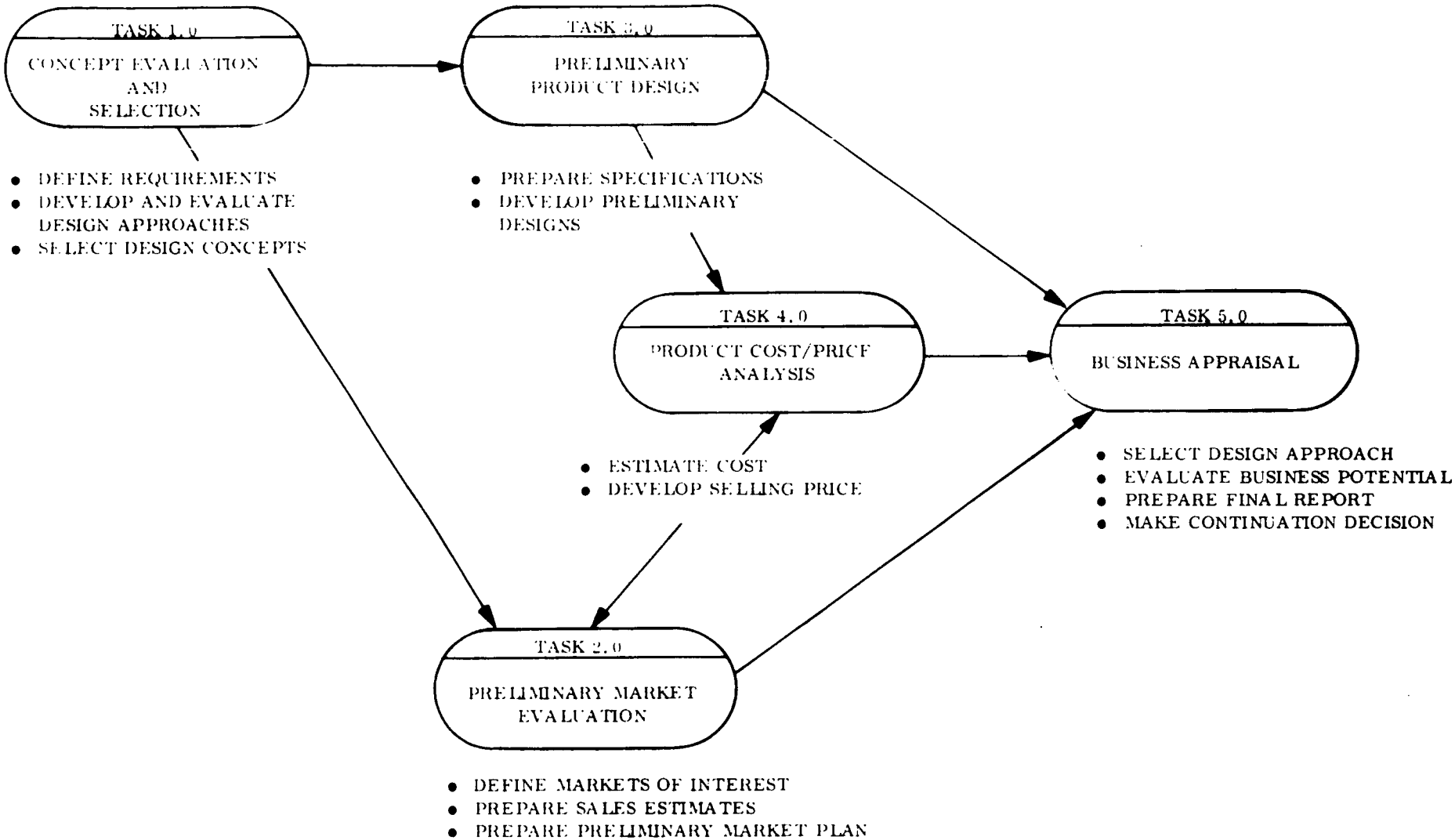


Figure 1-1. Phase I Flow Diagram

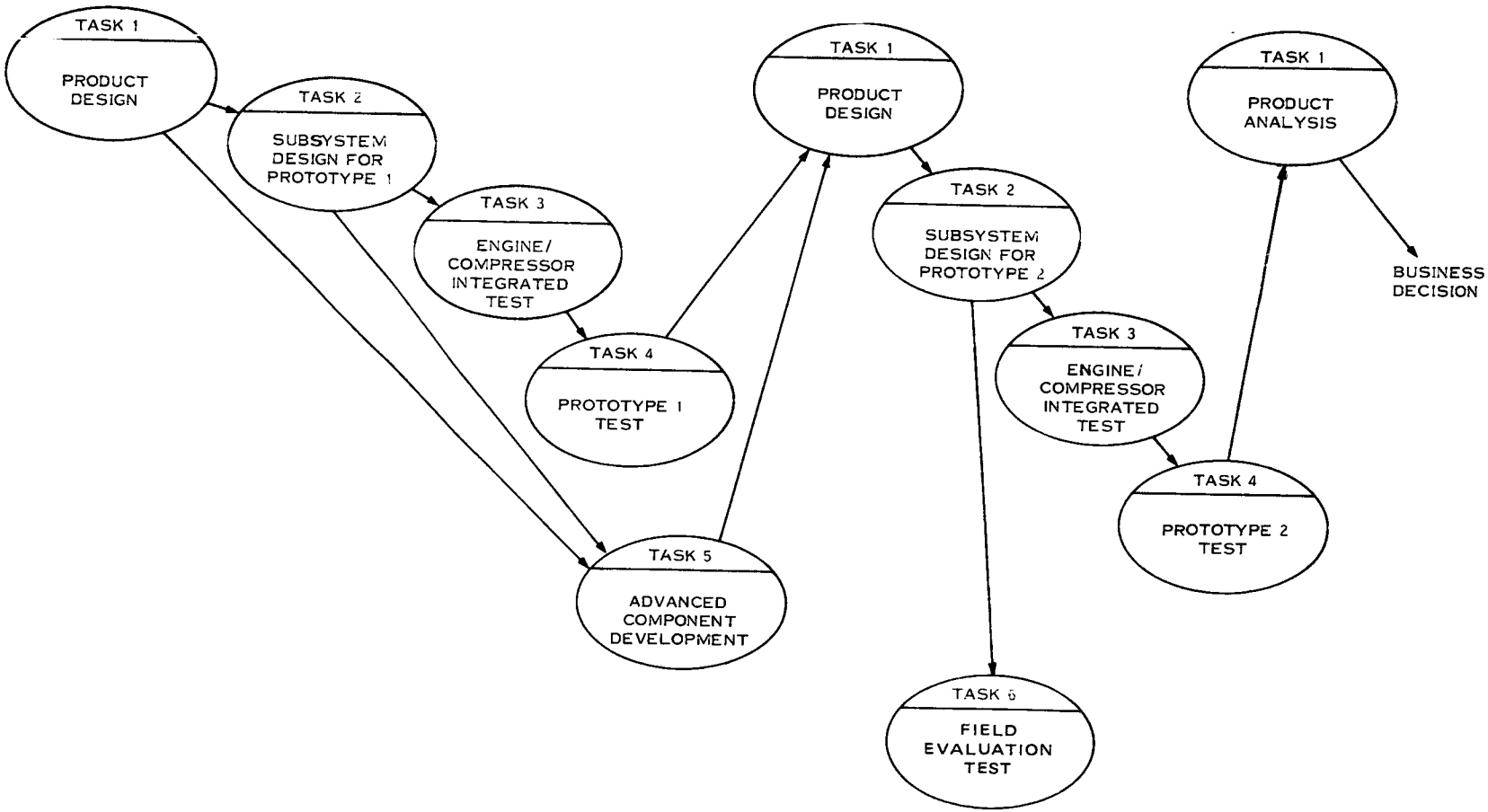


Figure 1-2. Phase II Flow Diagram

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 45°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 1500 watts. Based on these performance levels, the following performance goals were assigned to the major subsystems:

- Stirling Engine Cycle 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, product design effort was directed at the following key areas: (1) Proto 2 operating cost savings estimates, (2) HAHP eventual product savings estimates, (3) Simulation code improvements for engine testing.

In order to project a probable range of operating cost savings, a forecast of fuel prices was made based on available forecasts from other sources and the most recent developments relating to the sharp price escalations and curtailment of supplies during portions of 1979. The resulting fuel price forecasts continued to track well with actual experience as the reporting period progressed, and were adopted for use in subsequent calculations. Combining HAHP performance parameters, Nearest Equivalent Product (NEP) performance, and the fuel price forecasts for time periods of interest enabled operating cost savings for the HAHP to be estimated. Results of these projections continued to show economic viability of the HAHP for reasonable payback periods ranging from three to seven years depending upon geographic region and product manufacturing and distribution cost factors. These results, along with other market study data, were presented to Gas Research Institute and Department of Energy personnel near the end of the reporting period.

The computer simulation codes were improved in several respects. The TDP code was expanded to account for thermal losses and dynamic housing motion. The compressor model was made more realistic by using an analysis of independent control volumes rather than empirical correction factors.

Combustor

During the previous reporting period, component testing of the Proto 2 Unit 1 combustor was completed and projected system level combustor performance determined. The combustion air blower flow control damper design was finalized, and testing of a fabricated unit was completed. The testing of the White-Rodgers two stage gas value was also completed successfully. Combustor performance met all design requirements after minor hardware modifications.

Engine

The Proto 2 engine hardware was approximately 95 percent fabricated by the end of the previous reporting period, with one complete set of engine parts being assembled and installed in the Integrated Test facility. Instrumentation wiring harnesses were also completed except for the compressor leads.

SECTION 2

PRODUCT DESIGN

SECTION 2

PRODUCT DESIGN

2.1 INTRODUCTION

Integrated testing of the combined engine/compressor/combustor assembly for the Proto 2 HAHP was initiated during this reporting period, with the primary analytic focus placed on characterizing and improving subsystem performance. Product Design emphasis consisted of engine/compressor/combustor data reduction and interpretation, assessment of potential performance improvement with various hardware modifications, and improvement of analytical tools.

2.2 ENGINE/COMPRESSOR/COMBUSTOR PERFORMANCE

2.2.1 PERFORMANCE CHARACTERISTICS

The engine was started and run successfully in January 1980. Substantial performance improvements have been achieved since initial tests, through engine tuning, with some further improvements expected. Modified heat exchangers offer a potential for approaching design goal performance, and have been specified for fabrication. The engine second gas spring has been found to dissipate more power than originally expected. Various modifications are being evaluated to circumvent this problem.

Performance of principal configurations is shown in Figure 2-1 in terms of indicated power and efficiency. Indicated performance differs from shaft output by the second gas spring power dissipation (to be described later in this report) and by other minor losses. The data points have been normalized for constant heater tube and coolant inlet temperature. Details of the configuration and measured performance parameters, before corrections for constant heater and coolant temperature, are included in Table 2-1. Some of the performance gains were not continuous with time, especially in diagnostic testing periods when the second gas spring characteristics were being investigated (e.g., Builds 1-23 and 1-26). It should also be noted that further gains are achievable with the existing configuration, i.e., using a lightweight displacer and compressor unit number 2 in the Build 1-19 configuration, plus second gas spring optimization, etc.

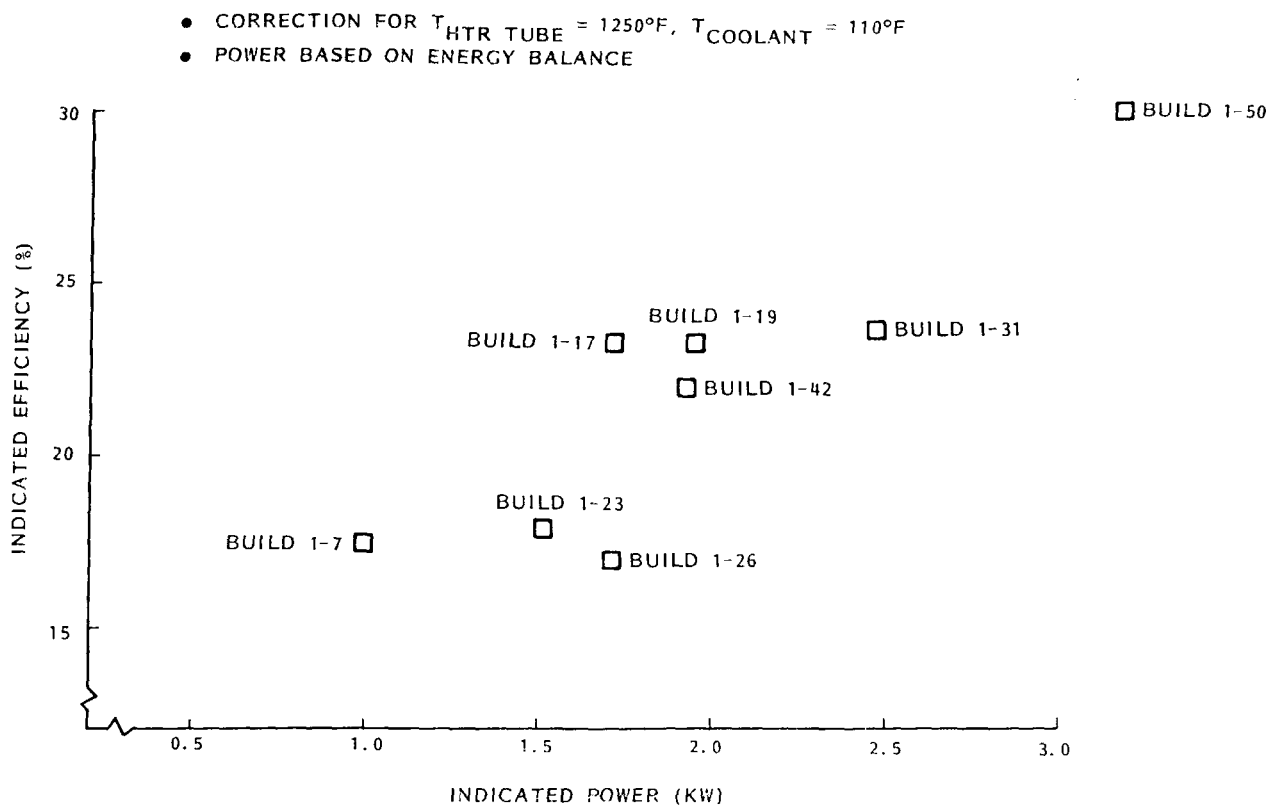


Figure 2-1. Performance Improvements in Proto 2 Engine

Data Reduction

Indicated power is currently calculated using two independent approaches:

1. Energy balance approach (primary approach): ($\dot{m}\,\Delta h$ Enthalpy gain of the refrigerant at the pressure vessel interface) + (pressure vessel heat losses to ambient) + (second gas spring heat losses to the coolant) - (heat conduction from the engine working space to the bounce space).

2. Summation of Fds on the power piston using a Fourier analyzer (secondary approach). This consists of reducing the dynamic pressure and displacement test data and calculating the force transmitted to the power piston from the compression space and the displacer gas spring.

There are some differences between the two calculations which are currently being investigated. Two of the differences noted so far are that larger refrigerant flow rates usually result in higher calculated power using the energy balance approach. In addition, as the engine heats up during a test, the power calculated using the energy balance occasionally increases

Table 2-1. Best Performing Configurations

Build No.:		1-19	1-42	1-31	1-23	1-17	1-26	1-7	1-50
Date:		3-14-80	7-2-80	4-29-80	4-2-80	2-29-80	4-11-80	2-6-80	7-17-80
Time:		15:53	11:38	14:53	9:22	14:17	10:30	16:08	9:32
Data Point:		#9	#5	#4	#3	#3	#2	#4	#3
Description:									
Regen. Porosity (%)		80	80	80	80	70	80	70	80
Second Gas Spring Area (in^2)		14	11.6	14.0	14.0	14.0	14.0	14.0	11.6
Second Gas Spring Vent Pattern		2 x 48 @1/4"	2 x 36 @ 1/8" & 2 x 12 @ 1/4"	Slots 1/8" → 1/4"	2 x 48 @ 1/8" & 2 x 48 @ 1/4"	2 x 48 @ 1/4"	Slots 1/8" → 1/4"	2 x 32 @1/4"	2 x 54 @ 1/8" & 2 x 18 @ 0
Displacer Weight (lb.)		2.1	1.76	1.76	--	2.3	2.1	--	--
Displacer Gas Spring Vol (in^3)		11	10.2	13	--	11	11	--	--
Rod Group		G-3	G-3	G-3	G-3	G-3	G-3	G-2	G-4
Compressor Unit No.		1	2	1	1	1	1	1	2
Q Into Engine (kBtuh)		28.4	29.6	36.0	29.4	24.4	34.8	16.9	~ 34.0
Heater Tube Temp. (^oF)		1215	1278	1103	1210	993	1149	1035	1147
Charge Pressure (psia)		1175	885	1175	895	1110	1175	1115	1005
Compressor Suction (psia)		75	101	104	88	121	105	147	99
Compressor Discharge (psia)		153	200	271	211	266	233	241	214
Pressure Ratio		2.04	1.98	2.61	2.40	2.20	2.22	1.64	2.16
Operating Frequency (Hz)		25.8	26.1	30.4	26.4	29.5	28.8	28.3	28.6
Piston Stroke/Housing (in)		0.81	1.23	1.12	1.25	1.05	1.05	0.89	1.12
Piston Stoke/Ground (in)		1.72	1.09	1.03	1.10	0.96	0.92	0.78	0.99
Displacer Stroke/Housing (in)		1.26	1.60	1.47	1.67	1.20	1.37	1.18	--
Phase Angle (degrees)		46	49	48	39	40	44	31	--
Refrigerant Flow (lb/hr)		268	357	241	209	--	184	--	424
Indicated Power (kW)	Fourier Analysis	1.50	1.73	1.76	1.75	1.40	1.60	0.76 (hand)	2.15
	Energy Balance Analysis	2.16	2.32	2.31	1.67	No-Temp. Data	1.74	No-Temp. Data	3.02
Indicated Efficiency	Fourier Analysis	18.0	19.9	16.7	20.3	20.0	15.7	15.0	19.7
	Energy Balance Analysis	26.0	26.7	21.9	19.4	--	17.1	--	30.0
Compressor Power (kW) $\dot{m}$ Δh at Pressure Vessel		1.54	1.69	1.58	1.19	--	1.03	--	2.36
Second Gas Spring Power (kW) - Fourier Anal.		0.45	0.67	0.86	0.93	--	0.85	--	0.77
Coolant Inlet Temp. (^oF)		70.5	72.9	68.5	75.0	38.0	67.0	50.0	63.0
Estimated Net Power (kW)		1.71	1.65	1.45	0.74	--	0.89	--	2.2

more than indicated by the Fourier analyzer. At the moment, the energy balance power calculation is preferred because it appears to be consistent with a third independent calculation, compressor power. This third independent calculation of compressor power is based on an oscilloscope trace of compressor pressures versus power piston motion. Integrating the enclosed area provides the gross power transmitted to the compressor from the power piston. This value agrees with the indicated power calculated from the energy balance (less the second gas spring power).

Performance Trends

The impact of various parameters on performance is described as follows. Figures 2-2 and 2-3 show that performance decreases with increasing cold side temperature and increases with higher hot side temperature. The sensitivity to these temperatures is somewhat larger than that expected based on a purely Carnot efficiency argument. This data and detailed TDP computer performance simulations, indicate the potential benefits of the more effective heat exchangers currently being fabricated for additional testing.

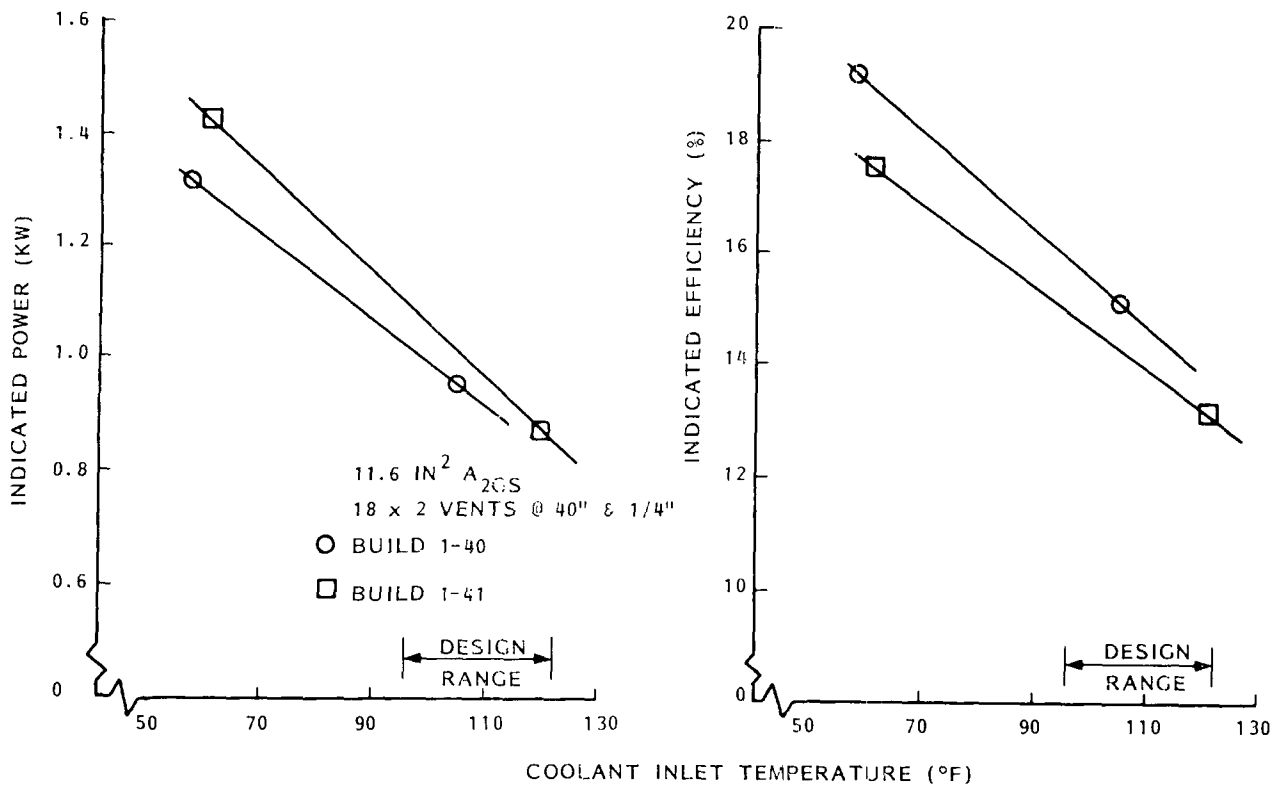


Figure 2-2. Engine Coolant Temperature Effect

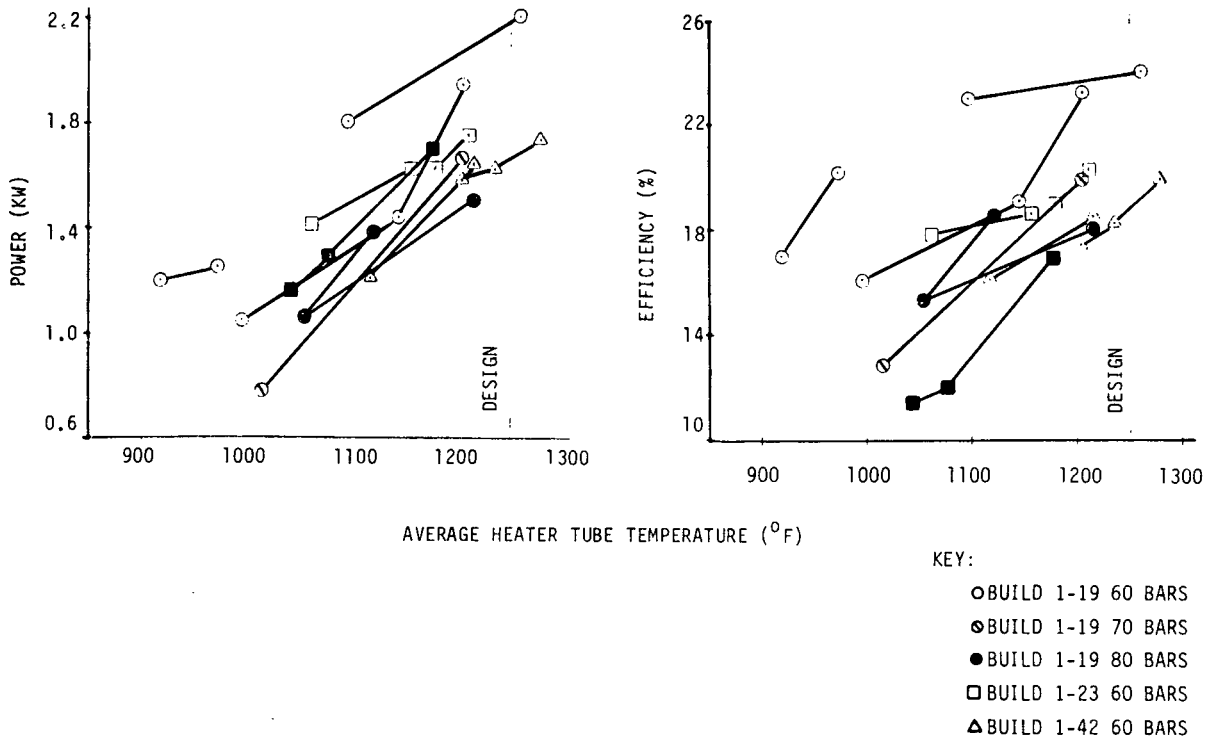


Figure 2-3. Heater Tube Temperature Effect

Figure 2-4 shows the importance of engine void volume (volume not swept by the power piston or displacer). A small increase in void volume leads to a large reduction in both indicated power and efficiency. In the redesign of the engine heat exchangers, attention to the void volume was critical; a balance must be maintained between more effective heat exchangers and the usually corresponding increase in void volume.

The effect of regenerator porosity is shown in Figure 2-5. The 80 percent porosity configuration provided smaller pumping losses and larger strokes, and thus larger power capacity. Results using detailed TDP simulation indicate that larger porosities result in lower regenerator effectiveness and that porosities in the range of 80 percent provide the best overall performance. As a result of these findings, heat exchanger modifications will use 80 percent porosity regenerators.

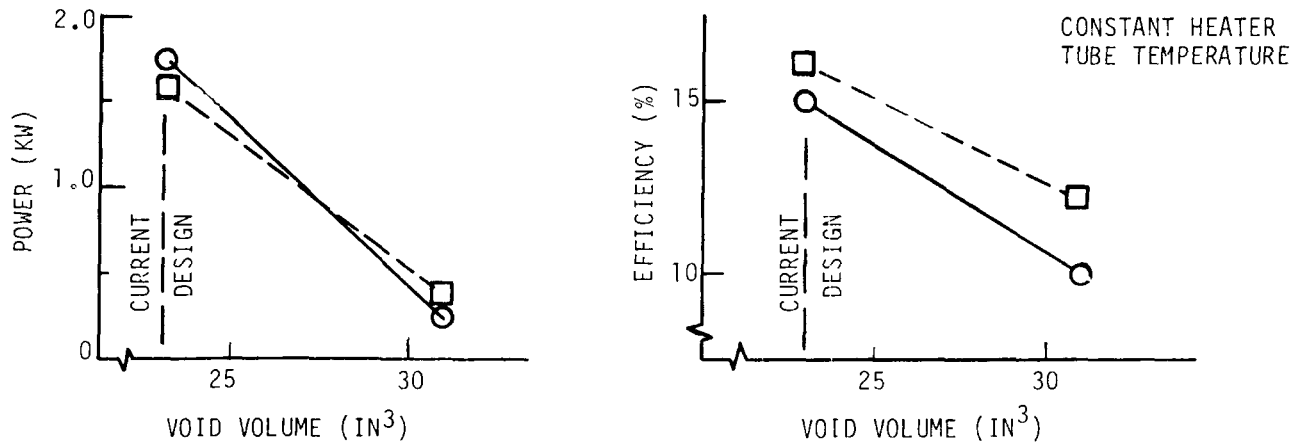


Figure 2-4. Void Volume Effect

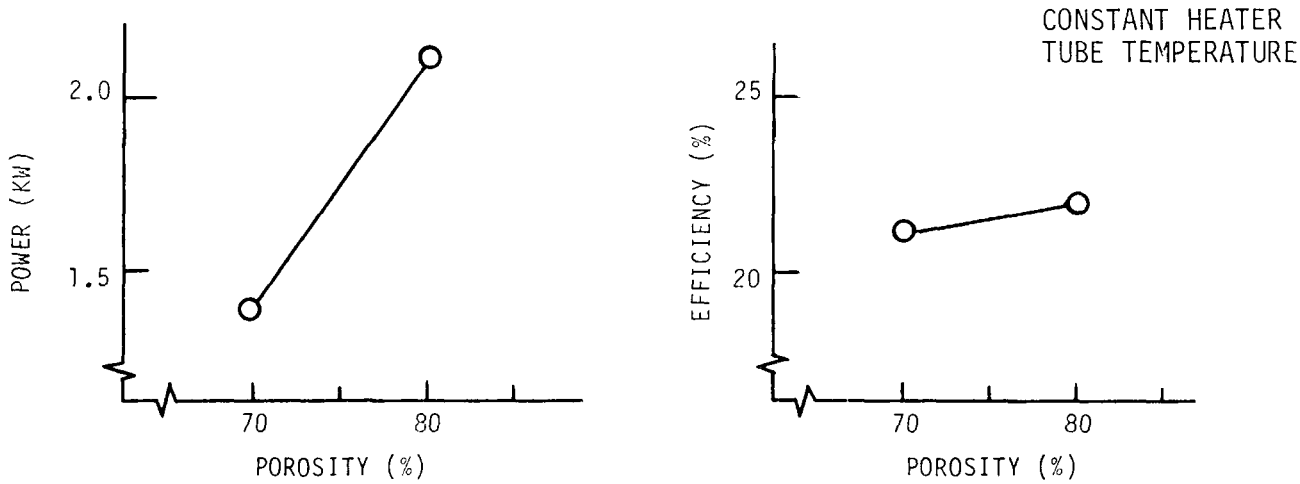


Figure 2-5. Regenerator Porosity Effect

Reduction of displacer weight resulted in better power and efficiency, as shown in Figure 2-6. This is primarily a result of increased displacer stroke which increases gas shuttling and thus heat input to the engine at constant heater tube temperature. Further weight reductions are possible in future engine designs.

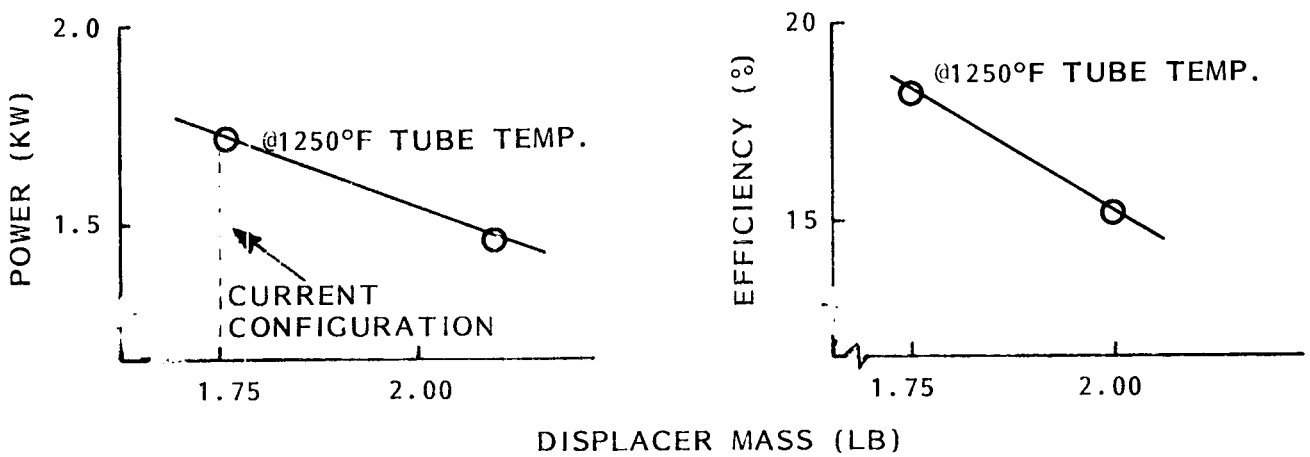


Figure 2-6. Lightweight Displacer Effect

2.2.2 SECOND GAS SPRING PERFORMANCE

Functions of the second gas spring (2GS) are: (1) to increase engine operating frequency in order to increase specific power, (2) to provide piston overstroke protection during transient periods such as startup, (3) to reduce compression space gas leakage and (4) to provide piston motion centering. Side effects determined through testing are that starting is inhibited due to the forces built up in the 2GS, and that the 2GS dissipates a significant amount of power.

Details of the 2GS are shown in Figure 2-7. It essentially consists of a piston and an enclosed volume of gas. The 2GS piston moves as an integral part of the power piston to compress and expand the gas. The inner set of vent holes were originally added to bypass gas around the piston to the bounce space (only at midstroke) to equalize the pressures and thus provide piston centering. The outer set of vent holes were added later to facilitate startup by minimizing pressure buildup during a portion of the piston mid-stroke. They also provide piston centering previously provided only by the inner holes.

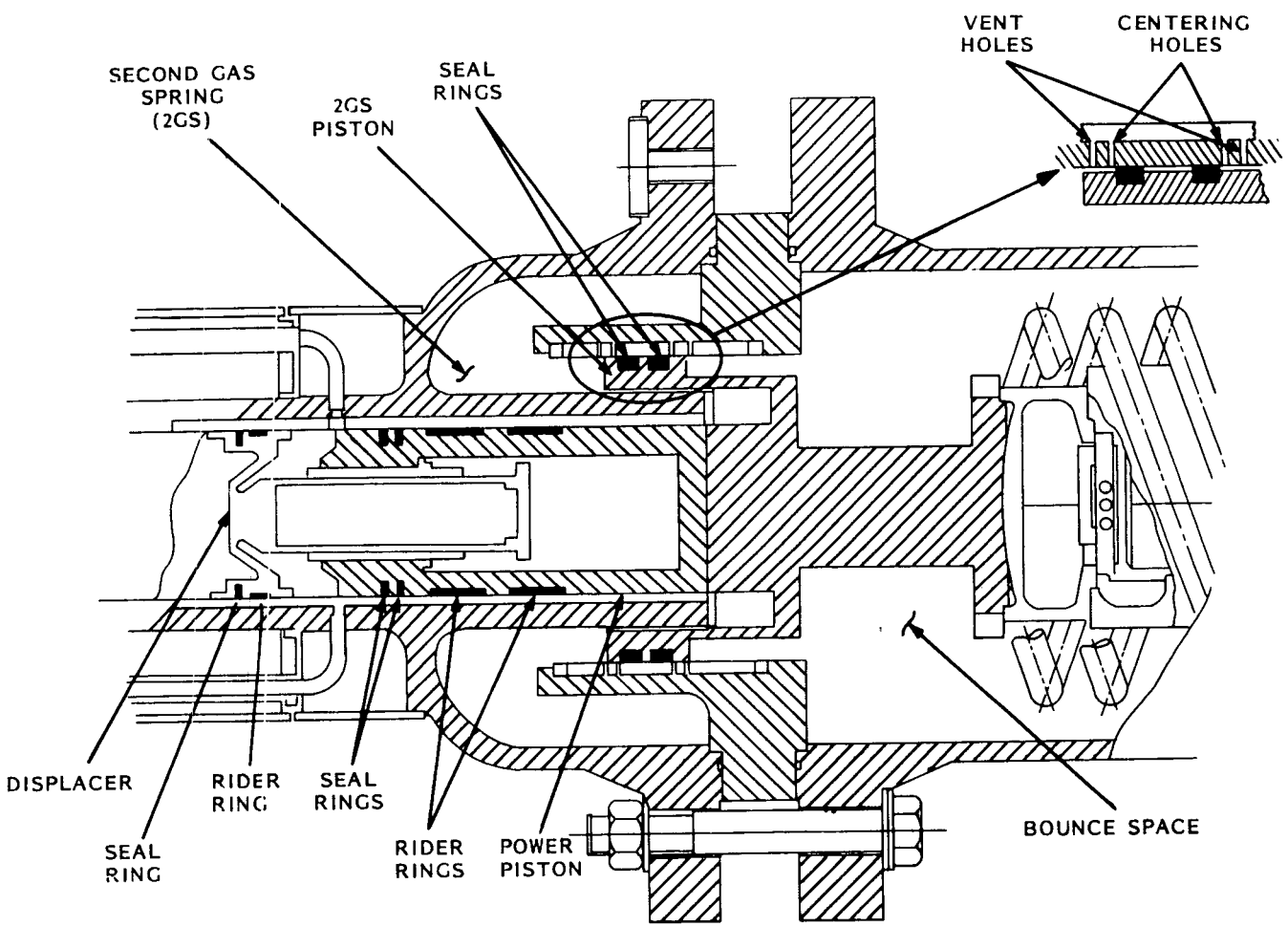
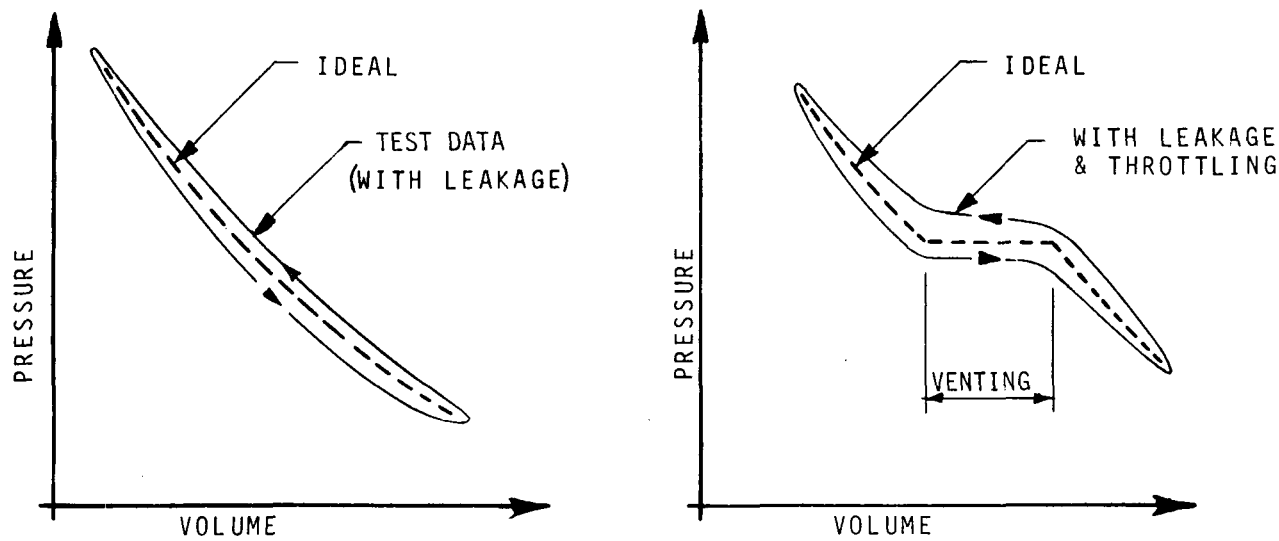


Figure 2-7. Proto 2 Second Gas Spring Design

Pressure versus displacement diagrams for the 2GS are shown in Figure 2-8. Under ideal conditions the dashed line in Figure 2-8A, would be expected for a complete cycle. Test data indicated the outer solid line implying the presence of leakage, with the enclosed area representing power dissipation. Similar characteristics were found with additional venting during the piston mid-stroke as shown in Figure 2-8B. In this case, power dissipation is attributed to both leakage and venting (throttling) at mid-stroke.



A. Without Venting

B. With Mid-Stroke Venting

Figure 2-8. P-V Diagrams for Second Gas Spring

Power dissipation is related to piston stroke, operating frequency and charge pressure. Measured values shown in Figure 2-9 are too high. Power losses in the range of 5 percent of indicated power, approximately 200W, are considered acceptable based on other Stirling engine designs. Two piston sizes for the 2GS are shown: the solid lines represent the larger piston area and show greater sensitivity to vent area than the individual points representing the smaller piston area. This has been found to be attributed to flow resistance between the leading edges of the piston and its liner. This problem will be corrected and should further reduce power dissipation by the second gas spring. Other areas are also being investigated to further reduce the second gas spring power dissipation.

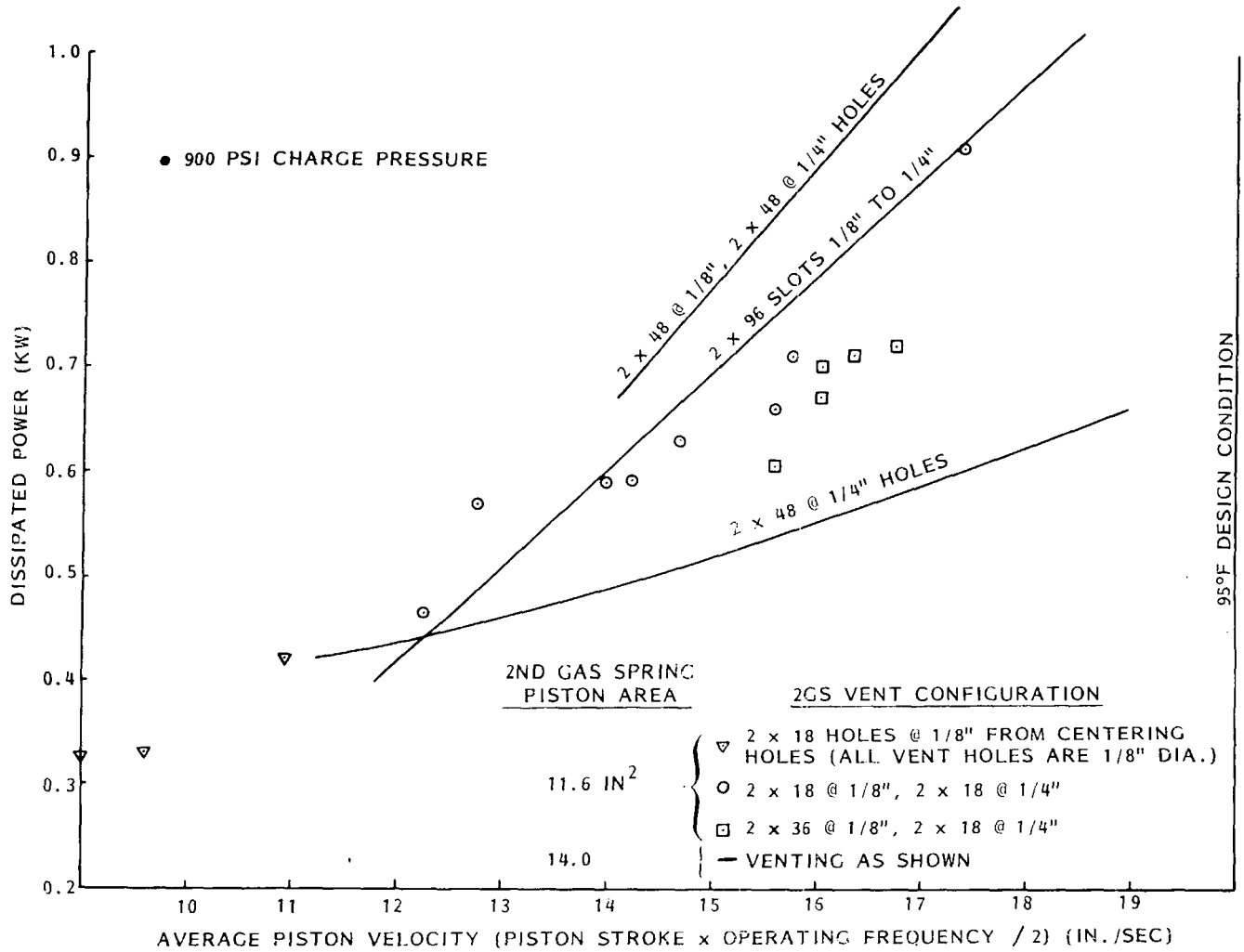


Figure 2-9. Second Gas Spring Power Dissipation

2.2.3 OPTIONS FOR PERFORMANCE IMPROVEMENTS WITH BASELINE ENGINE

Performance achieved with the major hardware builds was shown previously (Figure 2-1). A majority of the tuning changes were made one at a time in order to characterize parametric trends, and thus certain combinations of changes that have not been tested appear to have good potential.

Build 1-19, for example, provided relatively good performance including the lowest second gas spring power dissipation to date. A lightweight displacer subsequently fabricated provided a

large boost in power and efficiency, as evidenced by the change from Build 1-26 to Build 1-31, although a part of this change is attributed to a higher compressor pressure ratio. This change can be used in the Build 1-19 configuration to improve performance. A larger displacer rod (G-4) provided a similar improvement as evidenced by the boost in performance from Build 1-42 to Build 1-50.

As previously noted, there is currently a large gap between indicated power and output power. Compressor Unit #2 (Figure 2-1) has been found to be more efficient and provides greater refrigerant flow rate than the first unit, and thus can be expected to improve the output power of configurations tested with the original unit. Second gas spring dissipation is larger than expected and work in this area is expected to reduce this dissipation, and thus improve output power. When all these modifications are properly incorporated, performance improvements over current configurations are expected.

2.2.4 HEAT EXCHANGER OPTIONS FOR PERFORMANCE BOOST

Engine hot side and cold side temperatures have been shown to have significant impact on performance based on test data and TDP simulation. Consequently, the potential impact of redesign of the heat exchangers was studied and found to be attractive. The resulting configuration is being fabricated and will be tested. It is anticipated that the revised design will approach the design goal values of 3 kW and 30 percent efficiency.

A basic analysis of engine performance of one of the better configurations was undertaken using a second order analysis procedure. This calculation quantifies the various losses and is shown in Table 2-2. The heater tube metal temperature of 1260°F and coolant inlet temperature of 80°F resulted in calculated gas temperatures of 1134°F and 389°F. A more effective heater head and corresponding coolant temperatures will result in improved efficiency and power output. The flow friction losses associated with pumping the gas from the hot side to the cold side are excessively high as compared to existing kinematic-driven engines. New regenerators and heater and cooler heat exchangers will reduce this value. Other areas of improvements are also shown in Table 2-2.

Table 2-2. Engine Energy Analysis

	Proto 2 Build 1-19		Improvement Needs
Hot/Cold-side Gas Temp (^{o}F)	1134/389		Higher Carnot efficiency
Displacer/Piston Stroke (in)	1.6/1.1		Larger piston stroke } Higher power
Frequency (Hz)	27.2 Hz		Higher frequency } Higher power
Energy Breakdown	Watts	%	
Basic Power Output	3287	100	
Flow Friction Loss	1075	32.7	Reduce flow pumping work
Indicated Power	2212	67.3	
Basic Heat Input	7285	100	
Regenerator Inefficiency	1151	15.8	Increase regenerator efficiency
Shuttling Heat Losses	262	3.6	
Heat Cond. and Leakage	1938	26.6	Reduce leakage
Flow Friction Credit	(575)	(7.9)	
Gross Heat Input	10061	138	
Indicated Efficiency	22%		

- Need improved Carnot efficiency
- Potential improvement with HX modifications
- Leakage reduction also helps

The key parameters in heat exchanger redesign are the number of heat exchanger tubes and the tube diameter. Parametric studies using the second order analysis (Section 2.3.3) are shown in Figure 2-10. With 12 tubes, the present design is optimal, however, some benefits are shown for 24 sets of heat exchangers. This trend was confirmed with a more detailed TDP simulation and the configuration selected, therefore, was 24 heater tubes of 0.152 inch diameter, 12 regenerators of 0.95 inch diameter/80 percent porosity, and a more effective cooler. The performance attributed to each of the changes is shown in Table 2-3. The modified regenerators provide better efficiency with no power change. The heater and cooler will provide both efficiency and power improvement. A range is shown for power and efficiency, corresponding to different displacer configurations that tend to maximize either power or efficiency (Figure 2-11). Thus, under hardware option C, the configuration selected, the same hardware

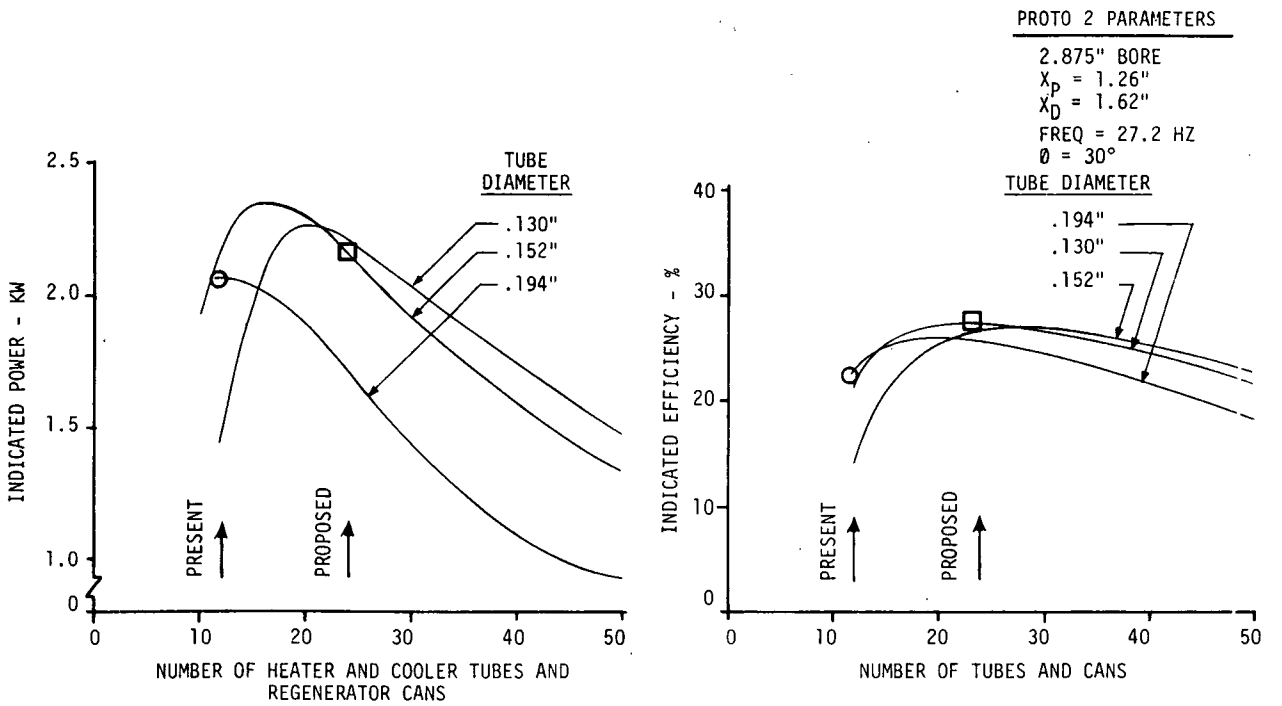


Figure 2-10. Engine Performance Trends with Heat Exchanger Options

Table 2-3. TDP Evaluation of Modified Heat Exchangers

Configuration	Indicated Power (kW)	Indicated Efficiency (%)
• Baseline - Build 1 - 19	2.2	22
A. Modified Regenerator (12 x 0.95" cans)	2.1 - 2.2	25.6 - 25.0
B. A + 24 x 0.152" I.D. Heater Tubes	2.2 - 2.5	28.5 - 27.2
C. B + Modified Cooler - 24 x 0.152" Tubes, or - Tube and Shell	2.4 - 2.7	30.0 - 28.5

- Modified regenerator improves efficiency but not power
- Heater and cooler modifications improve both efficiency and power
- Trade-off between efficiency and power determined by displacer gas spring tuning

can be tuned to provide either 2.4 kW and 30 percent efficiency or 2.7 kW and 28.5 percent efficiency by simple manipulation of the displacer configuration. Engine energy analysis with configuration C is shown in Table 2-4, along with tested Build 1-19 for reference. Carnot efficiency trends with the modified heat exchanger are shown in Figure 2-12.

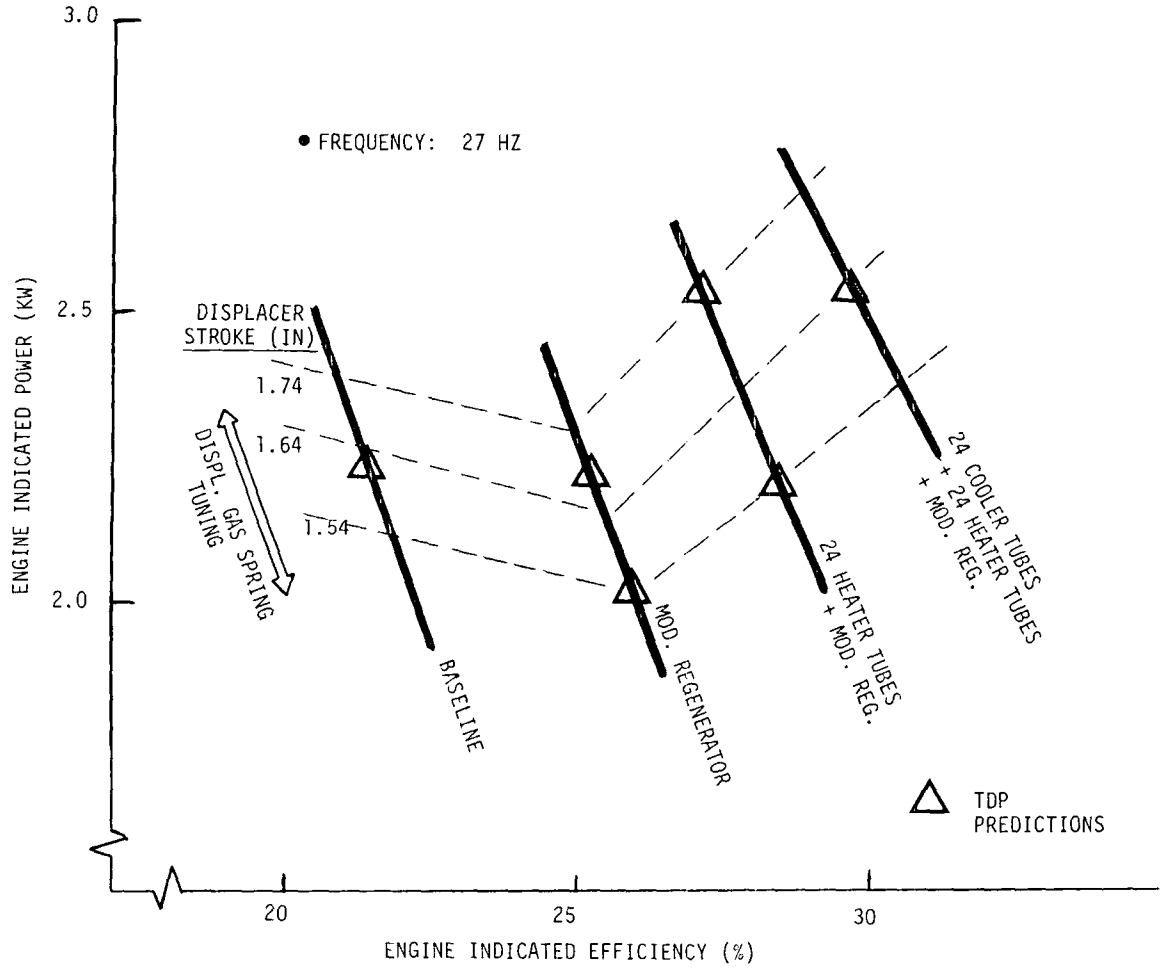


Figure 2-11. Engine Efficiency/Power Tradeoffs

2.3 ANALYTICAL TOOLS

Computer simulation has been used extensively in the design of Proto 2, as an aid in tuning tested configurations, and in predicting the potential of various hardware modifications. The computer simulation program (TDP) has been calibrated with Proto 1 data and has been found to match Proto 2 performance after some improvements. Other simpler computer codes have been developed for parametric studies. Improvement of the simulation code is continuing.

Table 2-4. Engine Energy Analysis with New Heat Exchangers

	Proto 2 Build 1-19		Proto 2 Potential(Configuration C)	
Hot/Cold-side Gas Temp, (°F)	1134/389		1198/320	
Displacer/Piston Stroke (in)	1.6/1.1		1.76/1.24	
Frequency (Hz)	27.2		27.6	
Energy Breakdown	Watts	%	Watts	%
Basic Power Output	3287	100	3490	100
Flow Friction Loss	1075	32.7	670	19.2
Indicated Power	2212	67.3	2820	80.8
Basic Heat Input	7285	100	6950	100
Regenerator Inefficiency	1151	15.8	889	12.8
Shuttling Heat Losses	262	3.6	507	7.3
Heat Cond. and Leakage	1938	26.6	2182	31.4
Flow Friction Credit	(575)	(7.9)	(459)	(6.6)
Gross Heat Input	10061	138	10071	144.9
Indicated Efficiency	22%		28%	

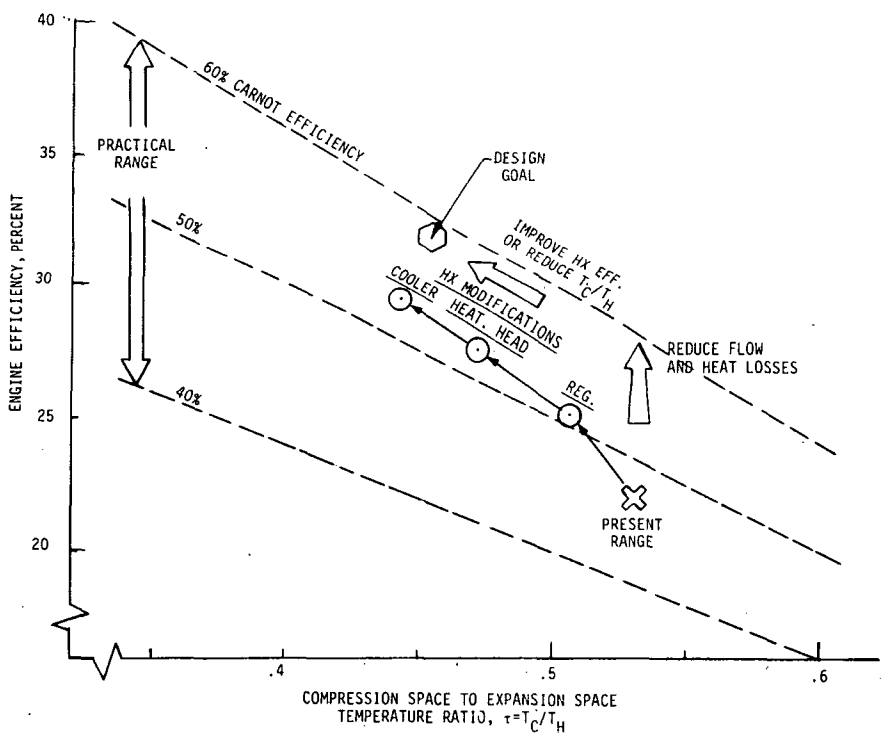


Figure 2-12. TDP Performance Predictions for Modified Proto 2 Engine

2.3.1 TDP SIMULATION

The Thermo-dynamic Program (TDP) is the key tool used 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. Faster temperature convergence of the regenerator elements was implemented and has substantially improved consistency of the results. The program has been streamlined and operating cost has been reduced to allow more extensive use of the program.

A comparison of TDP to test data has been found to be very satisfactory. Figure 2-13 shows a comparison of dynamic parameters. The strokes and frequency correlate well, however, the piston-to-displacer phase angle is consistently over-predicted. This is largely attributed to under prediction of piston motion and thus overdamping and overprediction of displacer stroke and thus underdamping. These trends tend to overpredict phase angle.

A comparison of thermal and power parameters is shown in Figure 2-14. The general trends correlate well. Correlation of the dynamic parameters, except for phase angle, is generally better than for thermal parameters. This is partly attributed to greater uncertainty in measurement of heat fluxes and power than the dynamic parameters.

The data used for TDP comparison covered a large range of test parameters:

Tube Temperature	$T_{Hot\ Side} = 901$ to $1260^{o}F$
	$T_{Cold\ Side} = 265$ to $297^{o}F$
Compressor Pressure	$P_{Suction} = 67$ to 120 psia
	$P_{Discharge} = 142$ to 232 psia

The range of other parameters is shown in Table 2-5, as well as the accuracy of TDP. This indicates that TDP simulation is suitable as a design tool, and can be used to analyze a wide range of parameters as would be needed in analyzing other design variations of this engine.

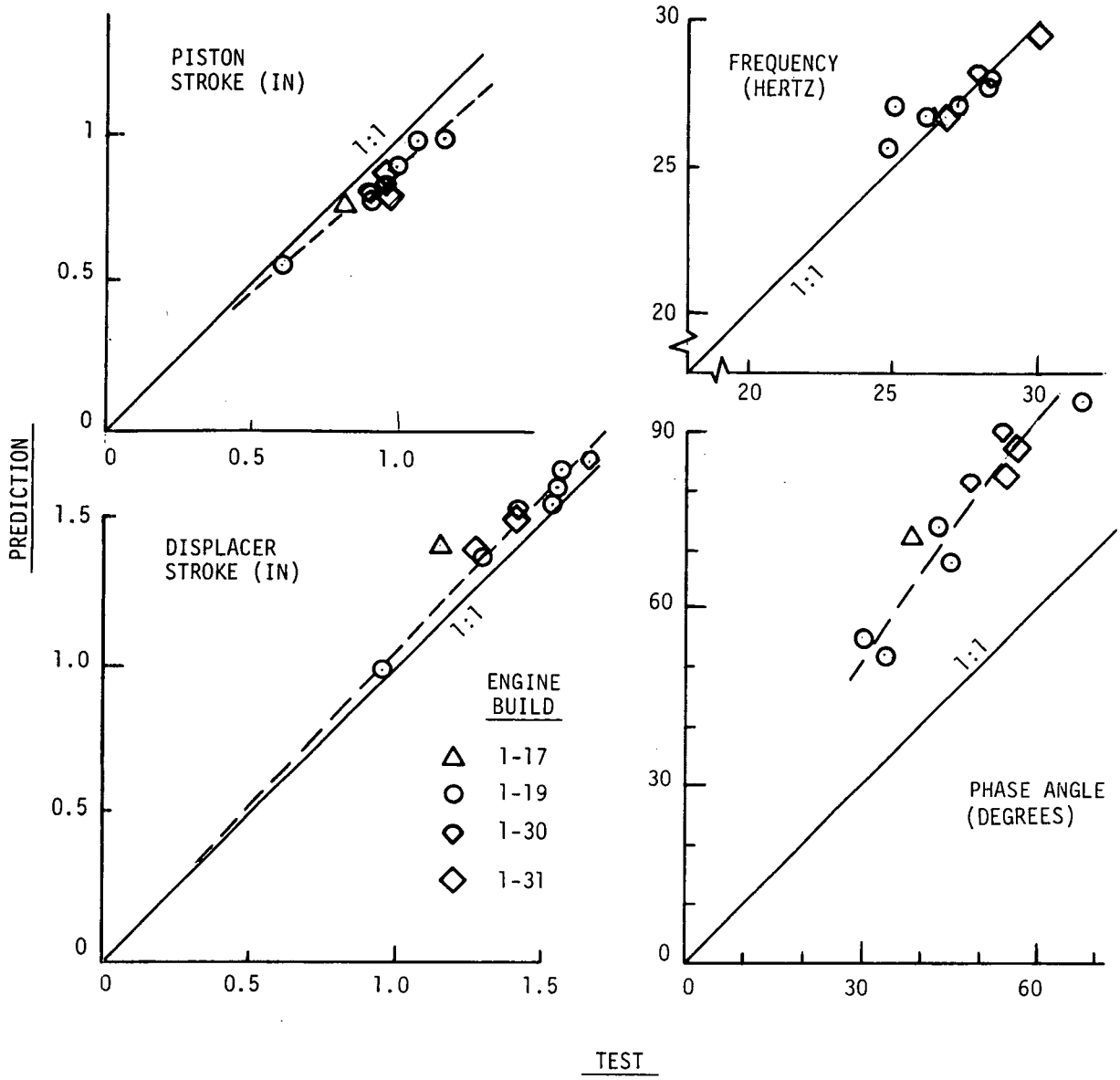


Figure 2-13. TDP Prediction/Test Data Dynamic Correlation

2.3.2 DYNAMICS CODE

A dynamics code has been written as an aid in parametric studies of engine tuning characteristics and as a cost saving prelude to TDP simulation. The code has been found to give trends close to those predicted with TDP. The code uses a Schmidt type analysis to calculate pressures that correspond to calculated strokes which are based on: 1) a three-body spring-mass system with linear gas springs, 2) linear damping and 3) adiabatic displacer and second gas spring compressions. Output of the program consists of the dynamic characteristics: amplitudes, phase angles and operating frequencies.

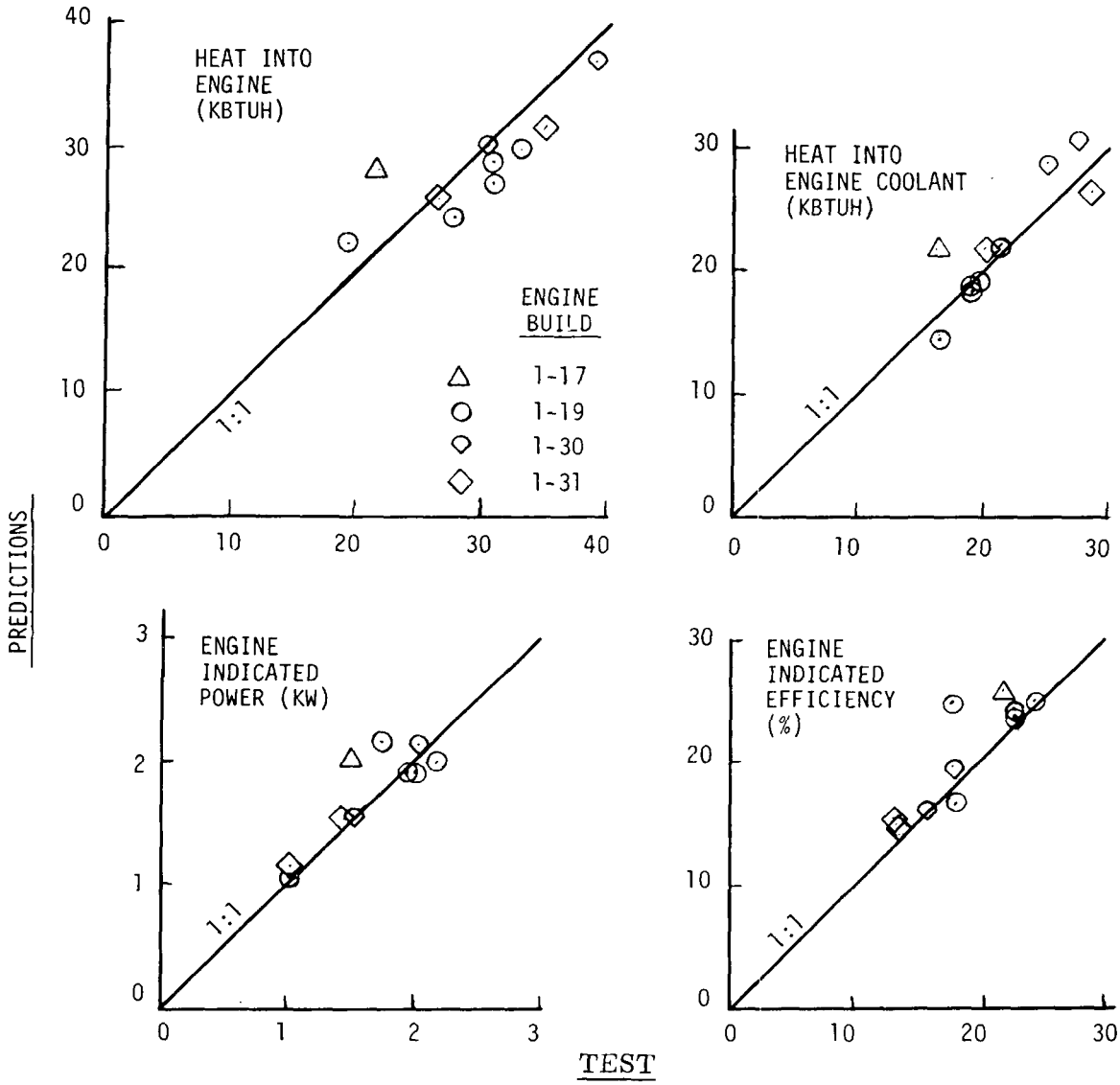


Figure 2-14. TDP Prediction/Test Data Thermal Correlation

Table 2-5. Summary of TDP Predictions Versus Test Data Correlation

Parameter	Range	Accuracy	
		Mean*	Std. Dev.**
Piston Stroke	0.6" - 1.2"	- 10.9 %	4.1 %
Displacer Stroke	1.0" - 1.65"	+ 9.4 %	9.5 %
Frequency	25.1 - 30 Hz	+ 1.6 %	2.9 %
Phase Angle	20° - 67°	+ 33.9 deg.	7.7 deg.
Indicated Power	1.02 - 2.2 kW	+ 9 %	18 %
Indicated Efficiency	14 - 24 %	+ 9.2 %	13.7 %

* Average deviation of TDP relative to test data

** Standard deviations from means

2.3.3 SECOND ORDER ANALYSIS

A simplified thermodynamic analysis to complement TDP and the dynamic analysis has been established using a second order analysis. The calculations are based on the Martini Design method and are considered to be of second order calculation detail, as opposed to the more detailed third order analysis used in TDP. Strokes, phase angle, frequency, and hot side and coolant temperatures are program inputs. The program uses ideal gas laws, isothermal gas spaces, fluid friction losses based on steady flow with empirical correction factors, and empirical heat transfer coefficients. Agreement with TDP and test data is satisfactory, as shown in Figures 2-15 and 2-16. This code is used for preliminary tradeoffs and performance trends.

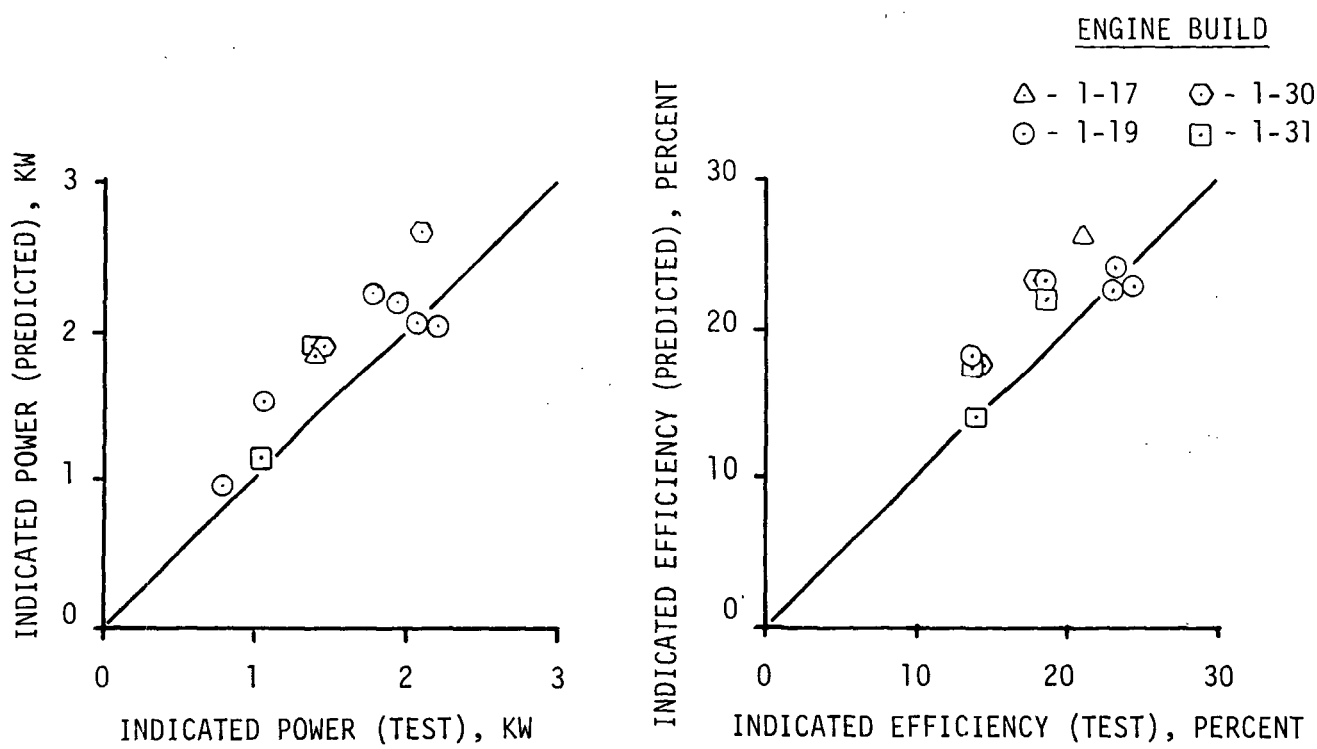


Figure 2-15. Second Order Analysis Versus Test Data Correlation

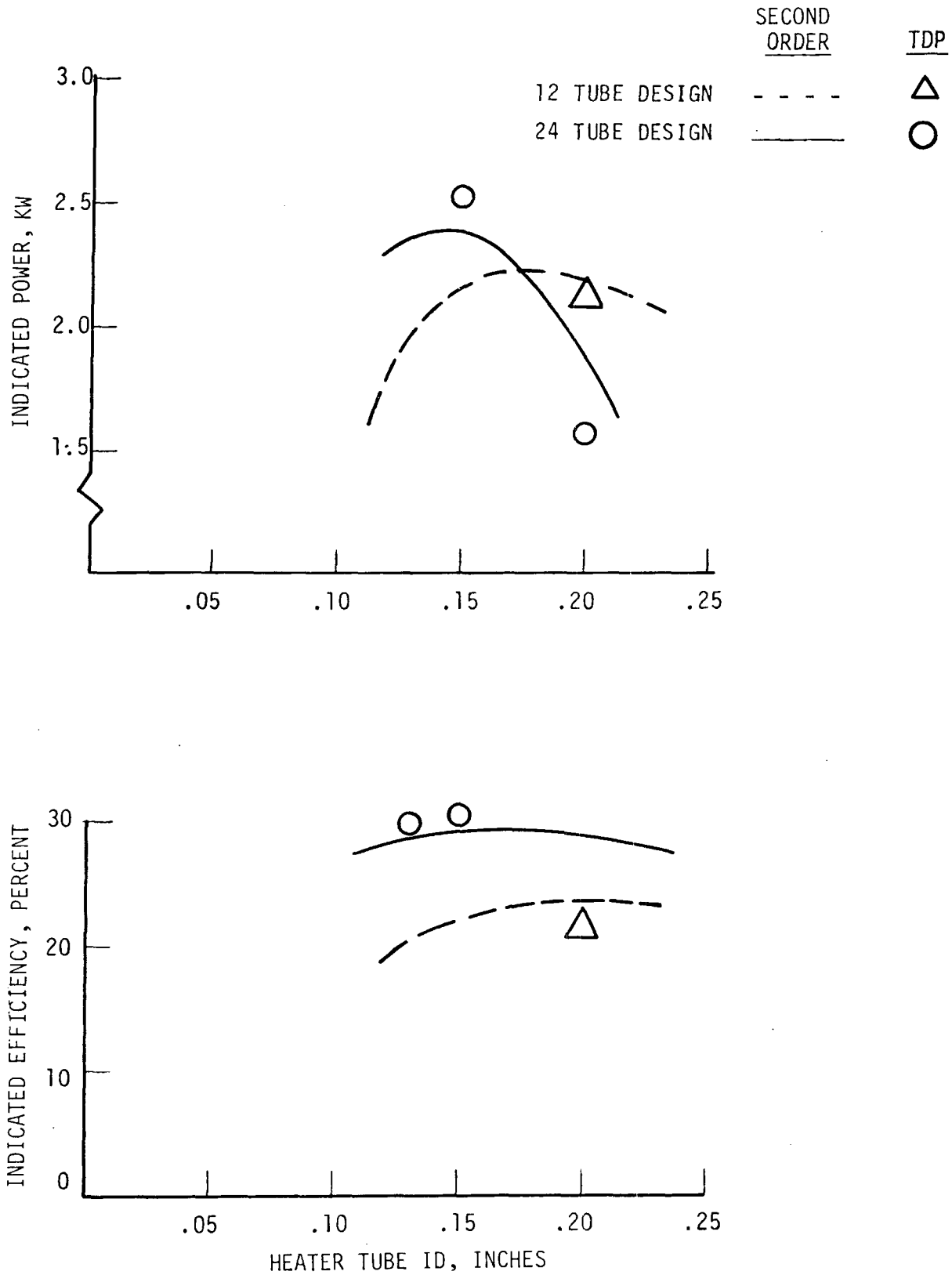


Figure 2-16. Correlation of TDP and Second Order Predictions

2.4 PLANNED ACTIVITY FOR NEXT PERIOD

Activity in the next reporting period will focus on continued data reduction and interpretation, improvement of the baseline engine and TDP simulation capability. Refinements in the data reduction procedures will be pursued, especially in the area of energy flow analysis, in order to verify engine output and to quantify losses. A number of engine tuning options need to be investigated in order to continue performance improvements, especially in the area of second gas spring power dissipation. As for TDP simulation, even though current agreement with data is satisfactory, a number of areas have been identified for more detailed modeling.

SECTION 3

PRODUCT DEVELOPMENT

SECTION 3
PRODUCT DEVELOPMENT

3.1 INTRODUCTION

The principal HAHP development effort during the current reporting period was directed at the integrated testing of the combustor/engine/compressor assembly. Engine start-up was achieved on the initial try with the first assembly of the first build configuration. The subsequent performance-trend test runs entailed interim configuration changes and substitution of modified components to upgrade performance and/or provide engineering data for analytical purposes. The compressor and combustor units both supported this test effort well. However, several combustor modifications were made to eliminate flame-out and flashback problems. In this test series, the compressor was not required to perform across its full operating range; therefore, its total performance remains to be assessed more fully. Component-level tests for the compressor were accomplished separately on its improved test stand, and were aimed particularly at improvement in piston centering under operational conditions. Combustor burner element life also received continuing development effort during this reporting period. Air handler and controls subsystem fabrication and testing work progressed satisfactorily in preparation for subsequent system testing phases. It was determined that the power output and efficiency of the engine were below design goals and would require further hardware modifications. Attention to defining and implementing this portion of the effort was given a high priority, with less critical tasks being temporarily deferred.

3.2 COMBUSTOR

3.2.1 SUMMARY

Results of the Proto 2 combustor performance tests at the component level were presented in the previous report. During this reporting period, combustor performance verification continued as part of the integrated engine/compressor test. Detailed descriptions of the present combustor design is contained in previous semi-annual reports; hence only the experience gained during the integrated tests is reported here.

3.2.2 COMBUSTOR TESTING

The Proto 2 combustor unit was assembled together with the engine on the integrated test stand as shown in Figure 3-1. The combustor unit is fully instrumented to provide measurement of the following items:

- Natural gas supply pressure
- Natural gas regulator discharge pressure
- Natural gas flow rate
- Combustor inlet air pressure
- Combustor outlet exhaust gas pressure
- Air flow meter inlet pressure
- Air flow meter ΔP

In addition, thermocouples are attached at strategic locations on both the inner and outer surfaces of the combustor assembly. The aforementioned instrumentation, in combination with exhaust gas analyses, is sufficient to allow evaluation of the combustor performance.

Exhaust Gas Analysis

In order to perform the exhaust gas analysis, a sampling tube is connected from the exhaust duct of the combustor to the gas analyzers. The water vapor in the exhaust gas is condensed before reaching the analyzer via an air side heat exchanger. Infrared type analyzers (Sun Model EPA-75, Lira Model 202 and 303) are used to monitor levels of CO, CO_2 and unburned hydrocarbons. A schematic drawing of the set up is shown in Figure 3-2. The Sun exhaust analyzer is used to determine the hydrocarbon concentration and CO concentration in excess of 0.1 percent (1000 ppm). The Lira 202 analyzer is used when CO emissions are less than 0.1 percent (1000 ppm). A Lira 303 analyzer is used to measure CO_2 emissions.

Control of CO emission to below 0.04 percent is one of the combustor design requirements. Knowledge of the CO and CO_2 concentration in the exhaust gas is used to determine the excess air level and the exhaust gas heat loss to ambient. Hydrocarbon level must be kept very low

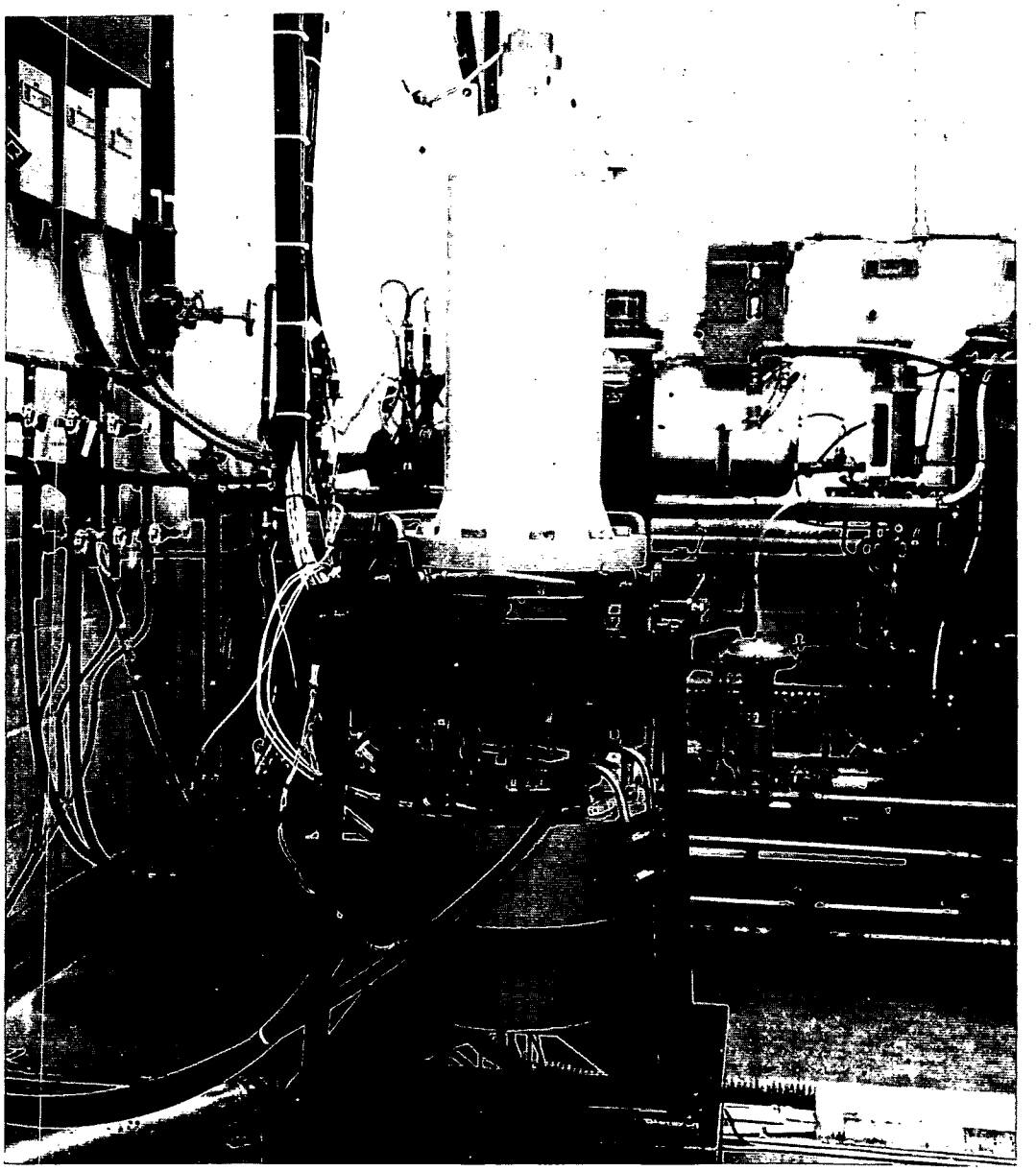


Figure 3-1. Proto 2 Combustor/Engine/Compressor Integrated Test

to assure complete combustion. The presence of NO_x is not significant because of relatively low operational temperature and small residence time of the air/fuel mixture.

Based on high heating value of the fuel, the heat lost in the exhaust gas is determined from

$$Q_{exhaust} = \left[\left(\frac{0.0193}{CO_2} + 0.0518\right) (T_e - 75) + 100 \cdot 245)\right] \dot{m}_{fuel} \text{ (Btu/hr)}$$

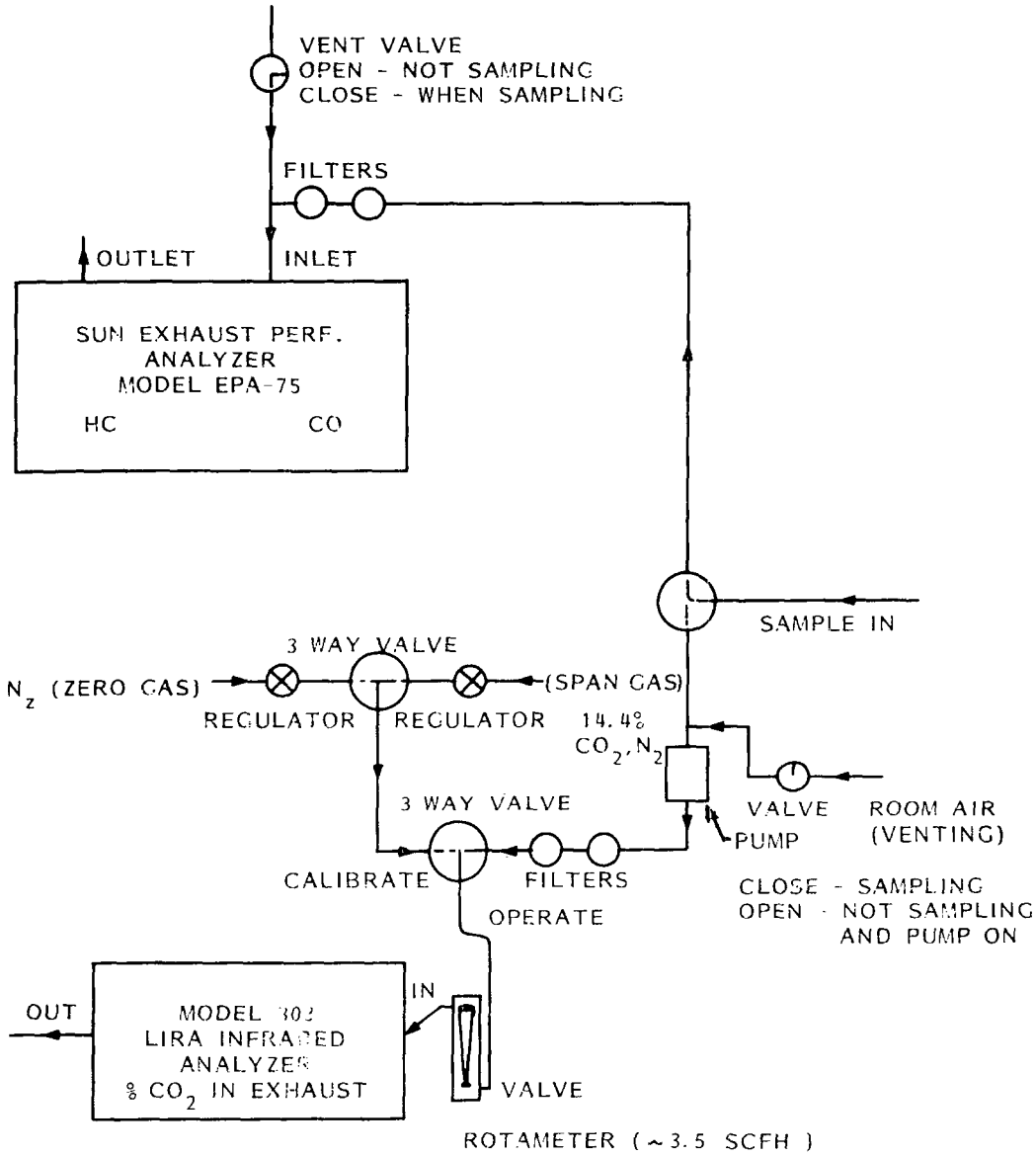


Figure 3-2. Emissions Equipment Schematic - CO_2, HC and CO

The excess air level is determined from the Siegert's formula.

$$\% \text{ Excess Air} = \left(\frac{0.1188}{[CO_2] + [CO]} - 1\right) \times 100$$

Performance Test Results

Throughout integrated testing, the combustor performed close to its design specifications. As discussed later, some flashback and flameouts were experienced during the early part of the

testing. In order to accomplish the primary integrated test objective of gaining as much engine running time as possible at various operating conditions, temporary fixes to the combustor were made to eliminate the flashback problems. It became essential to remove the insulation from around the preheater section to lower the air preheat temperature. This resulted in less than projected combustor efficiency; however, the preheater effectiveness remained as projected, namely about 66 percent. A typical temperature map of the combustor during integrated test is shown in Figure 3-3. The design insulation condition energy balance values for the combustor are also depicted in the figure. As expected, a resultant major heat loss occurs in the exhaust gases.

The overall pressure drop in the combustor is shown in Figure 3-4 as a function of firing rate. Throughout the operating range the pressure drop is well below the design goal of 2 inches of H_2O. The pressure drop is also relatively insensitive to the amount of excess air indicating that much of it occurs during the mixing of air and fuel.

Results of the exhaust gas analysis are shown in Figures 3-5 and 3-6. The CO_2 level in the exhaust gas is slightly less than expected for complete combustion. However, the trend shown on the graph follows the theoretical prediction very closely and is independent of the firing rate. The CO level in the exhaust gas is generally higher than the design goal and the Proto 1 test data. This results from the fact that the operating temperature conditions of the Proto 2 combustor burner and the air preheat are generally lower than the normally expected values. This was done deliberately in an effort to prevent pre-ignition of the air fuel mixture within the burner environment. Such a compromise was necessitated when the earlier engine configurations were unable to remove sufficient levels of energy from the combustion process and the resulting abnormally high heat sink and exhaust gas temperatures lead to flashback or pre-ignition problems. The characteristic knee of the CO versus excess air curve still seems to occur at about 0.01 percent CO and a nominal level of 30 percent excess air. The design point exhaust gas limit for CO is 0.04 percent at 15 percent excess air. For excess air levels of less than 30 percent, the resulting incomplete combustion yielded higher than desired levels of CO. However, as the excess air was increased, the combustor efficiency was maximized and the CO level fell within the acceptable limit. This is true for any given level of exhaust gas temperature. Previous data has shown that the exhaust gas temperature is relatively insensitive to the excess air level. Therefore, the overall effect in combustor thermal efficiency of going from the design value of 15 percent excess air to a nominal 30 percent excess air level will be minor.

BUILD 1-19, 10:54 DATA

AVERAGE TEMP. °F

INLET AIR	90
EXHAUST GAS	795
PREHEAT AIR	1120
HEATERHEAD EXHAUST	1410
HEATERHEAD TUBE	1260
FINS	1415

ENERGY BALANCE (HHV)

FIRING RATE	–	44.1 KBTUH
EXHAUST LOSS	–	12.5 KBTUH
SURFACE LOSS	–	1.8 KBTUH
HEAT TO ENGINE	–	29.8 KBTUH
COMBUSTOR EFFICIENCY	–	68%

EMISSIONS

EQUIVALENCE RATIO	=	.83
CO2	=	9.0%
HC	=	35PPM
CO	=	.3%

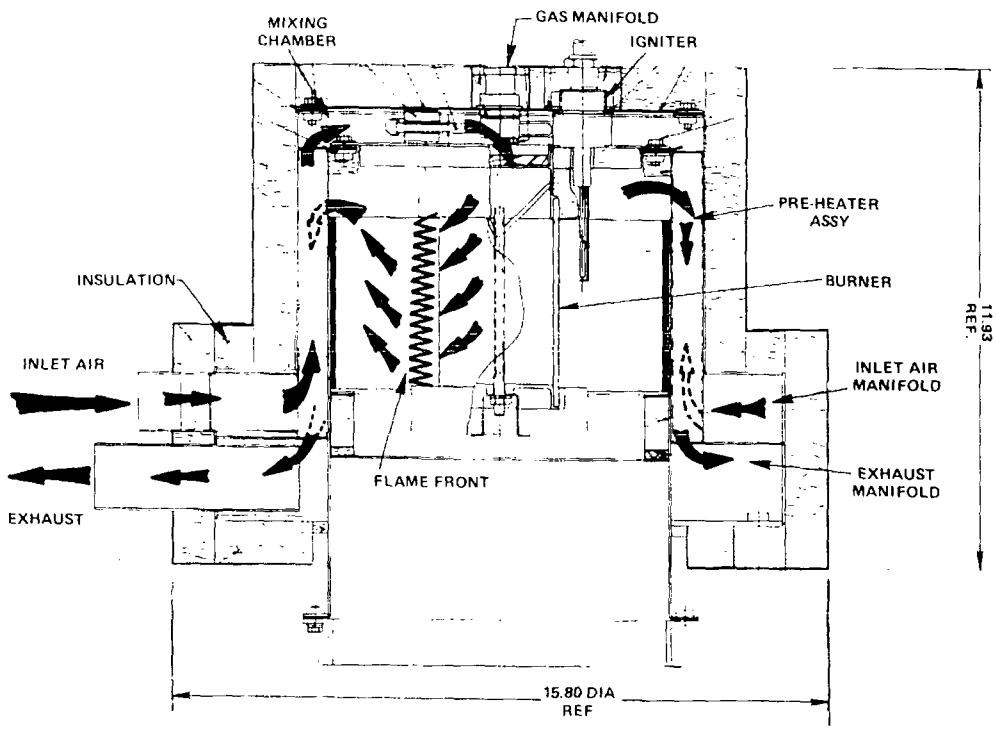


Figure 3-3. Proto 2 Combustor Assembly Temperature Map and Energy Balance

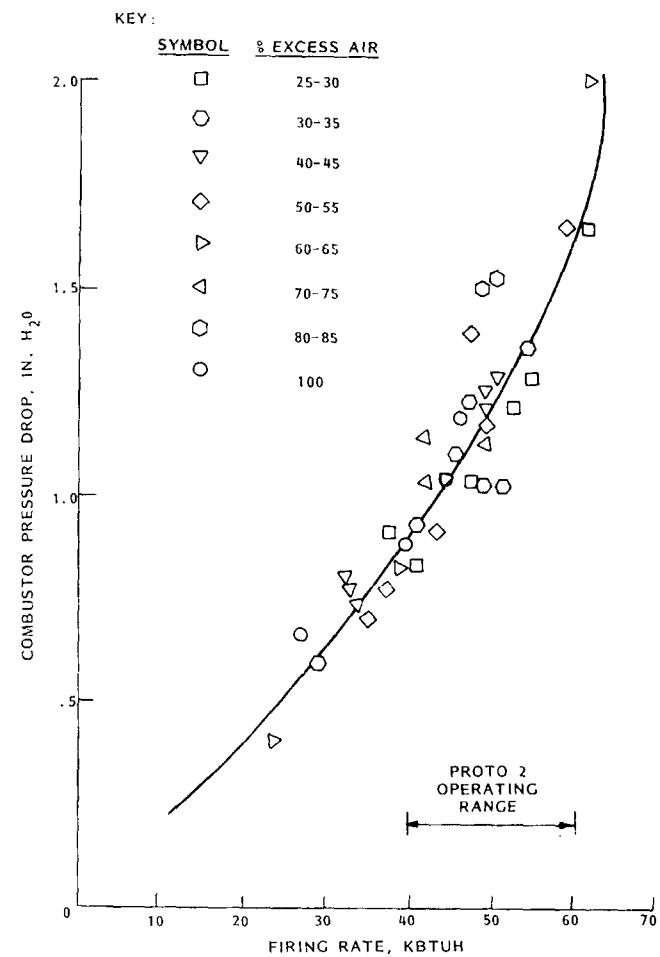


Figure 3-4. Pressure Drop in the Combustor as a Function of Firing Rate

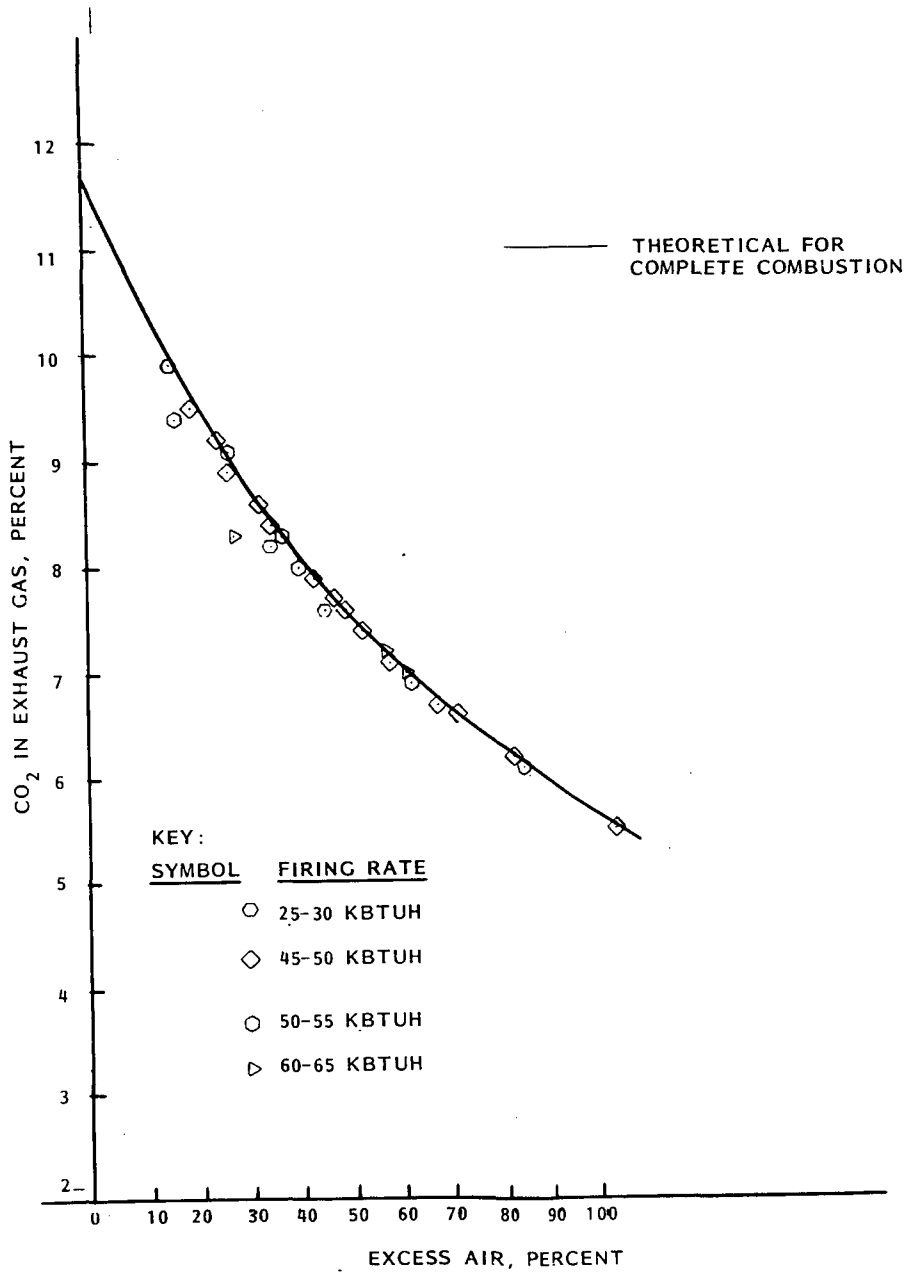


Figure 3-5. CO_2 in Exhaust Gas as a Function of Excess Air

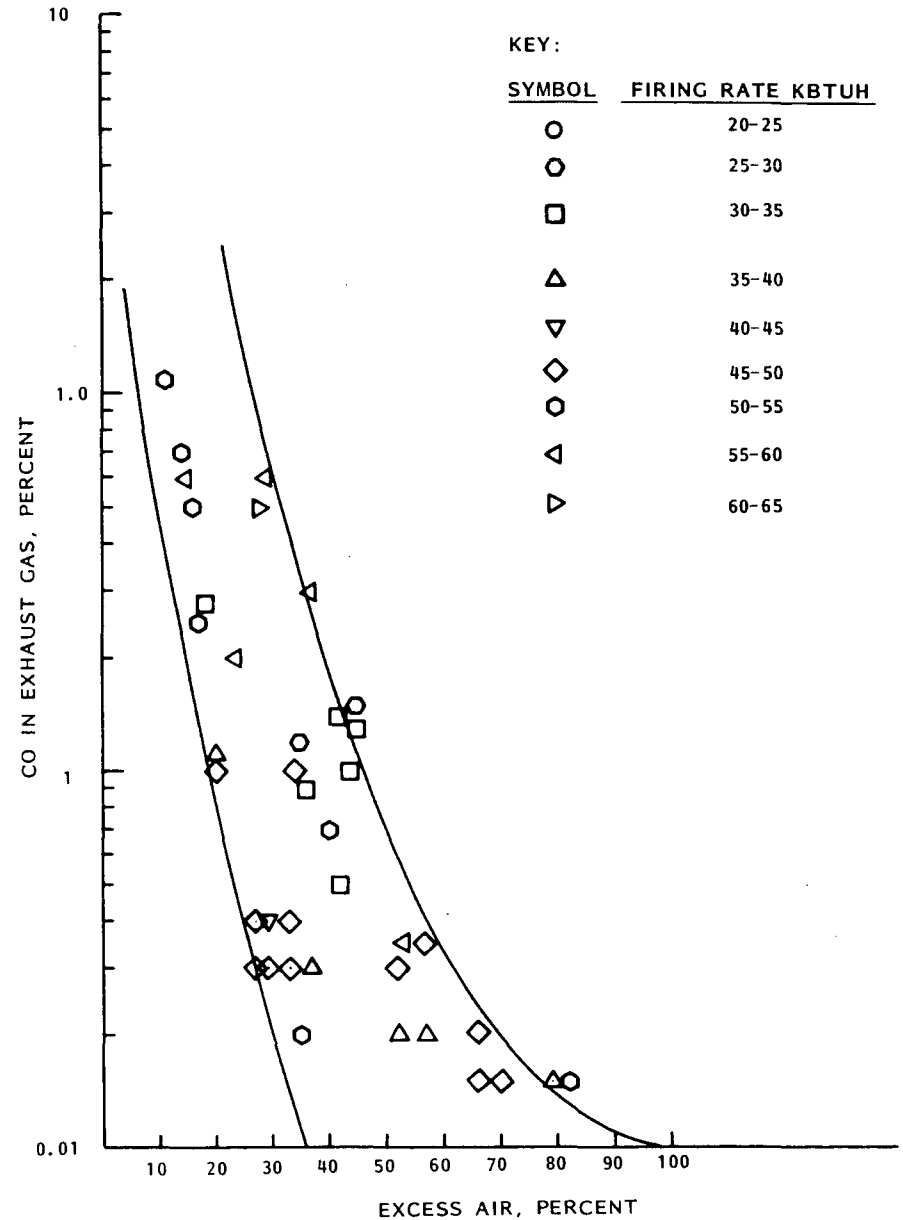


Figure 3-6. CO in Exhaust Gas as a Function of Percent Excess Air

Flashback Experience

During the integrated test, a combination of flashback and pre-ignition of the air fuel mixture within the burner was experienced. When this occurs, the flame detector circuit senses the absence of the flame, closes the gas valve and secures the combustor operation per the control's design safety procedure. The pre-ignition was found to be a combined result of higher than predicted pre-heater air temperature and a region of flow stagnation within the burner element in the vicinity of the cap. Lowering the pre-heater air temperature to less than 900°F by removing the insulation from around the pre-heater section and installing a concially shaped flow distributor has temporarily fixed the pre-ignition problem. The flow distributor serves two purposes. It reduces the residence time of the air/fuel mixture compared to ignition delay time, and thus prevents spontaneous ignition. Secondly, it acts as a safety feature by increasing the thermal inertia of the burner, thus preventing local melting of burner element fibers in case pre-ignition or flashback does occur.

The flashback phenomena occurs when the velocity gradient at the wall diminishes compared to the ratio of flame speed to the quenching distance. In the case of flashback, the flame may propagate upstream into the mixture chamber and attempt to stabilize there. However, the flame detector in the Proto 2 combustor assembly will prevent flame stabilization inside the burner since as soon as flame retracts from the burner surface, the fuel valve is shut off. When a Nextel ceramic sleeve was placed over the outer surface of the burner, to enhance the burner element life, the problem of flame out seemed to have worsened. It is theorized that despite the high temperature capability of Nextel material, due to inherent non-uniformity and increased pore size, the flame begins to retract and then burns inside the plane of the sleeve. This rapidly results in progressive deterioration due to reduced heat transfer, sintering of the sleeve, burner wall failure and ultimately in flame out.

Burner Element Life

The present burner element in the combustor is made of chopped, pressed and sintered Hoskin 875, 5.6 mil diameter wire with 80 percent porosity. The outside temperature of the burner element during operation is estimated to be in the range of 2200 - 2500°F. This leads to rapid deterioration of the element due to oxidation. The maximum operational time obtained on these burners thus far is approximately 60 hours. The elements before and after 60 hours of operation are shown in Figure 3-7. It was observed that the wires, after being oxidized, start to flake-off and the burner wall begins to be depleted. The maximum operational life of the

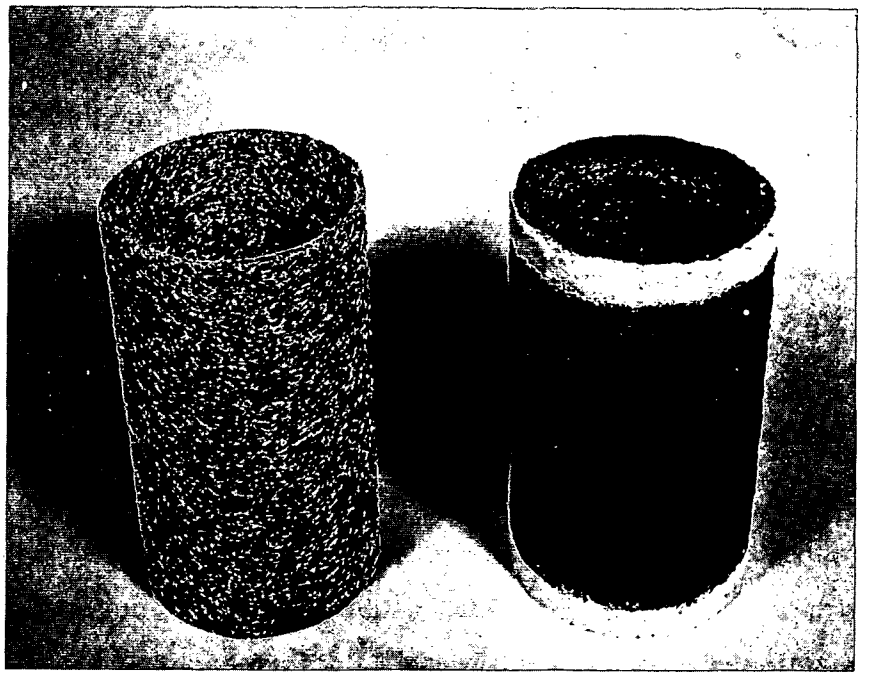


Figure 3-7. Burner Elements: New (left) and After 60 Hours of Operation (right)

existing burner is estimated to be less than 100 hours. However, it is desired that the burner last at least 1 year before replacement.

It was reported earlier that a substantial enhancement of the burner life could be achieved if the operating temperature of the element is reduced by 300 - 400°F. With this in view, as also mentioned in earlier reports, successful tests were conducted with a Nextel material ceramic sleeve surrounding the burner element. These tests were conducted at the component test facility and indicated a reduction in the operating temperature of the metallic burner surface of the order of 200°F. However, when the burner element was fired in combination with the Nextel sleeve on the integrated test stand, several incidences of flashback occurred. Visual observation indicated embrittlement of Nextel fibers and subsequent flaking of the fibers. Since the manufacturer claims the viability of the material up to 2600°F, it is apparent that in the burner assembly environment, the temperature of the sleeve exceeded it's capability and failed.

This led to re-evaluation of the concept of using a ceramic sleeve around the burner element and investigation of alternate possibilities for enhancing the burner life. Work on these are in progress and the results will be presented during the next reporting period.

Insulation

The Proto 2 combustor has surrounding the preheater section a 1 inch thick section of rigidized Kaowool insulation. During integrated testing, this thickness of insulation around the preheater was sufficient to raise the preheater air temperature above the auto-ignition temperature of the air/fuel mixture and thus resulted in the pre-ignition within the burner. To facilitate the continuation of integrated testing of the engine and compressor it was decided to remove the insulation from around the preheater section and, therefore, lower the preheater air temperature, thus eliminating the pre-ignition condition. Removal of the insulation results in reduced combustor efficiency. However, it is expected that by improving the engine heat absorption capacity and making minor design modifications to the combustor, the tendency for pre-ignition can be alleviated with only small reductions in insulation, thereby minimizing system efficiency effects. The optimization of combustor performance is forthcoming and will be reported in the next semi-annual report.

3.2.3 PLANNED ACTIVITY FOR NEXT PERIOD

During the next reporting period the following combustor related activity is planned:

1. Combustor performance optimization and minor design changes to prevent flashback
2. Investigations to substantially enhance the burner life, including re-evaluation of Nextel ceramic sleeve concept.
3. Re-evaluation of combustor insulation requirements and efficiency and at the same time eliminate the occurrence of pre-ignition due to high preheat temperature.

3.3 ENGINE

3.3.1 SUMMARY

The Proto 2 engine development effort during this reporting period saw the early completion of the fabrication of the few engine components reported incomplete in the last report. The primary effort, subsequently, consisted of incorporating configuration changes in support of

performance - trend testing of the engine/compressor assembly on the Integrated Test Stand. Some effort was also expended in the generation of engine housing design layout drawings, in concert with analytical studies, to identify heater head tube and fin, regenerator, and cooler configurations offering improved engine power output and efficiency. This study, initiated in the latter portion of this reporting period, will be completed early in July and the resulting hardware changes will be started immediately thereafter.

The following paragraphs present a more detailed discussion of the engine development effort.

3.3.2 ENGINE ASSEMBLY

Status

The first Proto 2 Free Piston Stirling Engine was assembled and installed into the Integrated Test Stand in January. It is noteworthy that start-up was completed on the first assembly of the first build configuration of the engine, and that a 30 minute run was completed for both engine and test facility check-out purposes. Many subsequent build configurations/assemblies were completed in support of the performance-trend test program conducted on the Integrated Test Stand. Component changes completed consisted of the following:

- Engine housing replacement to change the regenerator matrix porosity from 70 to 80 percent.
- Displacer rod and bearing sleeve set replacement to provide rod areas of 1.35 and 1.48 in^2.
- Displacer gas spring spacers added or deleted to provide mean gas spring volumes of approximately 6.5, 10, 12 and 14.5 in^3.
- Several displacer centering port locations were incorporated to center the displacer with respect to the power piston.
- Displacer replacement to provide displacer assembly weights of 2.25 and 1.76 pounds.
- Second gas spring piston and mating liner replacement to provide piston areas of 11.6 and 14 in^2.

- Multiple second gas spring vent port areas and patterns to provide engine start capability with minimal pumping losses under operating conditions.

- Second gas spring spacers to provide mean gas spring volumes of 81, 100, and 122.5 in^3.

See Figures 3-8 and 3-9 for cross-sectional views identifying the components noted in the aforementioned configuration changes.

Heat Exchanger Flow Versus Pressure Drop

Flow versus pressure drop tests were completed on the Proto 2 engine housing heater head assemblies prior to their initial installation into an engine. These tests were completed to establish the pressure drop characteristics of the assemblies in a "new unit" condition for future reference, as well as to permit comparisons between units having different regulator matrix porosities. A schematic diagram of the test set-up is shown in Figure 3-10. The results of the tests on the first assembly, having a 70 percent regenerator matrix porosity, and the second assembly, having a matrix porosity of 80 percent are shown in Figure 3-11. It is noted that the pressure drop characteristics obtained are within predicted values, with the unit having the 80 percent porosity matrices exhibiting a pressure drop of less than 50 percent of that of the 70 percent porosity unit.

3.3.3 HEATER HEAD DESIGN MODIFICATIONS

Design studies were initiated during this reporting period to identify and evaluate potential changes to the engine housing assembly to improve engine power output and efficiency. Design layouts have been started in support of this effort. Modifications being considered include increasing the number and/or varying the size of both the heater head and cooler tubes, plus increasing the diameter of the regenerators. Changes are being limited to those which are compatible with the existing and inter-relating Proto 2 hardware. Analytical studies conducted concurrently with the layout work will be instrumental in selecting the most desirable configuration. This selection will be made early in the next reporting period. Reference is made to Figure 3-12 for a cross-sectioned view of the engine housing assembly, showing the areas of interest in the re-design effort.

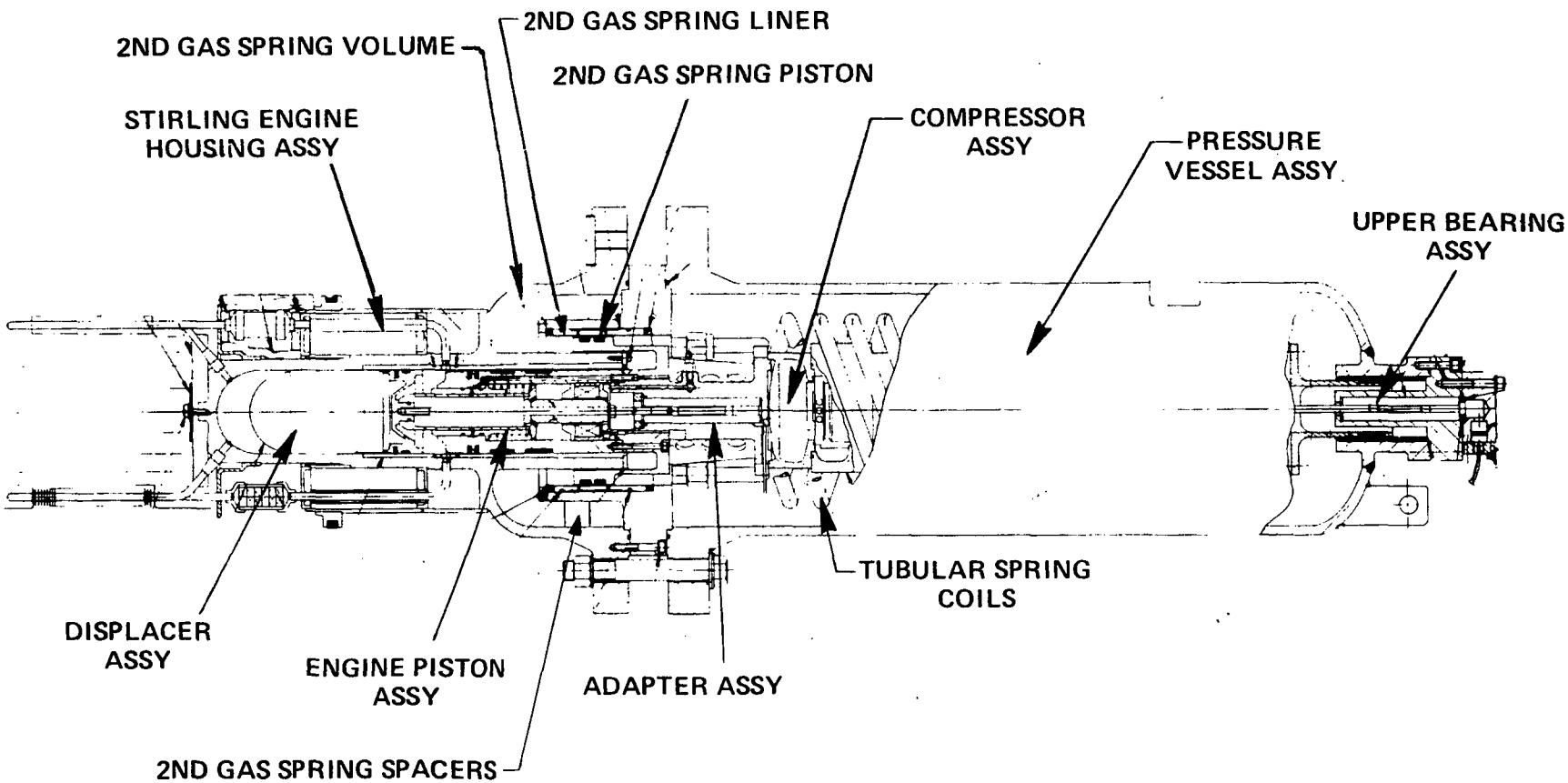


Figure 3-8. Proto 2 Engine and Compressor Assembly

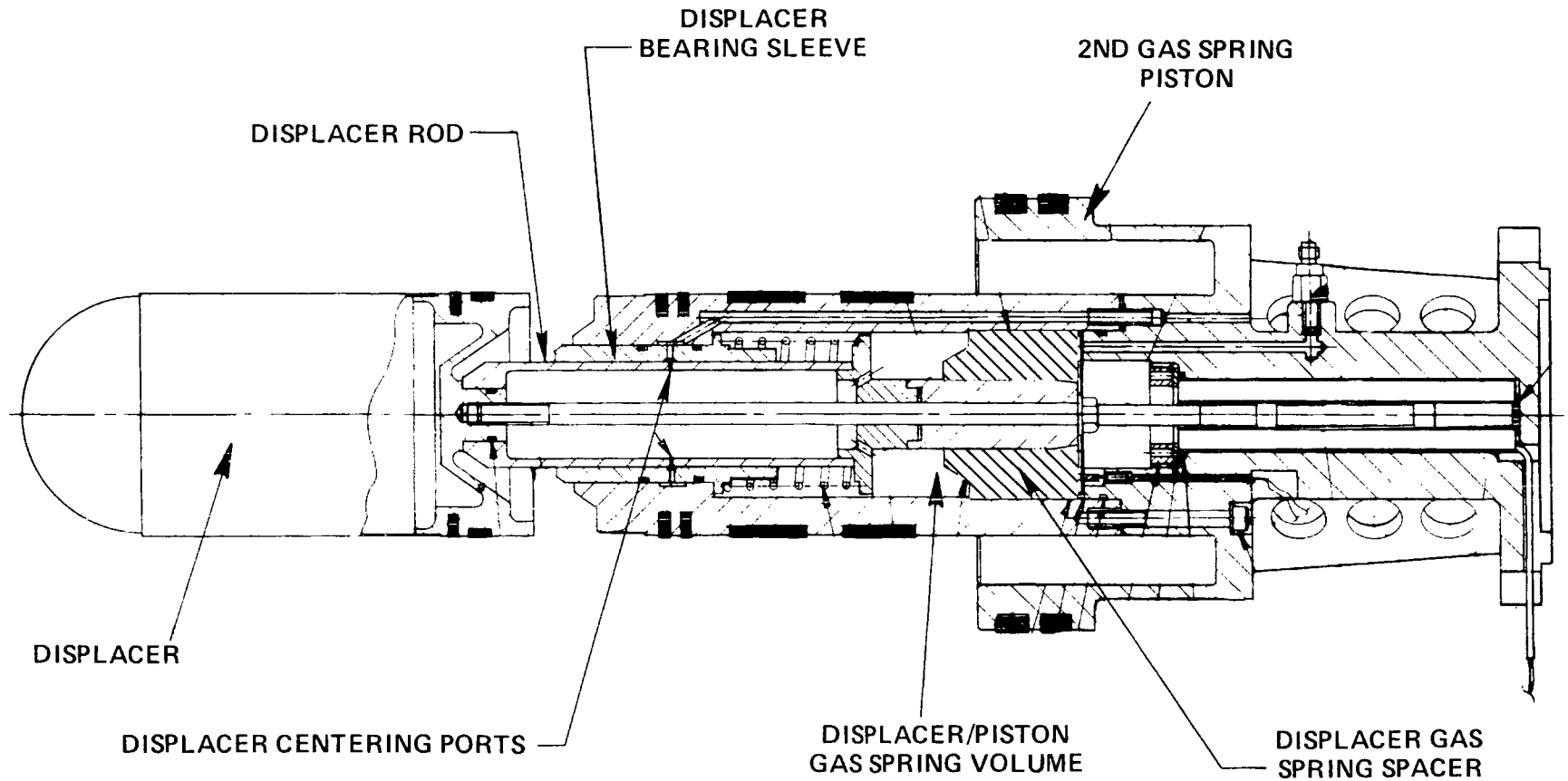


Figure 3-9. Proto 2 Displacer, Power Piston, and 2nd Gas Spring Piston Assembly

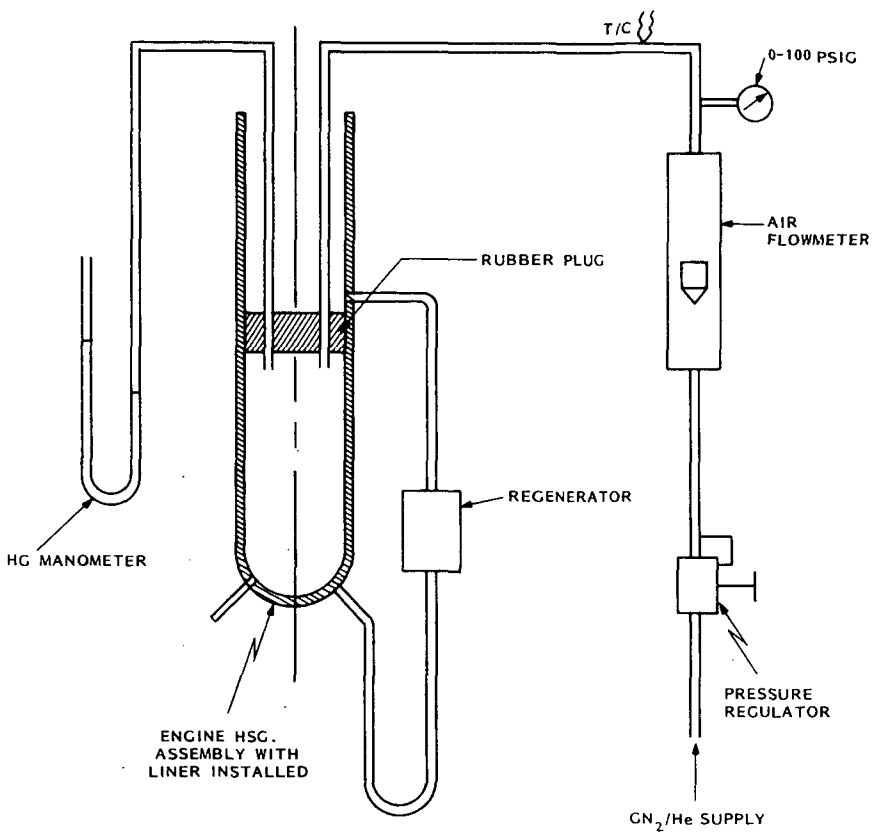


Figure 3-10. Schematic Diagram of Heater Head Pressure Drop Test

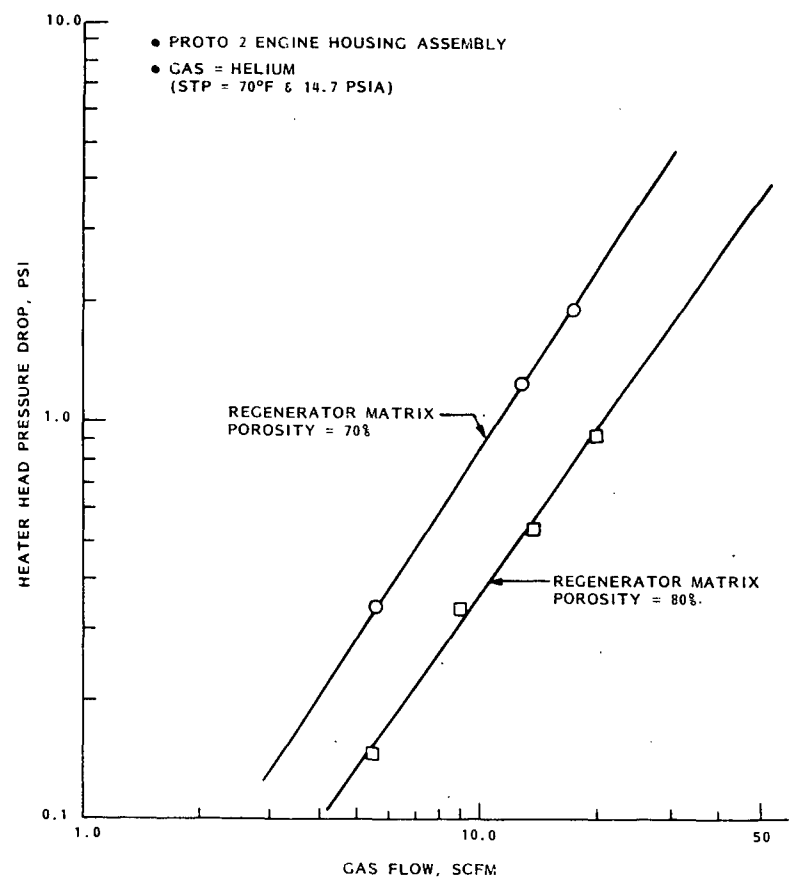


Figure 3-11. FPSE Heat Exchanger Pressure Drop

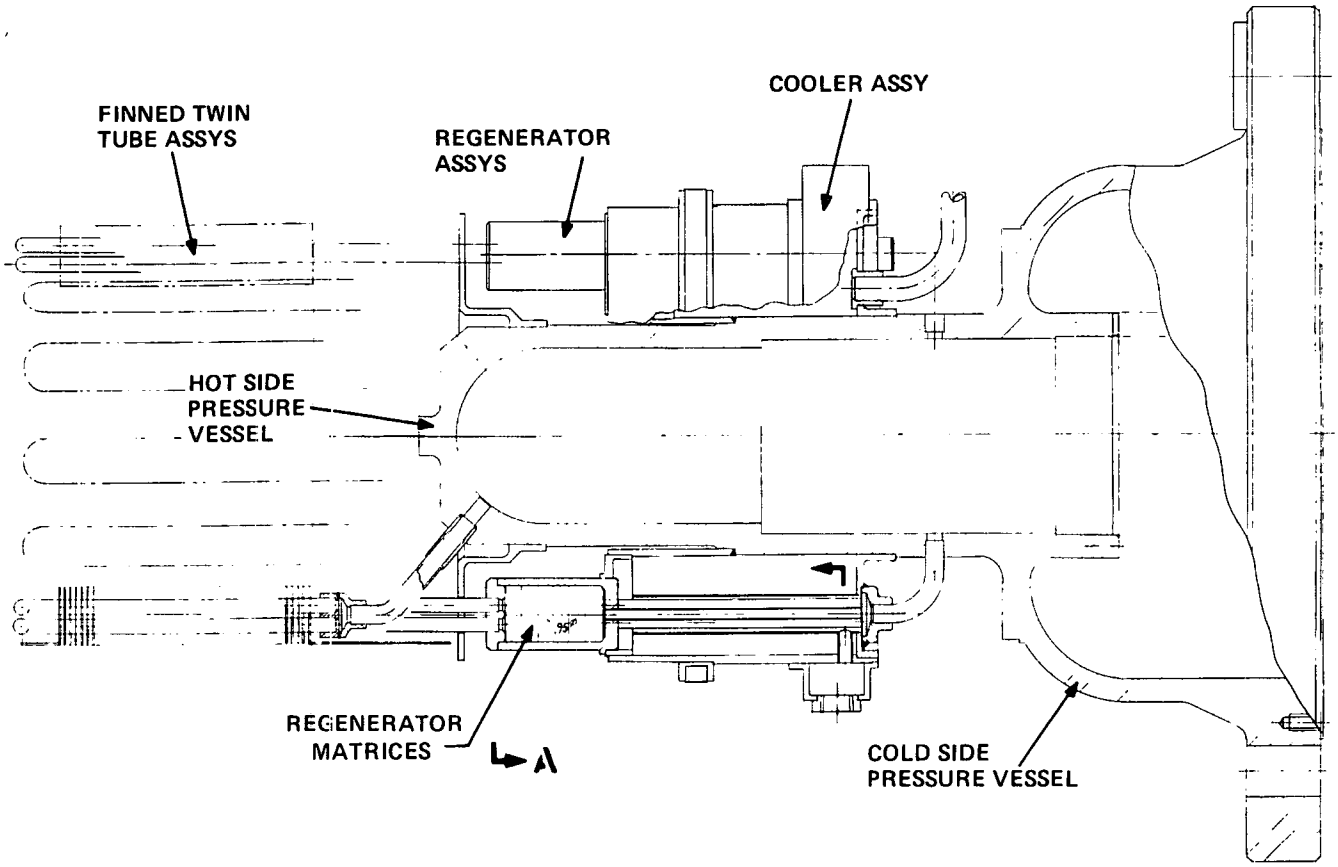


Figure 3-12. Engine Housing Assembly, Modified Configuration

3.3.4 PLANNED ENGINE ACTIVITY FOR NEXT PERIOD

Activities planned for the Proto 2 engine during the next period include selection of the modified engine housing configuration, detailing and issuing the engine housing rework drawings, fabrication of the detail parts, assembly of (brazing/welding) the modified engine housing, and installation of the assembly into the Integrated Test Stand.

3.3.5 ALTERNATE STARTER

Status

Several Proto 2 engine starting techniques were tested during this reporting period as potential alternates to the pulse-vent method developed during the Proto 1 engine test program. These alternates included an electromagnetic starter built into the engine power piston/displacer assembly, an engine housing impulse method, and an external pressure-pulse generator technique. The following paragraphs present a discussion of the results obtained with each starter mechanism/approach.

Electromagnetic Starter

A cross-sectioned drawing showing the electromagnetic starter components is shown in Figure 3-13. Tests conducted on this starter as part of the Integrated Test effort produced similar results to those observed during the initial checkout tests; insufficient motion due to excessive damping of the displacer motion because of seal friction. A modified seal ring, having approximately 30 percent of the frictional drag force of the original design, was also tested. While this configuration showed some improvement in the displacer motion, the resulting starter excitation was inadequate to generate an engine start.

A new magnetic starter core/plunger and related double-pulsing electronic circuit was subsequently designed to permit electrically pulsing the displacer in both the upward and downward direction. This revision was also implemented and tested, but the design still failed to produce the proper displacer excitation to start the engine.

Engine Housing Impulse Starter

Engine start-up was successfully accomplished by manually providing a sudden impulse on the engine housing. Even though the measured displacer and piston strokes were small (approximately 0.06 inch), a relatively large phase angle was created between the two moving masses by the external impulse, and the engine started. Due to the lack of a well-controlled mechanical impulse generator, no further tests were performed with this technique.

Pneumatic Pulse Starter

An external pressure-pulse starting technique was tested and the results are very encouraging. The test mechanism, shown schematically in Figure 3-14, is a refurbished pressure pulse generator, originally designed and built for producing high pressure pulsed-flow for a heat exchanger test. The plunger of a specially designed pneumatic cylinder is driven by a variable speed dc motor via a variable stroke crank mechanism. The pulsed-flow is introduced into the engine through a port communicating directly with the engine cold space. A manual throttle valve is provided to permit isolating the starter from the engine when desired. The device is capable of generating pressure-pulse amplitudes of $\pm$ 25 psi up to a frequency of 34 Hz. During check-out tests prior to firing the engine, the engine (with a soft second gas spring) was "motored" with the pulse generator. Figure 3-15 shows the displacer/piston phase angle

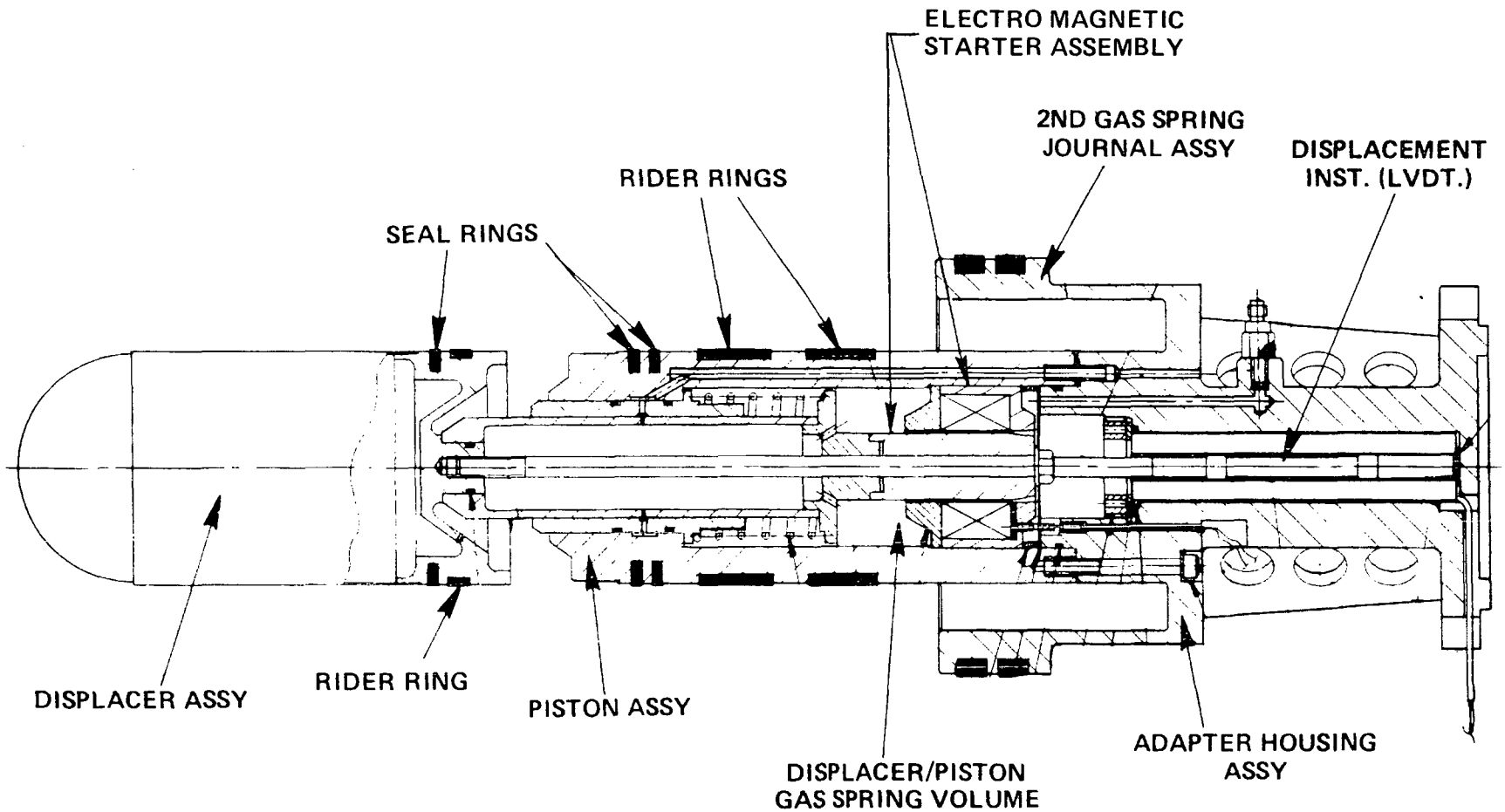


Figure 3-13. Proto 2 Electromagnetic Starter, Displacer, Piston and Adapter Assembly

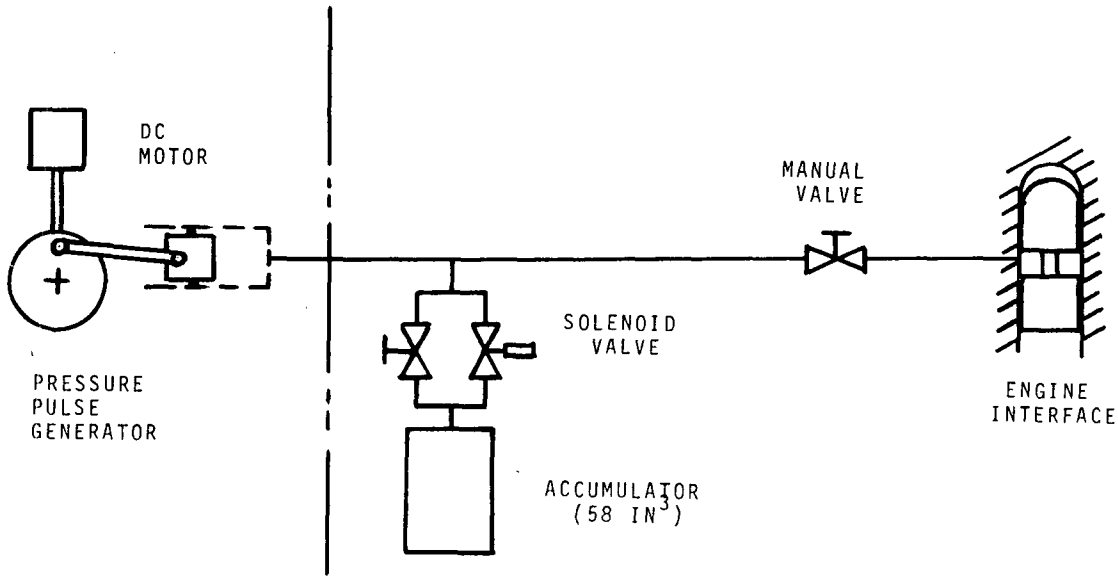


Figure 3-14. Pneumatic Starter Test Rig Pulse Flow System Schematic

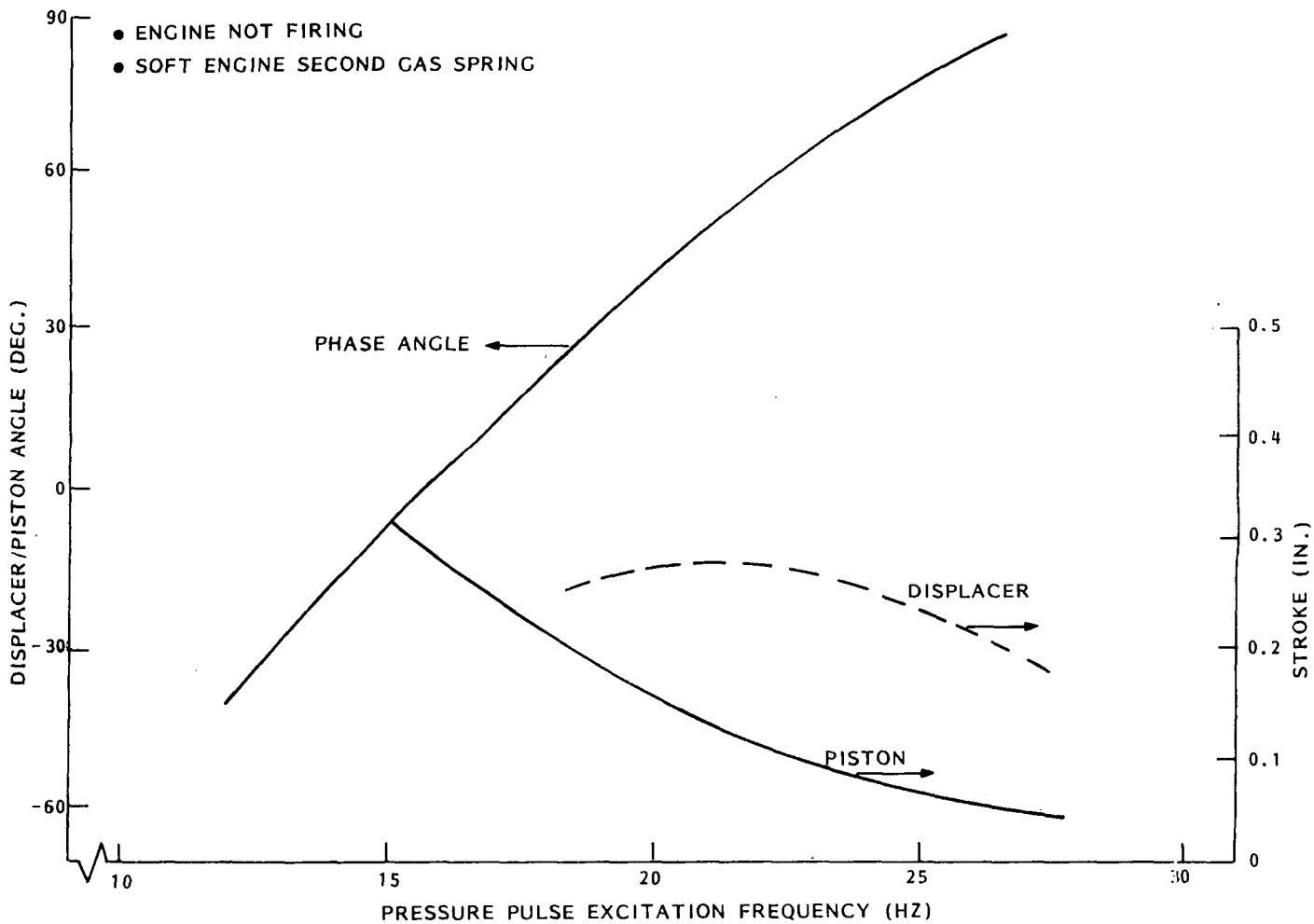


Figure 3-15. Dynamic Response of Displacer and Piston to External Pressure Pulses

and their respective strokes at various excitation frequencies. At low frequencies (below 15 Hz), the displacer lags the piston motion and would result in a non-startup condition if the engine was fired. At higher frequencies, a positive displacer/piston phase angle (displacer leads power piston) was obtained - a condition conducive to starting the engine.

After the engine was fired and the heater tubes reached elevated temperatures, the excitation frequency was set at 24 Hz and the engine was easily started after only a single pressure pulse. Many engine starts have since been successfully completed with this technique even at relatively low heater head tube temperatures.

Planned Activity for Next Period

No further activity is planned at this time in the development of the electromagnetic or engine housing impulse starter concepts, since the external pneumatic pressure-pulse generator appears to be much more attractive. A pneumatic pulse starter design, fabrication, and test effort is planned for the next reporting period, utilizing engineering design parameters established with the above mentioned starter test rig.

3.3.6 SEAL AND BEARING DEVELOPMENT

Status

All candidate seal and bearing test hardware fabrication and pre-test measurements were completed during this reporting period. The wear test stand was also assembled, and a five-hour trial test was conducted at design conditions. A potential oil contamination problem was identified during the trial run, resulting in a delay in the start of the test program. Oil droplets and/or vapors from the crank drive mechanism were observed migrating into the vicinity of the test articles, and the need to prevent this was recognized. A continuous gas purge adapter has been designed to counter the potential oil contamination and is being fabricated. This modification will be completed early in the next reporting period and the wear tests will then be started. A summary of the detailed seal and bearing wear test plan is presented in Appendix A.

Planned Activity for Next Period

The planned activity for the next period is to start and complete the seal and bearing test series, including the screening and extended wear test effort.

3.3.7 TRANSIENT LOAD ACCEPTANCE/OVERSTROKE PROTECTION

Summary

The transient load acceptance/overstroke protection system consists of a position sensor mounted to the engine power piston driving a modulated solenoid valve which, in turn, by-passes working fluid around the engine power piston. The by-passing of the working fluid (helium) controls the piston stroke. A Hall-effect position sensor was selected to provide the piston position signal.

Status

Sample Hall-effect position sensors were received and tested in several configurations to determine the proper placement of the sensor magnets. Based upon the results of these tests, a satisfactory configuration was identified and detailed drawings were completed for installing the components.

A circuit to drive a by-pass valve from the piston position signal was also designed, built, and functionally tested in conjunction with a sample load control valve. These tests showed that the electronic circuitry and associated sensor hardware successfully met their functional design requirements. A prime-hardware load control valve was received subsequent to these tests, and all of the electronic circuitry, sensor, and valve components are now available for installation into the engine.

Planned Activity for Next Period

The planned activity for next period is to install the transient load acceptance/overstroke protection hardware into a Proto 2 engine assembly, and to conduct system performance tests relevant to the operation of this hardware.

3.4 COMPRESSOR

3.4.1 SUMMARY

Several significant accomplishments in free piston compressor development occurred during this reporting period. Component testing, although not completed, was conducted at design conditions for the first time. This was made possible by the installation of a new component test stand. Compressor power consumption was measured at levels in excess of 3 kW. Compressor efficiency measurement, approximately 5 to 10 percent low, indicated a power loss mechanism which was identified as gas spring leakages. Modifications were made to correct this problem along with others while additional changes are planned. Two of the three gas spring pressure regulators, Carleton (prime) and Singer, were received and successfully tested, the first of these being tested in both component and integrated tests.

The compressor was assembled with the Stirling engine for integrated test. No compressor problems occurred in test, however, only limited performance measurements were possible on the compressor due to the engine's reduced power capacity. Integrated tests concentrating on compressor evaluation are planned next period.

3.4.2 HARDWARE CONFIGURATIONS AND MODIFICATIONS

Two compressor units underwent testing this period with various modifications to improve both functional and performance operation. Unit 1 compressor was tested in only one configuration and then assembled with the engine for integrated test. However, several modifications were made to the Unit 2 compressor prior to it's replacement of the first unit in integrated test. The key differences in each of these builds and configuration are as follows:

Unit 1

- Contoured internal suction passages to reduce pressure drop to lower cylinder.
- Three suction supply lines from the facility to duplicate the engine/compressor assembly.
- A suction valve with a normally slightly open position in upper cyclinder to reduce pressure drop.

Unit 2

- Includes modification of Unit 1 except standard, normally closed, valves were used.
- Internal suction gas flow was redistributed and balanced via flow inserts in the upper suction manifold passages.
- Connecting rod center porting between gas springs was increased to insure gas spring balancing.
- A one-piece bronze piston was installed for evaluation as a low cost alternative to the two-piece tungsten alloy (w-10)/cast iron combination.

Subsequent builds of the Unit 2 compressor also included:

- The Singer pressure regulator for control of gas spring reference pressure.
- The Carleton (prime) pressure regulator.
- A connecting rod with reduced center porting feature to minimize pumping loss between gas springs.

Following functional and limited component performance testing, the second unit replaced the first in integrated test. This Unit 2 build included:

- Suction flow inserts.
- Connecting rod with reduced center porting.
- Bronze piston.
- Two piston seal rings on each piston.
- Carleton pressure regulator (110 psia).

3.4.3 TEST RESULTS

The following paragraphs describe the compressor test results obtained during component functional, performance and integrated (with Stirling engine) testing. Functional testing refers to the various tests performed to evaluate physical operation of the compressor, i.e., piston centering, operating pressures, etc. The majority of tests were functional, performed on compressor assemblies with various modifications to eliminate off-center piston motion and gas spring leakages with associated power losses. Performance characteristics of the

compressor are obtained throughout all testing, however, incorrect functional operation has in some instances, adversely limited performance, thus requiring emphasis in this area. Component testing was initially constrained by test stand operational limitations to simulated engine operation at the low firing rate power levels (80°, 62°, 47°F). Full design power conditions (95°F up and 32°F down) were not obtained until the new facility was installed. Testing at the high power design point (1.3 inch stroke at 30 Hz) was therefore possible by the end of April.

Integrated engine/compressor testing was begun during this period but with limited compressor performance results due to engine operational limitations. In addition, the compressor was used as a load matching device by varying refrigerant suction and discharge pressure to off-design conditions.

Functional Testing

The primarily functional operation problems encountered in compressor operation were, as mentioned above, off-center piston motion and gas spring power losses. The off-center piston motion first encountered last period, is attributed to the larger suction gas pressure drop through manifolding and inter-connecting lines to the lower compressor cylinder. This produces a lower mean pressure in the lower cylinder forcing the piston off-center. The power losses in the gas springs were identified during high capacity testing conducted on the latter compressor assemblies (Unit 2). This power loss is produced by leakages out of the gas springs past the pistons, or between gas springs at the connecting rod seal. Unequal leakage from the gas springs also produces unbalanced pressures that contribute to off-center piston motion.

The first compressor assembly for this period (Unit 1) was modified by contouring internal suction passages and the addition of a third suction supply line from the heat pump facility to the unit. These modifications were only moderately successful in reducing off-center operation. This assembly performed satisfactorily until input power exceeded approximately 1.5 kW. In order to avoid a delay in engine testing, this unit was advanced to integrated test with the operational constraint of engine testing at only the low firing rate power levels.

Compressor component testing continued with assembly of the second compressor unit which received additional modifications to the internal suction supply. The latter consisted of

adding inserts in the upper manifold passages to redistribute and balance the gas flow to the lower cylinder. The connecting rod center porting between gas springs was increased and the low cost bronze piston was installed for evaluation. The bronze piston performed well while the other changes improved piston centering, thereby expanding the compressor operating ranges. However, some off-center motion still persisted in that piston biasing still occurred at low suction pressures < 75 psia.

In compressor assemblies completed to this point, the gas springs were referenced to suction pressure rather than a regulated 110 psia due to unavailability of the pressure regulators. The low reference (suction) pressure decreased the relative strength of the gas springs and their lack of centering influence permitted piston biasing.

The Singer gas spring pressure regulator, discussed in more detail in Section 3.4.4, was installed atop the compressor housing to control this effect by maintaining a preset pressure reference independent of suction conditions. Incorporation of the regulator extended the range of compressor testing as shown in Figure 3-16 and Table 3-1 which also compares test results of Unit 1 with Unit 2. Test of this pressure regulated Unit revealed several detrimental trends; drift of the piston center of motion, larger than predicted piston travel and leakage out of the gas springs. The gas spring leakages were most significant, accounting for approximately 25 percent of input power. This power loss is shown in the pressure-volume oscilloscope photo of Figure 3-17 which may be integrated to determine work and subsequently power.

The next compressor build incorporated several changes to alleviate these problems. The modified connecting rod, with reduced porting, and a second piston seal ring, positioned adjacent to the gas spring volumes to limit leakage, were installed. Functional operation, especially centering, were improved with this change; however, power losses of approximately 25 percent were still occurring. This power loss has the two-fold effect of reducing compressor useful work and creating unacceptably high compressor operating temperatures which aggravate the suction gas heat transfer. These combined factors contribute to low flow rates and higher than desired power consumption. To achieve the design goal, this problem must be corrected.

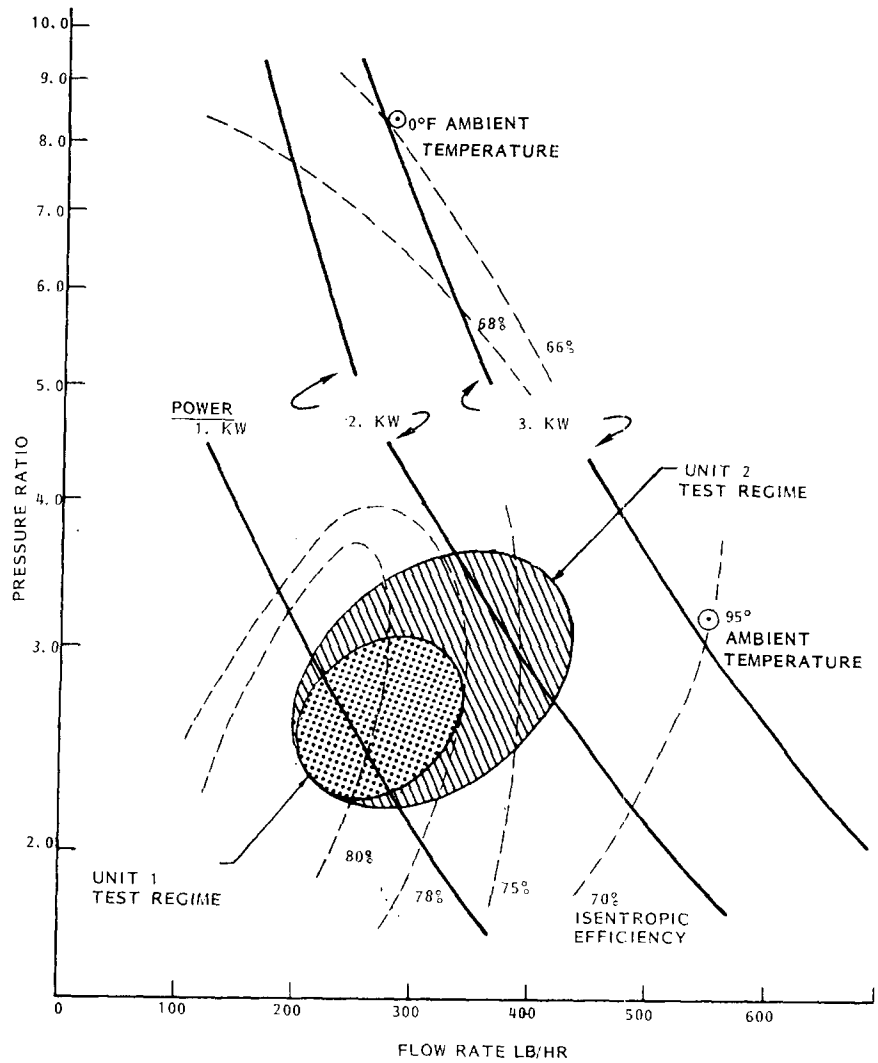


Figure 3-16. Proto 2 Compressor Test Results

Table 3-1. Comparison of Compressor Operational Test Conditions

	Unit 1	Unit 2
• Stroke	1.1 in.	1.1 in.
• Frequency	30 Hz	30 Hz
• Mass Flow (Max)	330 lb/hr	440 lb/hr
• Pressure Ratio (Max)	2.8	3.6
• Power (Max)	1.6 kW	2.5 kW
• Efficiencies (Range)	63 - 83 %	58 - 70 %
• Pressure Regulator	No	Yes

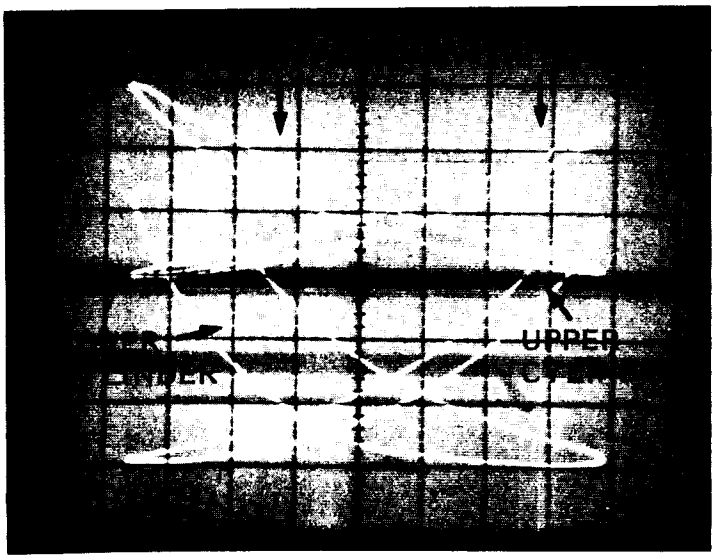


Figure 3-17. Compressor Pressure-Volume Diagram

Analysis of gas spring operation had indicated minimal pumping work associated with flow into and out of the gas spring external volumes. Minimal leakage between the two gas springs at the connecting rod was also determined. Further investigation identified gas leakage past the piston ahead of the second piston seal ring as the problem. The second seal ring was so positioned along the length of the piston to seal the gas spring external volume port at all times. However, this placement does allow leakage along the piston to the reference volume for a period of the compressor stroke.

A third piston seal ring projected to have minimal effect on the dynamics (drag) was identified as a solution to the problem. This will be installed and tested in the first compressor build during the next reporting period.

Following the compressor component performance testing as presented in the following paragraphs, this unit was assembled with the engine for integrated test.

Performance

The functional operation of the compressor has detracted from its overall thermodynamic performance. The losses in the gas springs directly reduce the available compression work

in the cylinders. This loss has average approximately 25 percent of the input power or, for example, 0.85 kW at a power input of 3.4 kW. This loss decreases compressor efficiency by about 15 percent, thus the unit would run at 53 percent rather than 68 percent efficiency.

These losses, as heat, increase the degree of suction gas superheat which increases the gas compressor work required for similar suction and discharge pressures. The suction superheat has average 50°F in the upper manifold and 80°F in the lower manifold, with a design goal of 50°F in both. This design goal was established for the compressors installed in the engine where additional superheat due to the interface ring and tubular springs is expected. The overall effect of this superheat can reduce compressor efficiency by about 5 percent. When the losses in the gas springs are corrected, the superheat levels should also be reduced, eliminating this problem.

Component Test

The component testing conducted to date has been primarily at the 80°F and 95°F ambient operating design points. Test results near the 95°F design point are presented in Table 3-2 while the ranges of measured performance are presented in Table 3-3 for both the 80°F and 95°F ambient degree day conditions.

Further testing is required on the next compressor build to evaluate performance with reduced gas spring losses. Tests at heating mode, low temperature degree days are also planned.

Table 3-2. 95°F Ambient Design Point Component Test Results

Frequency, Hz	29	~ 30
Stroke, inches	1.3	1.3
Flowrate, lb/hr	400	470
Power consumption, kW	2.6	3.6
Gas spring power losses, %	25	25

Table 3-3. Ranges of Measured Compressor Performance During Component Testing

~ 95° Day	Power	2.0	to	4.0	kW
	Flows (R22)	312	to	440	lb/hr
	Gas Spring Loss	24	to	27	%
	Efficiency				
	Inlet	44	to	51	%
	Manifold	49	to	56	%
	Efficiency w/o Gas Losses				
	Inlet	59	to	70	%
	Manifold	65	to	76	%
~ 80° Day	Power	2.6	to	3.0	kW
	Flows (R22)	300 lb/hr			
	Gas Spring Loss	~30 %			
	Efficiency				
	Inlet	39	to	44	%
	Manifold	44	to	50	%
	Efficiency w/o Gas Losses				
	Inlet	58	to	61	%
	Manifold	65	to	68	%

Integrated Test

No functional operation problems were encountered with either of the compressor units tested with the engine. The cast iron and bronze pistons with cast iron seal rings have performed and continue to perform satisfactorily in combination with the cast iron liner. Piston oil lubrication has not been a problem with either the gas springs referenced to suction or the gas spring pressure regulator. Much of the testing conducted was at off-design conditions using the compressor as a variable load matching device for the engine by establishing different suction/discharge pressure conditions. The various pressure levels change the characteristics, i.e., natural frequency and power consumption versus stroke and frequency, of the compressor thus controlling engine operation. This allowed more extensive evaluation of engine operation without hardware changes.

Compressor piston operation has been well centered in integrated testing as indicated by cylinder and gas spring operating pressures. The gas spring power losses can not be determined by direct pressure-volume diagram integration as the compressor piston-position sensor (LVDT) is not available, however, test results comparison to analytic prediction indicate somewhat reduced losses. This may be due to many factors, one of which is the combined effect of reduced compressor housing temperature from helium bounce space cooling and the bounce space operating pressures. These factors control the compressor cylinder expansion with respect to the piston and the effective piston to cylinder clearance. This gap identified as a leak path in component test would be less because of these factors and probably reduces the gas spring losses. Further testing is planned to evaluate and verify this effect.

Compressor performance evaluation has been limited because of the concentrated effort on engine operation. The engine's delivered power to the compressor was low, thus most of the testing was conducted at approximately 80°F ambient design point conditions. Some limited compressor performance is presented in Table 3-4. Integrated testing to more fully evaluate compressor performance has been scheduled.

3.4.4 COMPRESSOR REGULATOR

The two compressor gas springs have a reference port to maintain an initial pressure in the volume with the piston acting as a spool valve to seal the port on compression. The refrigerant gas pressure is regulated from discharge at this port via an absolute pressure regulator to a nominal value of 110 psia. The regulation of this pressure maintains effective and constant gas springs compensating for and limiting the effect of variable suction pressure (95 to 20 psia) in the cylinders resulting from changes in ambient conditions.

Three regulators were selected for testing and evaluation. The prime unit, made by Carleton Control Corporation and the unit which is a standard off-the-shelf refrigerant constant pressure expansion valve made by Singer Company were received and tested and are shown installed on

Table 3-4. Compressor Performance During Integrated Test

~ 80° Day	Power	1.0	to	2.0	kW
	Flows (R-22)	280	to	430	lb/hr
	Gas Spring Loss	25 % assumed			
	Efficiency				
	Ring	43	to	60	%
	Manifold	50	to	69	%
	Est. Efficiency w/o Gas Loss				
	Ring				
	Manifold	67	to	92	%
~ 95° Day	Power	1.02 kW			
	Flows (R22)	219 lb/hr			
	Gas Spring Loss	25 % assumed			
	Efficiency				
	Ring	56 %			
	Manifold	68 %			
	Est. Efficiency w/o Gas Loss				
	Ring	75 %			
	Manifold	90 %			

the compressor in Figures 3-18 and 3-19. The third, a backup, made by AUSCO Incorporated was scheduled for delivery by the end of July

The Singer regulator, obtained first, was installed and tested on the compressor during component testing in February. The Singer regulator was not specifically designed for this application with the severe vibratory environment and high external pressure of the engine bounce space. Initial bench testing of the unit indicated regulated pressure level variations outside the design tolerance band (± 2.5 psia) and it was sensitive to both the amount of regulated flow and variations in discharge pressure. However, the unit was adjusted on the compressor to compensate for these effects and testing successfully conducted. Total accumulated test time under vibratory conditions was approximately 12 hours.

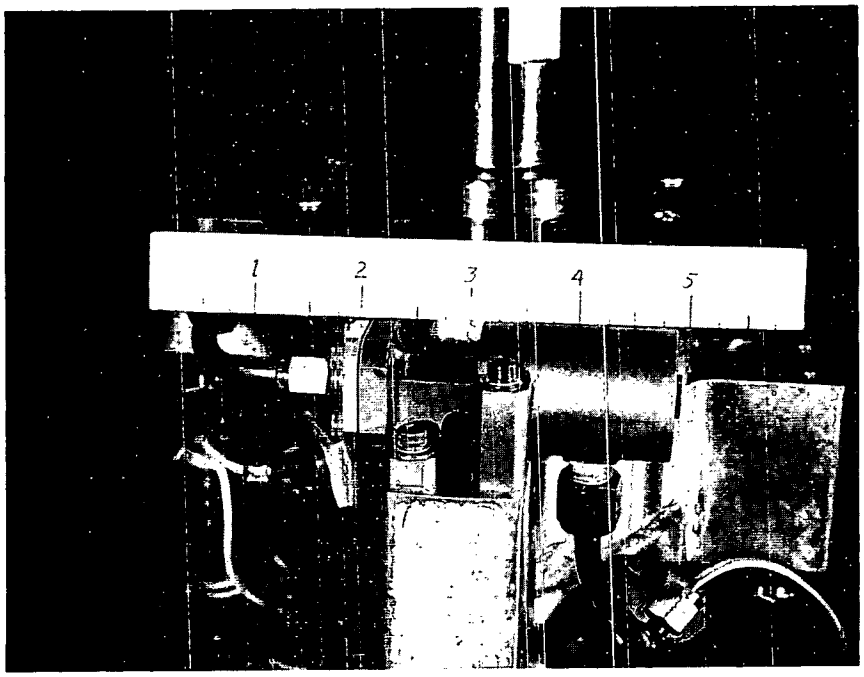


Figure 3-18. Carleton Pressure Regulator

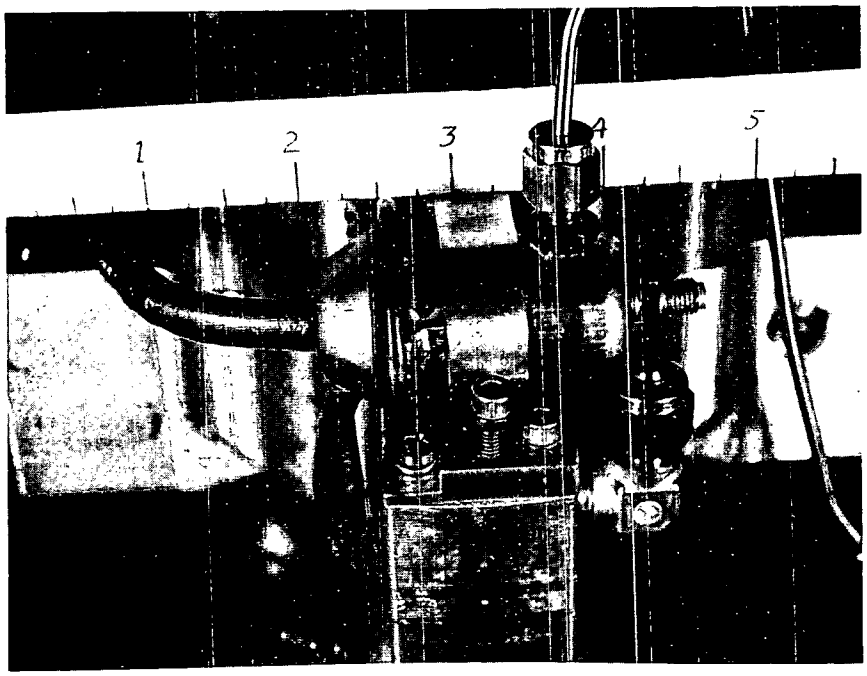


Figure 3-19. Singer Pressure Regulator

Two different regulator settings were evaluated - 110 and 125 psia, with the former being the design value. Results of these tests are tabulated in Table 3-5. The effect of higher regulated pressure is obvious: reduced mass flowrate, and reduced operating pressure ratio capability.

Table 3-5. Compressor Regulator Operation

Parameter	Measured Values			
Reg. Pressure, psia	125	110	125	110
Ambient Day Conditions, ^{o}F	62	62	47	47
Pressure Ratio	2.4	2.5	2.6	3.1
R-22 Flow, lb/hr	300	340	230	280
Power, kW	1.3	1.5	1.4	1.7
Efficiency,	69	70	65	64

Note: 1.12 inch stroke at 27 cps.

The higher regulated pressure stiffens the gas springs, reduces the piston stroke, reduces mass flowrate and, in turn, consumes less power for identical test conditions of driving stroke and frequency. In engine operation, where power input is fairly constant with varying stroke, the higher regulated refrigerant pressure would force the engine to a longer stroke and the compressor would absorb the engine power output. Thus, it is possible to tune or match the engine/compressor via adjustments in the regulated gas spring pressure level.

The prime pressure regulator (Carleton) was received and installed on the compressor during the latter portion of this period. This regulator accurately controlled the regulated pressure on the compressor such that no additional adjustments were necessary as was the case with the Singer unit. This unit performed satisfactorily during the 10 hours of testing conducted on the compressor stand, with performance similar to that obtained with the Singer regulator. The compressor assembly, including the Carleton regulator was then assembled with the Stirling engine for integrated test.

The compressor in integrated test prior to this was configured with the gas springs referenced to suction with majority of testing being performed in the cooling mode where suction pressure

is fairly constant above 90 psia. This unit will now permit testing at the low suction pressure level heating mode conditions. Although this compressor with regulator was tested at design conditions on the component test stand, engine power output limitations prevented the same conditions being achieved in integrated test. The unit has performed satisfactorily during all integrated testing, accumulating approximately 30 hours of operation. The Carleton pressure regulator unit has met design requirements, surviving the vibratory (1.3 inch stroke, 30 Hz) and external pressure (1170 psia) requirements with only life verification remaining to be demonstrated. Further testing is planned to evaluate heating mode operational performance of the unit in integrated test and life verification will occur in parallel with the engine life testing.

3.4.5 TUBULAR SPRING COIL

The tubular spring coil which successfully completed 2000 hours of life testing was instrumented to measure stress levels during dynamic operation. This coil of 0.5 inch diameter Almar 362 tubing had been subjected to over 2.0×10^8 cycles of alternating stress, well past the number of cycles where the material strength stabilizes under fatigue conditions. This testing was performed in a test facility providing a 1.75 inch coil deflection at 30 Hz, actually in excess of the design stroke of 1.3 inches.

The stress levels resulting from the measured strains, along with predicted values are shown in Table 3-6. The measured values agree quite well and are below the fatigue endurance limit of 90 kpsi for normal stress and 45 kpsi for shear stresses. These comparisons are shown in Figure 3-20.

The predicted natural frequency of the coil is 2.5 times (with insulation, 3 times without) the engine operating frequency. Stress magnification due to possible resonance in the coil was not expected and did not occur although harmonics in the measured stresses were observed. These harmonics were not judged to be a problem; however, future tube configuration should maximize this natural frequency to limit the harmonics. There is also the possibility that the test facility could have induced these harmonics. The coil is firmly supported at one end and the other attached to a drive piston which produces the required stroke at frequency. Vibration and rotation of the piston over it's travel could produce non-uniform (harmonic) coil motion and

Table 3-6. Tubular Spring Coil Stress Levels

Stress Region	Stress (psi)		
	Design	Test Values	
		Predicted	Measured
Coil - 1000 psid External Pressure Hoop	- 7150		
Long.	- 3600		
Dynamic Stroke (inch) (@ 30 Hz)	1.3	1.75	1.75
Coil Straight End - Norm.	± 26,850	± 32,600	± 36,000
Coil End @ 90° Bend - Shear	± 14,050	± 17,100	± 18,800
Coil Turns - Shear	± 13,600	± 16,550	± 13,100

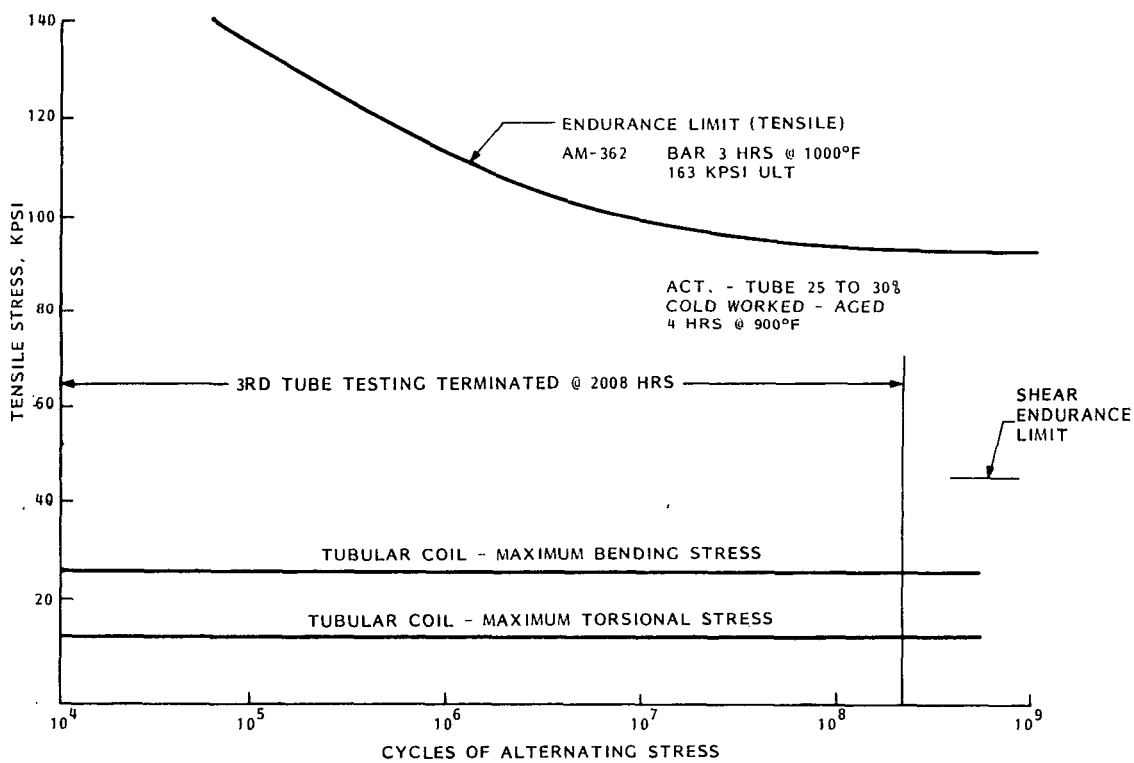


Figure 3-20. Fatigue/Endurance Data AM-362 Material

stress. The major facility modifications that would be necessary to eliminate this possibility were not performed because of the measured results and the safety margin that already exists on stress levels.

3.4.6 PLANNED ACTIVITY FOR NEXT PERIOD

Component testing will continue next period with several modifications to reduce losses and improve efficiencies. The major factor to accomplish this should be the addition of the third piston seal ring. Following this, more extensive testing at the various heating and cooling mode design points is planned.

Integrated testing at conditions specifically established to evaluate compressor operation are also planned. This will permit a more accurate evaluation and verification of compressor performance when driven by the Stirling engine. Heating mode operation is also necessary for identification of suction gas heat transfer effects on the compressor and engine. The modified compressor from component testing will be installed in next engine build for performance evaluation.

3.5 CONTROLS

3.5.1 SUMMARY

The major accomplishments of this reporting period are the fabrication of the controls for the second Proto 2 Unit, controls development for the magnetic starter, transient load acceptance development tasks and continued integrated test support. At the end of the reporting period, assembly of the second Proto 2 unit controls was to the point of installation of the control panel into the outdoor unit. Wiring and testing of the panel was initiated but has not been completed.

3.5.2 PROTO 2 UNIT 2 FABRICATION

Fabrication of the second unit followed the same steps used previously for the first unit. The first step was to wire wrap the circuit board, check for continuity and make corrections after which the electronic components were installed. The board was then functionally tested. Next, mounting of the components on the orifice panel was accomplished and the panel wiring installed. Subsequent to continuity checks the power was applied and the entire panel was tested with the

control panel test box. When the test was satisfactorily completed, the panel was installed in the outdoor unit.

3.5.3 MAGNETIC STARTER CONTROLS

The design of the Proto 2 engine included hardware for a magnetic starter. These components consisted of a core, which was part of the displacer rod assembly, and a driving coil which was mounted inside the power piston. It was originally configured to provide an upward directed force on the displacer when power was applied to the coil. This was to start the engine by either providing enough motion of the engine components on one pulse or by slowly building the displacement amplitude of the engine components by repetitive pulses.

The excitation to the coil was from a pulse generator which was enabled by a push button. The pulse width and repetition rate were manually adjustable. The pulse amplitude was adjustable and was set to a level where further amplitude increases did not result in greater or faster displacer motions. However, the engine could not be started using this technique.

By changing the core design and position with respect to the coil, it was planned to obtain both upward and downward directed forces on the displacer. A circuit was designed and breadboarded to coordinate the excitation of the coil with the displacer position by using the signal from the displacer to power piston LVDT. During tests with this circuit, it was found that the core position had not been modified enough to permit sufficient downward force on the displacer. Further starter development using this electrical approach were deferred in favor of the pneumatic starter development.

3.5.4 TRANSIENT LOAD ACCEPTANCE AND OVERSTROKE CIRCUITS DEVELOPMENT

The approach to this task was divided into two categories. The near term approach was to use the LVDT sensor to drive a bypass valve when the piston position approached either of its mechanical stops. The long term approach assumed that the LVDT would not be available and that the valve would have to be driven by an alternate sensor which must be developed.

For the near term, a circuit was designed which compared the LVDT signal with adjustable references, amplified signals which exceeded the set points and provided a compatible drive

signal to the control valve. This circuit was built in a version which will allow testing with the valve on the intergrated engine test stand.

The alternate configured circuitry was designed with a Hall-effect sensor. This provides a voltage output which is proportional to the strength of the magnetic field passing through it. Permanent ceramic magnets were used to generate the field. Several physical configurations of these magnets were tested with a Hall-effect sensor until the sensor output voltage was proportional to the sensor position with respect to the permanent magnets, over the desired range of travel. The chosen configuration provides the most nearly linear characteristic with adequate output signal level. Drawings were prepared for the required brackets to mount both the sensor and the magnets. The magnets will be mounted on the adaptor housing (between the power piston and the compressor) while the sensor will be mounted to the interface ring. The brackets were fabricated, the assemblies of the magnets and sensor completed and the circuitry is now ready for test.

An electrical circuit was designed to condition the sensor output to be similar to that obtained with the LVDT sensor. The output of this circuit can then be used with the same near term electronics to drive the bypass valve. This circuit was also fabricated and is available for testing with the integrated engine.

A sample valve was obtained and used to verify the compatibility of the drive circuit to valve interface. The primary valve has been received from the vendor and is also available for test.

3.5.5 PLANNED ACTIVITY FOR NEXT PERIOD

The planned activity in the controls area will be limited to modifications to the hardware identified through testing.

3.6 AIR HANDLERS

3.6.1 SUMMARY

The main accomplishments of the reporting period were to complete the Air Handler component test series on Proto 2 Unit 1 and to assemble the major portion of Proto 2 Unit 2; the Field Test Unit. The assembly of Unit 2 was 90 percent complete at the end of the reporting period (see Figure 3-21).

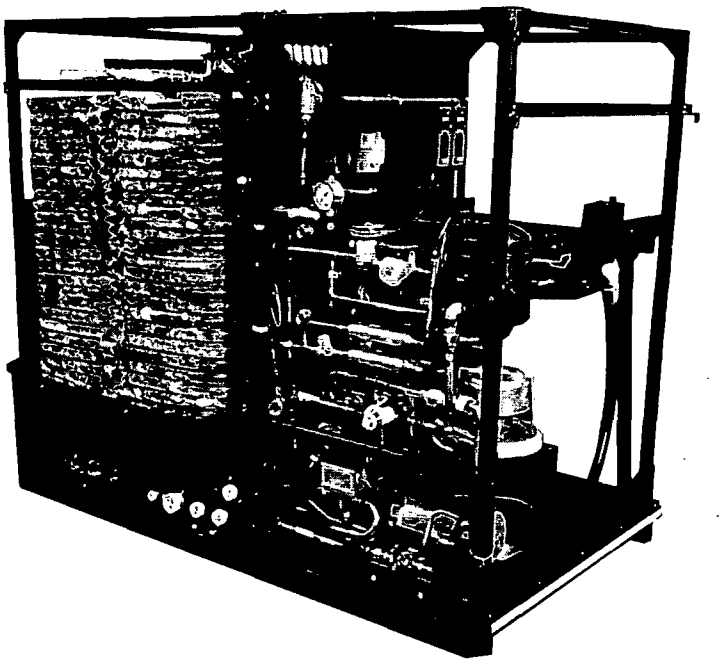


Figure 3-21. Proto 2 Field Test Unit Approximately 90 Percent Complete

3.6.2 PROTO 2, UNIT 1 TESTING

The test series, conducted in accordance with the Component Test Plan, utilized a commercially available electric motor-driven hermetic compressor in place of the free piston Stirling engine-driven Linear Inertia Compressor. Tests were conducted to evaluate key performance characteristics of the heat pump refrigerant cycle. Tests were conducted to determine:

- Optimum Refrigerant Charge
- Capacities at the Heating and Cooling Rating Points
- Performance at Off-Design Conditions
- Parasitic Power Losses
- Frost Accumulation/Defrost Characteristics

Refrigerant Charge Optimization Tests

Charge optimization tests were conducted in the heating and cooling modes at the high rating points. The test results are shown in Figures 3-22 and 3-23, respectively. The amount of refrigerant coincident with the predominant performance peak is the main factor in determining optimum charge. (Capacity and efficiency usually track fairly closely.) In the case of the air handlers, the predominant performance peak occurs in the cooling mode at 18.5 pounds of R22 - see Figure 3-23. The optimum charge therefore is determined to be 18.5 pounds. The charge response in heating, however, occurs as a plateau rather than a discrete peak - reference Figure 3-22.

Rating Point Capacity Tests

The pertinent heating and cooling mode capacity tests were all completed and the results are tabulated in Table 3-7. The electric compressor used in this test series did not have the capability to modulate capacity, thus producing a deviation from the targeted R22 mass flow rates at the lower rating points. Consequently, appropriate corrections were made to the measured capacities at a portion of the rating points, so that they corresponded to values that would result from R22 mass flow rates equal to the primary system's predicted values. The

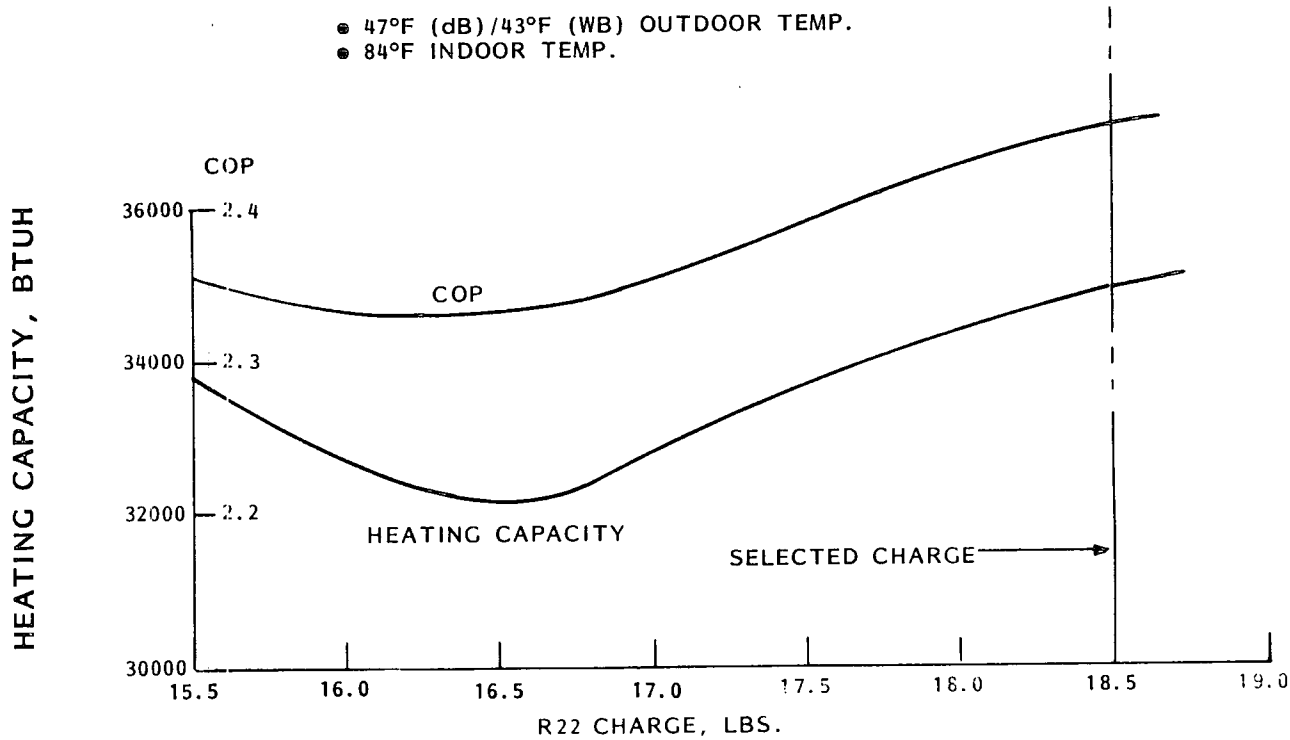


Figure 3-22. Heating Charge Optimization

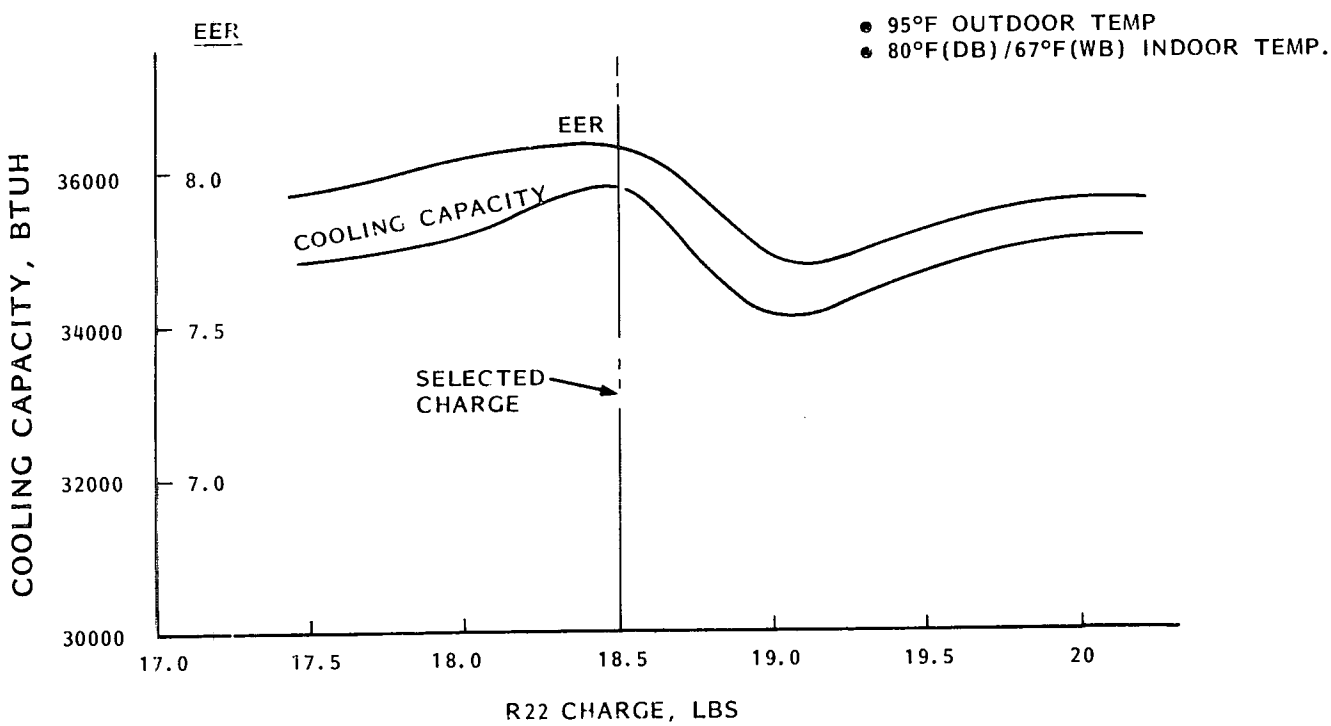


Figure 3-23. Cooling Charge Optimization

deviation between predicted values and corrected measured values is in the range of 5 percent for most points, and is considered to be acceptable. The condenser capacity at 80°F outdoor cooling, however, exceeded that predicted by some 13 percent. This is attributed in part to the fact that the efficiency of the calibrated electric compressor motor decreases at light loads resulting in increased condenser load.

Table 3-7. Proto 2 Air Handler Test Results

Temperature, °F		Mode	Condenser Capacity, Btuh		Evaporator Capacity, Btuh		R22 Flow lb/hr
Ambient	Indoor		Measured*	Predicted	Measured*	Predicted	
95 DB	80 DB 67 WB	Cooling	47180	46900	35920	36000	530
80 DB	80 DB 67 WB	Cooling	38360	34000	28510	30000	400
47 DB 43 WB	70	Heating	26420	27300	20210	21300	295
17 DB 15 WB	70	Heating	27780	29200	21000	20000	300

* Normalized from 3-1/2 ton test compressor flow to 3 ton level

Off Design Performance

The off-design performance characteristics of a heat pump are evaluated by varying indoor air flow ± 10 percent from the nominal value (i.e., 1200 scfm in this case) and determining the resultant change in capacity and efficiency. The performance changes produced in the test were so small they were obscured by the error due to measured accuracy of the test facility. This speaks favorably for the unit's relative insensitivity to fluctuations in air flow that could result from a partially restricted air filter or a partially blocked duct.

Parasitic Losses

A 20 percent reduction in parasitic power consumption below the design goal has been achieved. Successful load matching of the various drive motors was the principal contributor to this improvement - see Table 3-8.

Table 3-8. Parasitic Power at Proto 2 Design Conditions

	Power, Watts	
	Projected	Test Results
Hydronic Pump Motor	250	140
Indoor Blower Motor	300	350
Outdoor Fan Motor	450	320
Controls	70	35
Combustion Blower Motor	30	30
Total	1100	875

Frost Accumulation/Defrost Characteristics

The frost accumulation test was conducted at the outdoor ambient rating point conditions of $32^{o}F$ (DB)/$30^{o}F$ (WB). The indoor ambient rating point of $70^{o}F$ was not held, however, because of a facility limitation. While the higher prevailing indoor ambient, i.e., $84^{o}F$, did reduce the absolute heating capacity of the air handlers, a meaningful frost related characteristic was established. The characteristic defined is the relationship between outdoor coil ΔP due to frost buildup and COP as a precentage of COP produced with a frost free coil at the same set of conditions - see Figure 3-24.

The time/temperature terminate function of the defrost control was bypassed for this test series so that the relationship between frost build-up and COP could be explored over a wider range than normally allowed. With the defrost control activated and set at the 90 minute interval, defrost would have been initiated as the outdoor coil ΔP was passing through 0.19 inch H_2O. The COP would have degraded at that point to 97 percent of the frost free COP. The other characteristic established by this test series was the defrost time necessary to yield a frost free coil. Figure 3-25 shows a typical time/temperature relationship for the air handlers. It also shows that the coil will be frost free in just under 3 minutes.

3.6.3 DESIGN CHANGES - PROTO 2 UNIT 2

The base pan design of the coil component in the Outdoor Unit was modified to improve the handling of condensate which could otherwise accumulate inside of the base pan as ice during

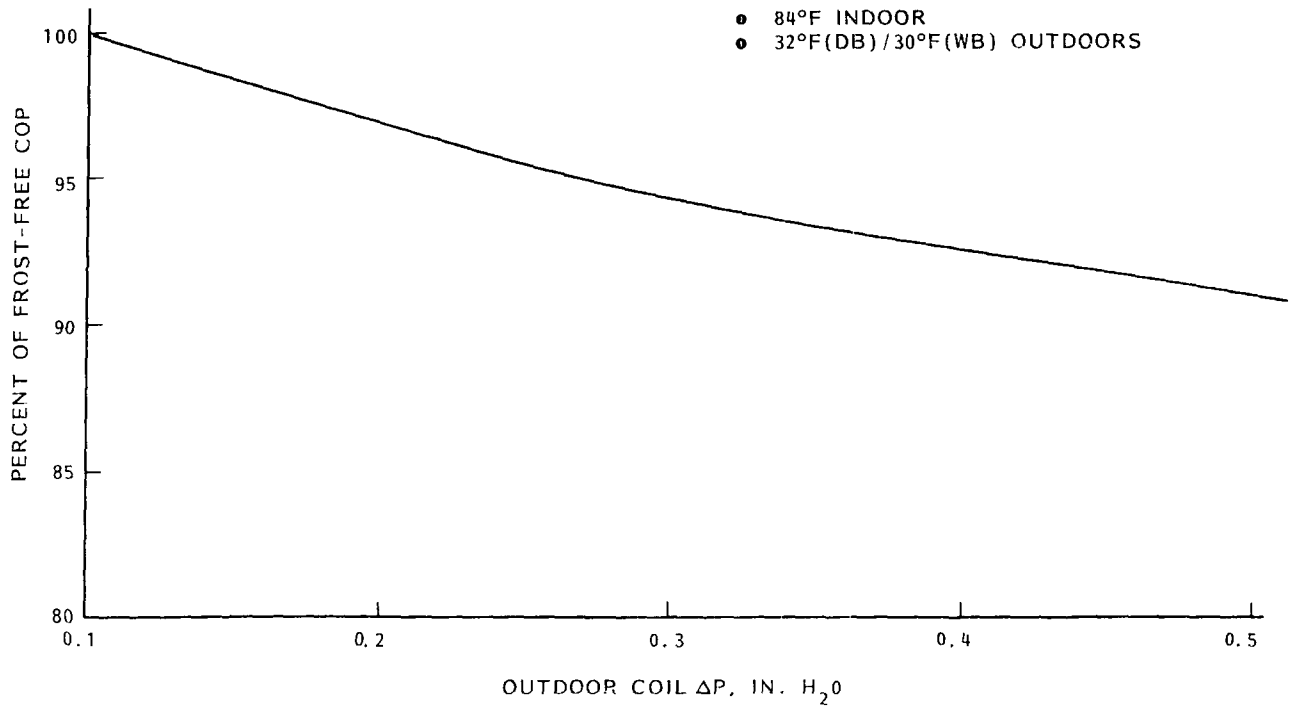


Figure 3-24. Effect of Frost Build-up on COP

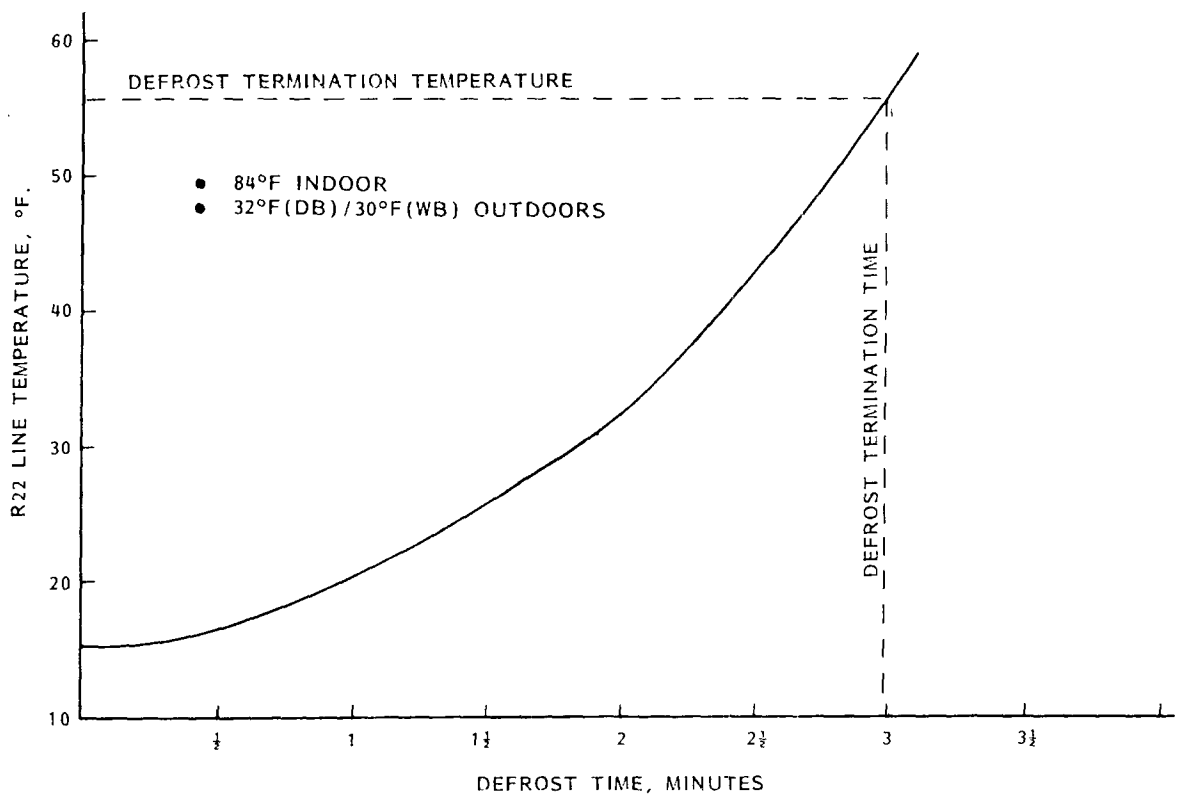


Figure 3-25. Defrost Response Time

frosting conditions. The base pan drainage holes were enlarged and a "semi-demand" heat source was added adjacent to the holes to prevent blockage due to ice build-up. The heat source used is a loop of bare 3/8 inch O.D. copper tubing in the R22 circuit, which is an integral part of the liquid line. It is located downstream from the defrost termination bulb site during the defrost cycle. The heat needed to melt base pan ice is delivered to this loop only during the final stages of a defrost cycle thus providing an energy efficient heat source.

A number of design changes were made to the refrigerant and hydronic circuits in the Outdoor Unit to cut down on line lengths and reduce clutter wherever possible. The hydronic circuit involving the expansion tank was redesigned so that the tank became a flow-thru element thus simplifying the task of properly charging the loop.

A design change was made to the Indoor Unit which addressed user comfort during the defrost cycle for the field test system only. This involved the addition of supplemental heat in the form of electrical resistance heaters typical of that found in most residential heat pump applications. In the final product configuration, this function will be provided by engine waste heat in conjunction with the gas fired extended heating capacity subsystem.

3.6.4 FABRICATION AND ASSEMBLY OF UNIT

The fabrication of all parts, including redesigned parts and exterior cabinet parts, is completed. Assembly of the Outdoor Unit was 90 percent complete at the end of the reporting period.

3.6.5 PLANNED ACTIVITY FOR NEXT PERIOD

During the next reporting period, fabrication of the Unit 2 components will be completed to support preparations for the Field Evaluation Test.

SECTION 4

INTEGRATED TEST

SECTION 4
INTEGRATED TEST

4.1 ENGINE INSTALLATION

The first Proto 2 Free Piston Stirling Engine Assembly was installed in the Integrated Test Facility early in this reporting period. Photographs of the installation are shown in Figures 4-1 through 4-3. Figure 4-1 shows the engine assembly in its upright/operational position. Figure 4-2 shows a similar orientation, but with the pressure vessel removed to expose the compressor and refrigerant interface tubular spring coils, and Figure 4-3 shows the unit in its tilted position for servicing the combustor. The demonstrated tilt feature was designed into the engine mount assembly to permit servicing the combustor without the need for lifting the engine out of its cradle, and, therefore, without having to vent the helium charge or disconnect the refrigerant lines and/or engine coolant lines from the unit.

4.2 TEST SUMMARY

The first Proto 2 engine assembly in its first build configuration was started during the first start-up attempt and operated for a total of 30 minutes for both engine and test facility check-out purposes. This is considered a noteworthy accomplishment. The unit was operated with a low firing rate of approximately 24,000 Btuh and with the second gas spring bypassed for the initial start-up. Start-up was accomplished using the pressure-pulse technique developed during the Proto 1 FPSE test effort. Subsequent testing throughout the reporting period consisted of the evaluation of several starting mechanisms and techniques plus performance testing to investigate performance variations resulting from selected engine configuration and/or component changes. The following paragraphs describe the engine configurations and test variables evaluated. Detailed test results are presented in Section 2.

Engine Starters

Three different starter concepts were tested, an electro-magnetic starter built into the engine power piston/displacer assembly, an external pressure-pulse generator mechanism, and an engine housing impulse method. Reference is made to Section 3.3.5 of this report for a detained description of these starters, the performance of each, and the planned engine starter activity for next period.

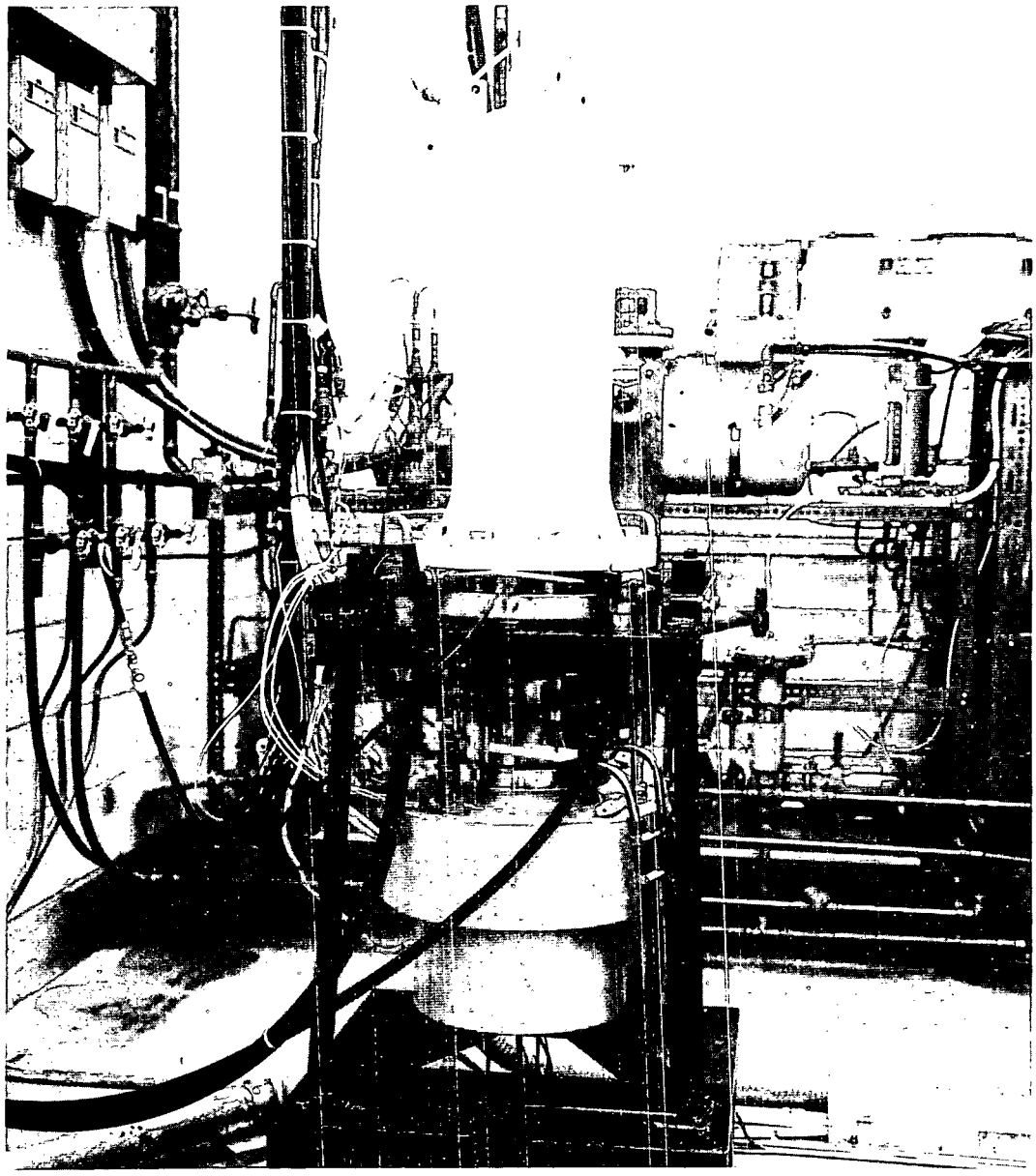


Figure 4-1. Proto 2 Combustor/Engine/Compressor Integrated Test

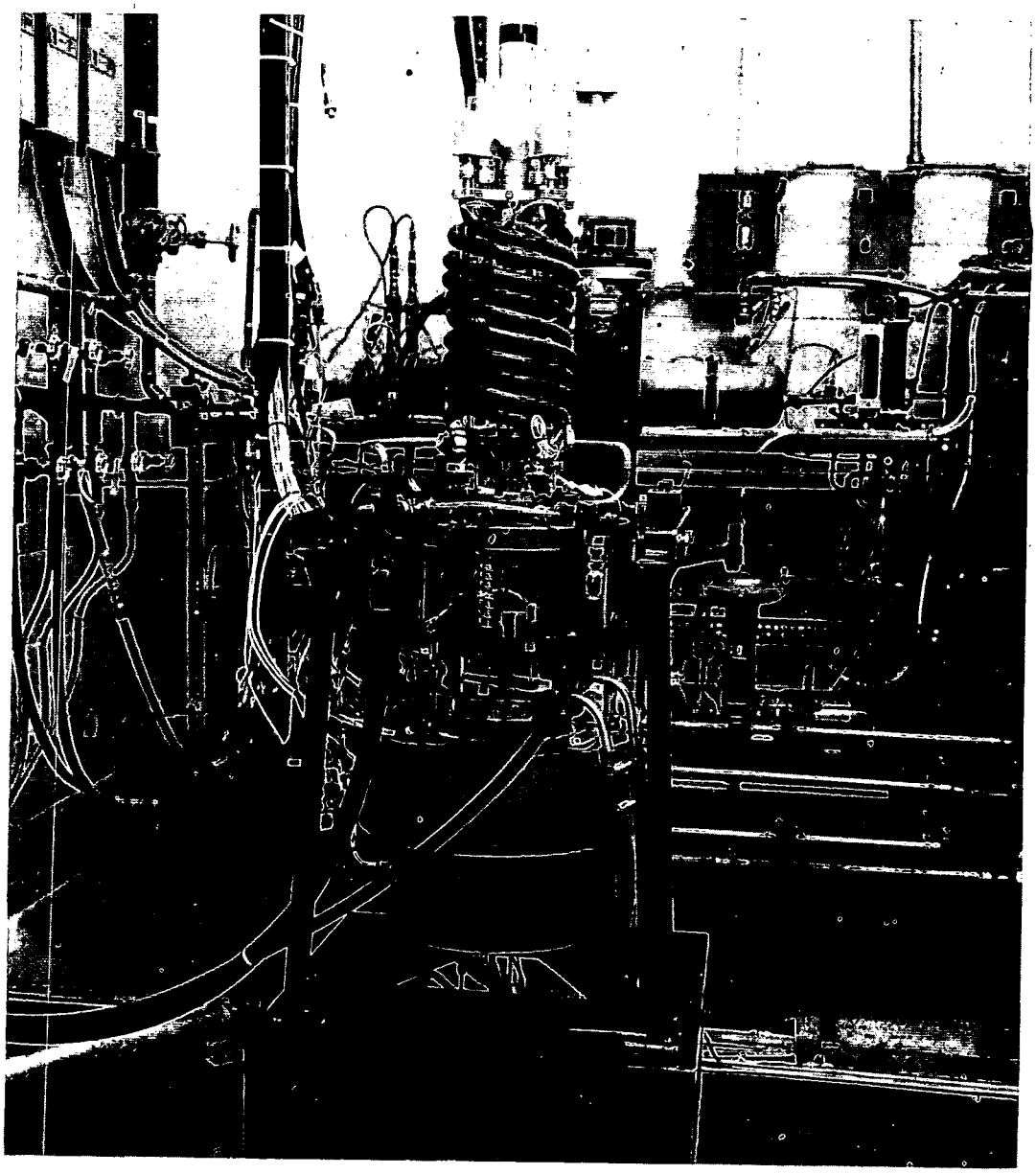


Figure 4-2. Proto 2 Integrated Test Unit Pressure Vessel Removed

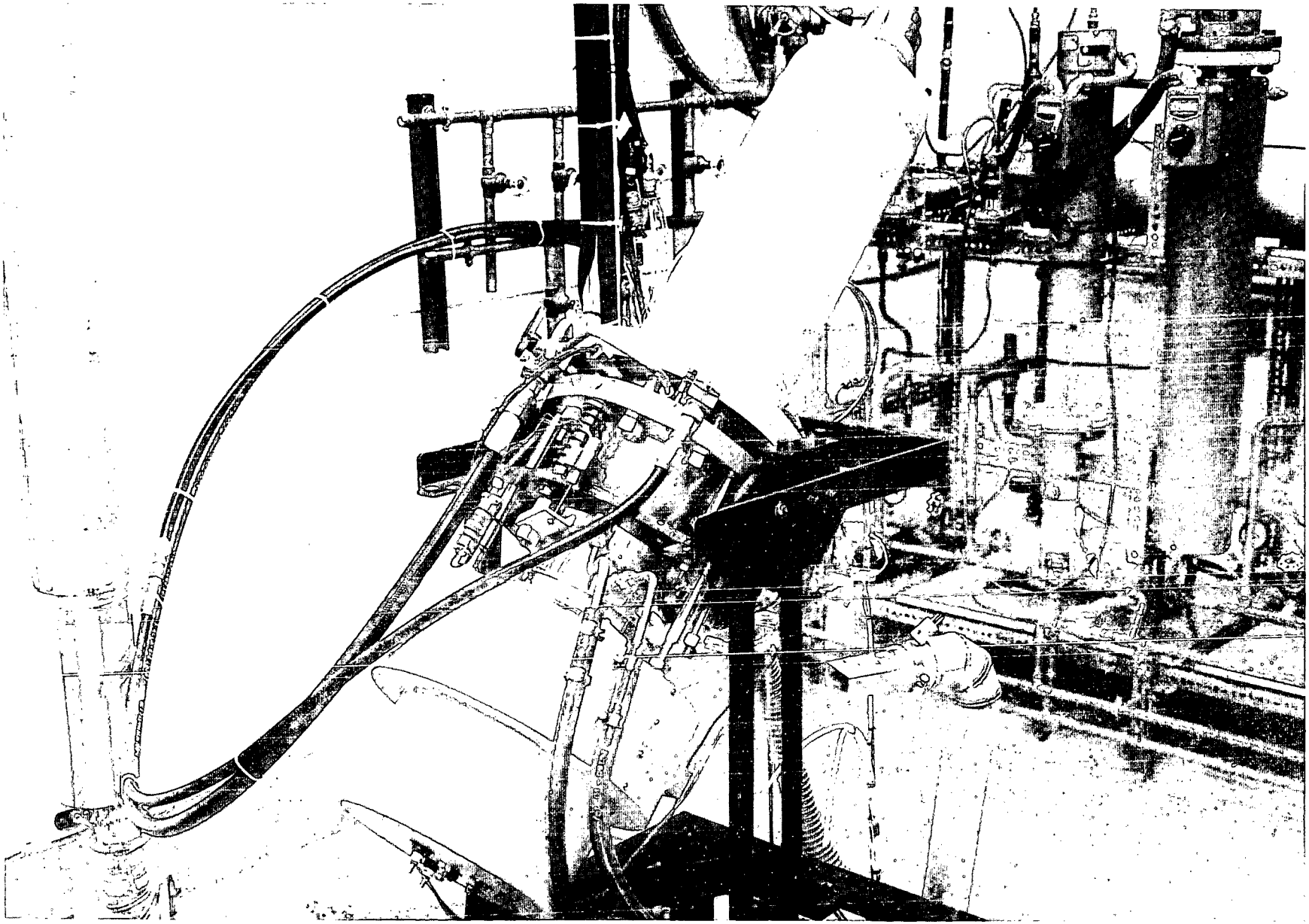


Figure 4-3. Proto 2 Combustor/Engine/Compressor Unit With Engine Support Structure

Engine Configurations Tested

A total of 41 different engine build configurations were tested during this reporting period, with an accumulated running time of approximately 82 hours. Engine/component parameters changed during these tests include: regenerator matrix porosity, displacer rod area and gas spring volume, displacer centering port size and location, displacer assembly mass, second gas spring area and volume, second gas spring vent port area and hole pattern, and compressor Unit 2 to mate the engine with the latest compressor assembly. Reference is made to Section 3.3.2 of this report for a more detailed description of the variations tested.

Tests Completed

Initial testing of the Proto 2 FPSE showed that engine starts could not be accomplished with the second gas spring fully effective as originally designed. Start-ups were made with the second gas spring by-passed. Testing, therefore, included experimentation directed toward achieving engine starts with the second gas spring partially effective to increase the engine operating frequency as near as possible to the design goal of 30 Hz. The problem was successfully resolved by incorporating a porting change into the second gas spring liner to increase both the centering port area and its effective length. This change unloaded the gas spring at its mid-stroke position, but retained the spring effect at either end of the stroke. Figure 4-4 presents a sketch of one of the second gas spring liner modifications incorporating the associated vent ports.

With an engine configuration that incorporated a partially effective second gas spring which could be readily started, the test effort was directed toward investigating the engine performance trends as a function of various component changes and operating conditions. The varied operating conditions included:

- Engine helium charge pressure levels between 700 and 1170 psia
- Compressor suction pressure levels between 70 and 160 psia
- Compressor pressure ratios between 1.3 and 3.2
- Firing rates of 20,000 through 56,500 Btuh

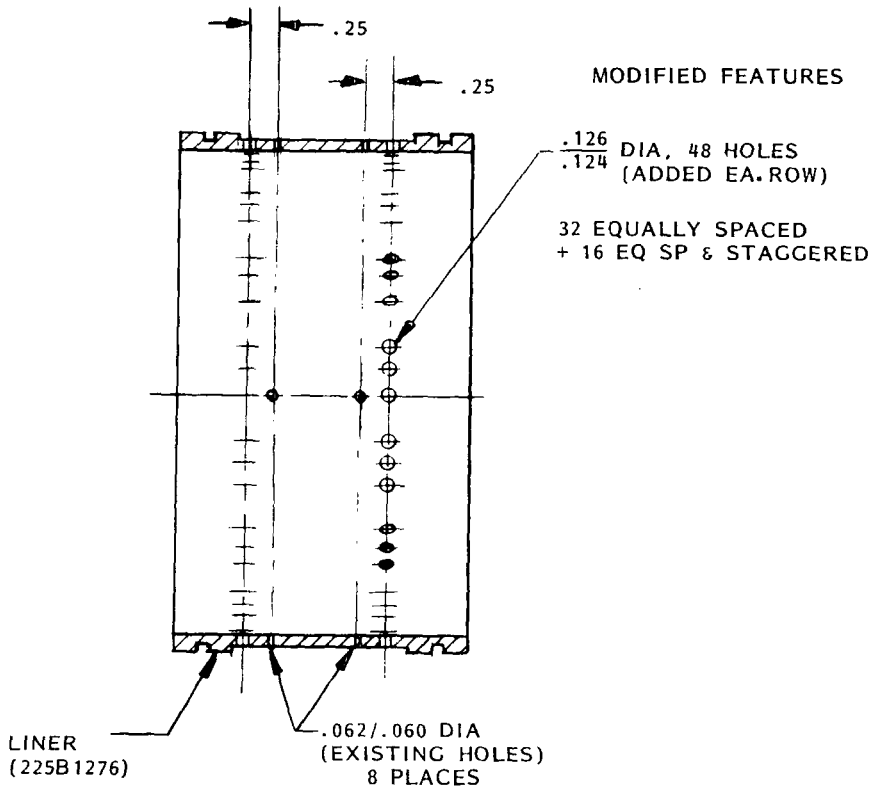


Figure 4-4. Modified Second Gas Spring Liner Incorporating Vent Ports

Significant performance improvements were attained as the engine dynamics were improved over that of the original build configuration. Reference is made to Section 2 for a description of the engine performance results versus build history/configuration.

A combustor flame-out problem was also encountered during the above noted test effort. The problem was found to be the result of auto-ignition of the fuel/air mixture within the burner element, with subsequent flame-out being produced via the flame detector circuitry as the flame withdraws from the detector area. The problem was found to be a combination of higher than predicted pre-heater air temperatures, and a flow stagnation condition within the burner support element area. The pre-heater air temperature was reduced to the desired value of approximately 900°F by removing the insulation from around the pre-heater section of the combustor. This was a temporaty fix to expedite testing. Permanent changes will not be addressed until engine performance testing is completed. The flow stagnation problem was resolved by the installation of a conical shaped flow deflector in the top of the burner element.

Satisfactory combustor operation has subsequently been demonstrated at firing rates of 20 k through 50 k Btuh. A total of 82 hours of running time has been accumulated on the integrated test engine assembly to date.

4.3 PLANNED ACTIVITY FOR NEXT PERIOD

The planned activity for next period is to complete the test program providing performance-trends versus engine configuration/parameter changes, so that Proto 2 performance optimization can be realized. Also, additional engine heater head, regenerator, and cooler modifications are being analytically evaluated at this time, and future Integrated Test efforts will be implemented to test any hardware changes completed as a result of this study.

SECTION 5

TEST FACILITY UPGRADING

SECTION 5

TEST FACILITY UPGRADING

5.1 INTRODUCTION

Additional test facility modifications have been completed to accommodate changes dictated by configuration differences between Proto 1 and Proto 2 assemblies. The following paragraphs describe the modifications completed on the Compressor and Integrated Test Stands.

5.2 COMPRESSOR TEST STAND

Modifications were made to the Compressor Test Stand to increase stroke and operating frequency.

Test Stand Modifications

1. The re-designed compressor drive mechanism with built in counterbalance capability was installed, aligned and checked out for optimum operation.

2. A forced air cooling system for the cross-head drive bearings was installed to allow continuous running.

3. New LVDT's to measure compressor/piston stroke were received and installed.

5.3 INTEGRATED TEST STAND

Test Stand Modifications

1. Orientation of both condenser and evaporator in the R-22 loop for the compressor was changed to eliminate occasional slugging of the compressor at low flow rates.

2. An auxiliary vernier control valve was added to permit fine adjustments to the firing rate.

5.4 PLANNED ACTIVITY FOR NEXT PERIOD

No major activity is planned for subsequent reporting periods other than follow-up monitoring of test stand operational performance. Sensor replacements and minor adjustments to the test stands and related facilities will be accomplished on an as required basis.

SECTION 6

FIELD EVALUATION TEST

SECTION 6

FIELD EVALUATION TEST

6.1 INTRODUCTION

During the January-June 1980 reporting period, the field evaluation and test activity included selection, ordering and receipt of the major components for the site instrumentation rack, continuing preparation of fabrication and assembly layouts and sketches, and submetering of the gas and electric energy consumption by the existing test site gas furnace. Long lead items of the GE-furnished portion of the installation materials were also ordered and received. In April 1980, work was rescheduled on the field test preparation activity in order to accommodate changes in the HAHP schedule directed at increased efforts in engine performance testing and analysis. Continuity of effort was maintained by completing initial instrumentation rack component installations and by periodic coordination with Philadelphia Electric (PECO) on submetering results and program schedule revisions. The following paragraphs describe the work accomplished during this reporting period as outlined above.

6.2 SITE PREPARATION AND MODIFICATION

6.2.1 SUMMARY

The outdoor pad and the changes to the air duct system permitting a dual HAHP and conventional HVAC installation were completed during the previous reporting period. In January 1980, gas and electric submeters were connected to a standard utility demand recorder located in the test site basement. Normal furnace operation was monitored throughout the 1980 heating season, with no adverse effects being noted as a result of the duct modification work. Figure 6-1 shows the ducting arrangement and the planned equipment arrangement.

6.2.2 ADDITIONAL MODIFICATIONS

No other major modifications are anticipated since the plans for the installation of the HAHP provide for piping and cable access from the outdoor unit to be accomplished via a basement window which will be temporarily replaced by a fitted panel with suitable openings. At the time of HAHP installation it will, however, be necessary to make water supply, drain and electric

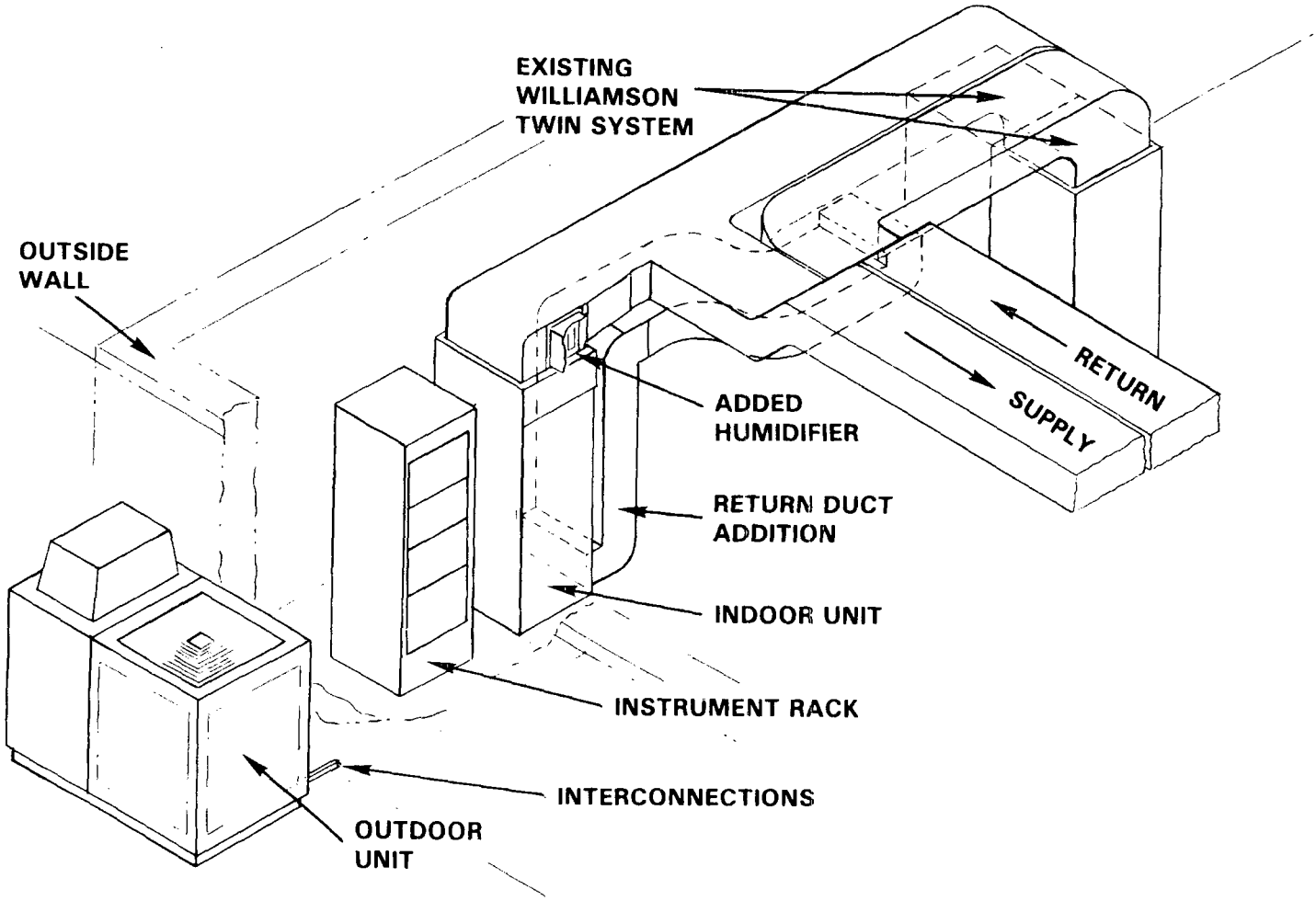


Figure 6-1. Test Site Air Duct and Equipment Arrangement

power connections to the HAHP system. The gas supply pipe will also be extended to the outdoor unit. These facilities will all be provided by simple connections made within the test site basement.

6.3 INSTALLATION PLANNING AND DESIGN

6.3.1 SUMMARY

The principal activity during this reporting period consisted of review of the site layout in more detail in conjunction with the selection of installation material for the HAHP. PECO will provide common HVAC installation materials and supplies while GE will furnish the HAHP system-peculiar material. The latter category includes instrumentation cable harnesses, refrigerant

lines, quick-disconnect fittings, multi-conductor control cables, refrigerant and hydronic tubing, insulation, and miscellaneous related hardware.

6.3.2 CONTROL SYSTEM

A thermostat position - logging circuit was devised using a Hall-effect transistor. This circuit was incorporated in a standard GE heating/cooling thermostat, tested, and found suitable for recording of changes in the residence thermostat setting. This circuit will not eliminate manual recording of the thermostat position, but will provide a positive correlation with any other system changes that are recorded by the data logger. Figure 6-2 shows the output characteristic of this circuit which is essentially a straight line over the normal operating range from 55°F - 80°F. The modified thermostat control will be installed in the test residence living quarters adjacent to the existing HVAC thermostat, providing independent control of the HAHP when it is in operation.

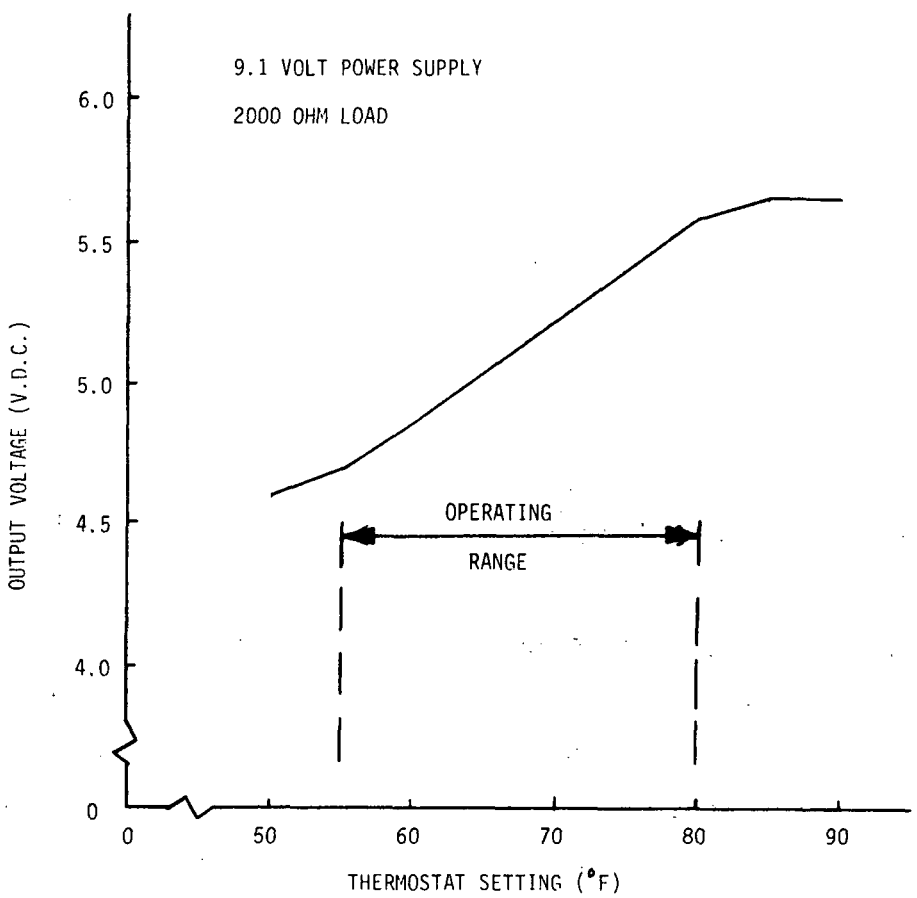


Figure 6-2. Thermostat Logging Circuit Response Curve

6.3.3 CABLE ROUTING AND EQUIPMENT INTERCONNECTION

During this reporting period, installation sketches were prepared for equipment interconnection and cable harness fabrication. All parts and materials available but not yet required have been identified for temporary return to stock. Additional refinement of installation sketches and ordering of the remaining (short lead time) installation material will be accomplished when the field test activity is resumed in accordance with the revised schedule.

6.4 INSTRUMENTATION SYSTEM DESIGN

6.4.1 SUMMARY

The HAHP test site instrumentation design considerations included determination of requirements for sensors, signal conditioning, data logging and data recording. After review of these requirements, major equipment components were selected and interconnections detailed.

6.4.2 SENSORS

Maximum use will be made of thermocouples and motion sensors already built into the HAHP for factory test purposes. Other external sensors to monitor test site ambient conditions and energy consumption are being added. A complete list of sensors and their functional assignments were presented previously in the July-December 1979 Semi-Annual Report. This list will be updated in the future as required.

6.4.3 DATA LOGGER

It was determined that any of several standard data logger units could meet the general requirements for the data to be taken at the HAHP test site. There are significant differences, however, in the ease of programming, type of input, modification flexibility, and availability of user options. A hardware-oriented unit was selected, based on competitive bids, rather than a software-oriented one due to the fixed and fairly long term repetitive measurements anticipated at the test site as opposed to more changeable test requirements of multi-program laboratory use. Table 6-1 lists the major data logger capabilities available in the selected unit, an Esterline-Angus PD 2064.

Table 6-1. Data Logger Capabilities

Option/Assignable Function	Purpose	Channel Capacity
Thermocouple Inputs	System and Ambient Temp. Measurements	
Type J	System and Ambient Temp. Measurements	14
Type K	System and Ambient Temp. Measurements	9
Millivolt Input Signals	Pressures, Flow Rates, T-Stat Setting	12 - 17
Milliamp Input Signals	Elec. Energy Monitoring (Option)	5 - 0
Digital Input Signals	Gas/Elec. Consumption Monitoring	2
Digital Pulse Counting	Gas/Elec. Consumption Monitoring	2
Reference/Calibration	Channel Calibration	3
Spare Channels	Supplemental Measurements	3
		50
On-Board Printer	Operational Monitoring	N/A
9-Track Mag. Tape Interface	Detail Data Accumulation	N/A
Dual Rate Output	Separate Printer/Mag. Tape Rates	N/A
Individual Channel Programming	Flexibility of Programming	N/A
Program Printout	Program Verification and Retention	N/A
Engineering Notation	Direct Readout of Key Values	N/A
Dual Set Point Alarms	High/Low Limit Flags	N/A
Remote Scan Trigger	Higher Scan Rate During HAHP Equipment Operation	N/A
Memory Save	Auto Restart After Short-Term Power Outages	N/A
Soft Halt	Reduces Reprogramming	N/A

6.4.4 MAGNETIC TAPE RECORDER

A nine track digital magnetic tape recorder, Kennedy Model 1600/360 was selected for maximum compatibility with the computerized data processing equipment available at the GE-Valley Forge facility which will perform the computational analysis of field test performance data. This recorder unit will interface directly with the Esterline data logger. The only option required with this recorder was one for automatic re-start after a power interruption. This option makes the recorder and data logger operationally compatible.

6.4.5 SIGNAL CONDITIONING

Daytronics signal conditioning modules were selected to provide proper translation of pressure transducer, fluid flow meter and motion transducer signals and signal levels for compatibility with data logger input ranges. The modules are housed in a module receiver in the instrumentation rack. Appropriate intra cabinet and external cabling will be provided for interfacing with the sensors located in the HAHP outdoor and indoor units and the fluid flow meter assembly.

6.4.6 OTHER INSTRUMENTATION COMPONENTS

Humidity Sensor and Signal Conditioning Package

A solid state humidity probe was purchased and will be mounted in the return duct at the test site, ahead of HAHP and conventional HVAC duct interface. The conditioning package contains both the power supply and the calibration circuitry and controls. The probe unit makes use of a thin film capacitive sensor which changes capacitance as a linear function of relative humidity. Both units are made by Weathertronics Inc. Use of this equipment eliminates the need for manual RH readings and provides positive time correlation with temperatures and other related data.

Pressure Transducer Panel

This panel assembly in the instrumentation rack provides a convenient location for mounting the pressure transducers for the refrigeration loop. In addition, oscilloscope terminals are available on the front of the panel to allow observation of the phase relationships of the engine displacer and power piston. This latter data is not recorded but provides an important checkout and maintenance capability for the initial test program.

Control Panel Assembly

This assembly contains electrical and electronic circuitry for control of both the HAHP system as well as portions of the instrumentation. This assembly provides a convenient controls location for the first field evaluation unit configuration, however, its functions will be reduced and relocated in later systems.

6.5 INSTRUMENTATION RACK ASSEMBLY

6.5.1 SUMMARY

Major components for the instrumentation rack were selected and placed on order early in the reporting period, along with pressure transducers, flow meter assembly components and cable connectors. Fabrication and assembly sketch details were prepared and material accumulations commenced. The major components were mounted in the instrument rack following receipt and inspection. This work was completed in late April 1980. The inter-unit wiring was only partially completed due to rescheduling of field test preparation activity.

6.5.2 COMPONENT MOUNTING

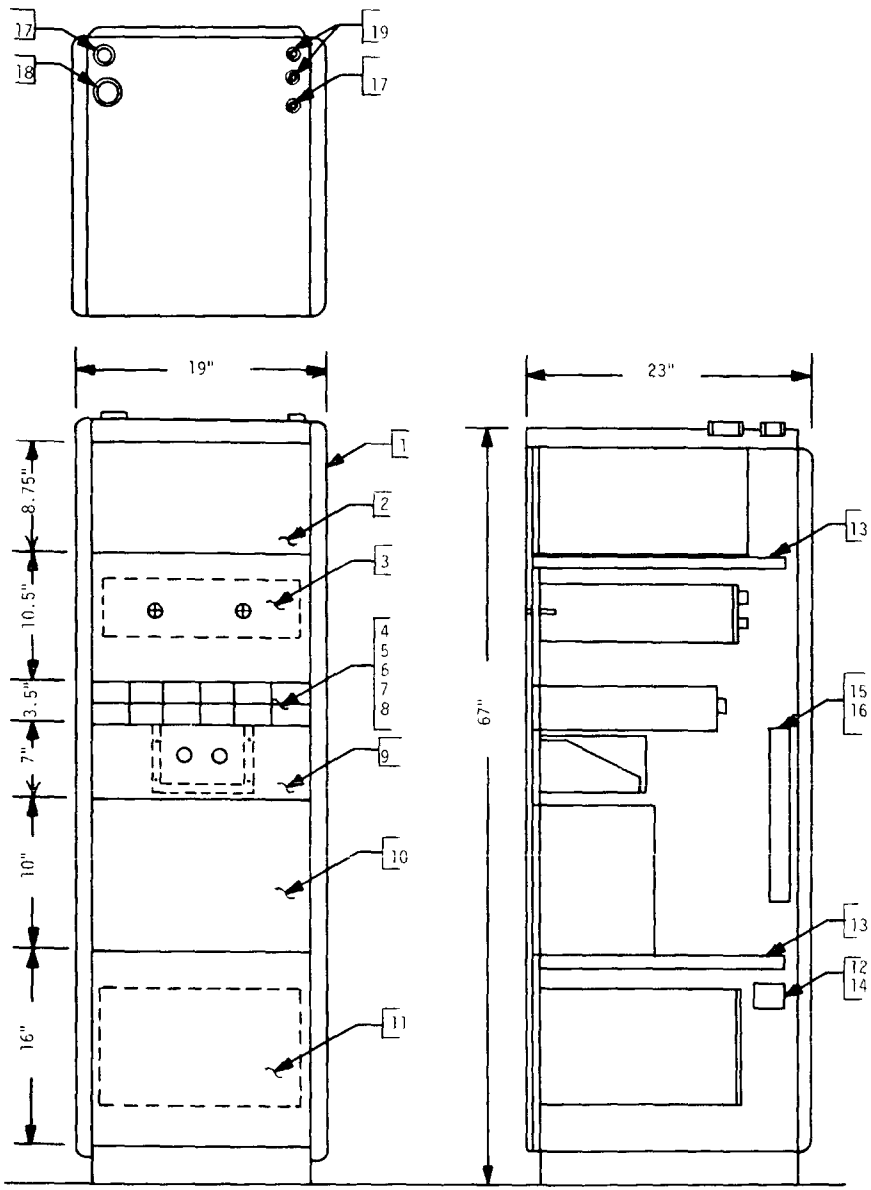
Figure 6-3 shows the instrument rack configuration and identifies the principal components. Figure 6-4 is a photograph of the rack with its major components in place. The mounting brackets and support shelf arrangement are adaptations of previously used GE engineering test designs which have proven highly satisfactory.

6.5.3 INTRA CABINET WIRING

All electrical connections between the major instrumentation rack components will be made within the rack enclosure either by multi-pin connectors or terminal board connections. Placement of components was selected to minimize intra cabinet wire run lengths and bulk.

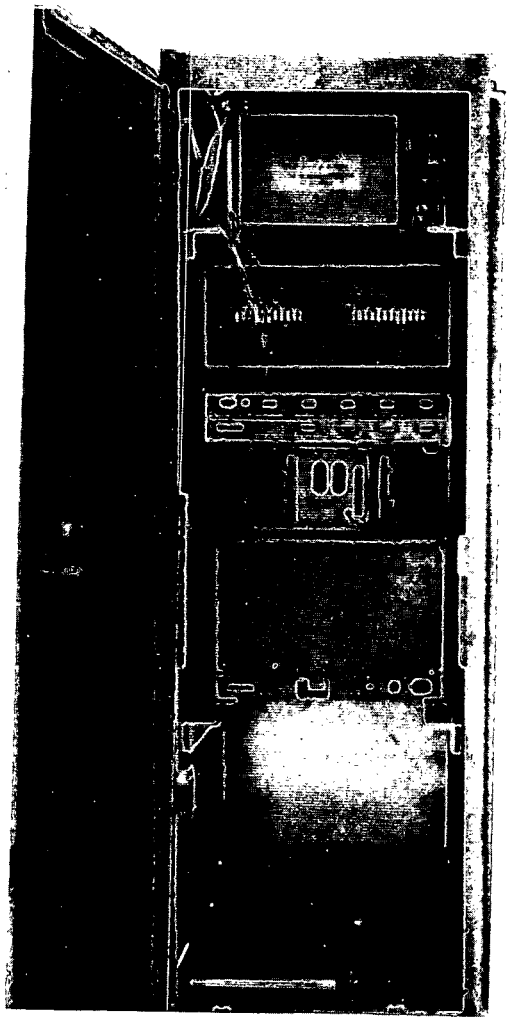
6.6 SUBMETERING TEST RESULTS

Recorded 1980 heating season energy consumption at the test site was analyzed using a standard PECO computerized analysis program. Figure 6-5 shows a prepresentative twenty-four hour cycle during the test period. The correlation between gas demand and periods of high and low outside ambient temperatures as well as sun loading can be clearly seen. The wind velocity effect is less clear and is not segregated from the other load factors. The weather data inputs reflect conditions at a nearby PECO monitoring station which are believed to be representative of the surrounding residential neighborhood. Additional on-site temperature correlation will be provided when the HAHP installation is completed. Figure 6-6 shows the test site peak demand sample data overlaid on the projected heating/cooling capacity curve for the HAHP. For

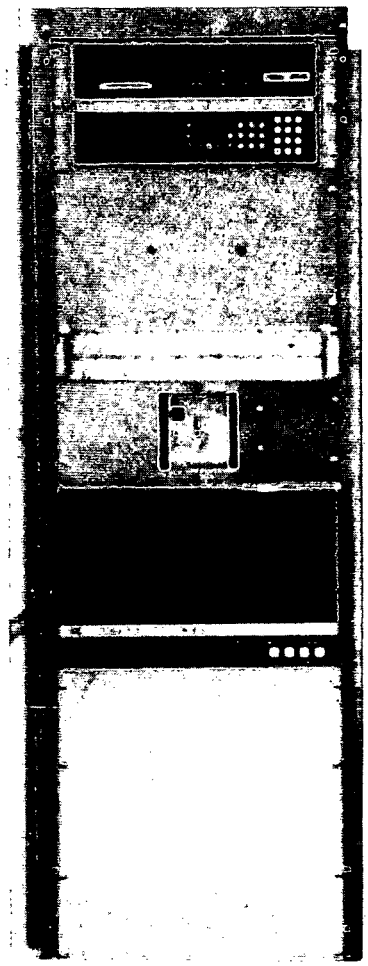


PART NO.	NAME	IDENTIFICATION	QTY.
1	Rack, Cabinet	BUD 91F087	1
2	Data Logger	Esterline PD2064	1
3	Panel Assem., Press. Trans.	GE SK-FT-005	1
4	Module Receiver	Daytronic 9010 RM	1
5	Str. Gauge Mod. (S-29)	Daytronic 9170X1	1
6	Str. Gauge Mod. (S-13, 14, 20, 21)	Daytronic 9170	4
7	Freq.-Volt. Mod. (S-10, 10a)	Daytronic 9140	2
8	LVDT Mod.	Daytronic 9130	2
9	Panel Assem., Humid. Meas.	GE SK-FT-006	1
10	Dig. Mag. Tape Recorder	Kennedy 1600/360	1
11	Control Panel Assem.	GE	1
12	Junction Box	GESCO 54151-L	1
13	Support	GE SK-FT-007,008	4
14	Bushing, Insulated	GESCO BBL50	3
15	Wiring Strip	SPC TECH. 60N2444	1
16	Wiring Strip	SPC TECH. 60N2401	1
17	Bushing, Insulated	GESCO BBL150	2
18	Bushing, Insulated	GESCO BBL200	1
19	Bushing, Insulated	GESCO BBL75	2

Figure 6-3. Instrumentation and Control Rack Assembly



Rear View



Front View

Figure 6-4. Assembled Instrumentation Rack

the 1980 season, the sample data indicates a capability of the HAHP to meet the test site demand load down to approximately $0^{o}F$, which is below the normal low temperature design point for the area.

6.7 PLANNED ACTIVITY FOR NEXT PERIOD

Field test preparation activity has been scheduled to restart late in the next reporting period. However, in the interim, appropriate coordination will be maintained with PECO and the test site owner. Cooling season data from the conventional air conditioner at the test site will be evaluated when available.

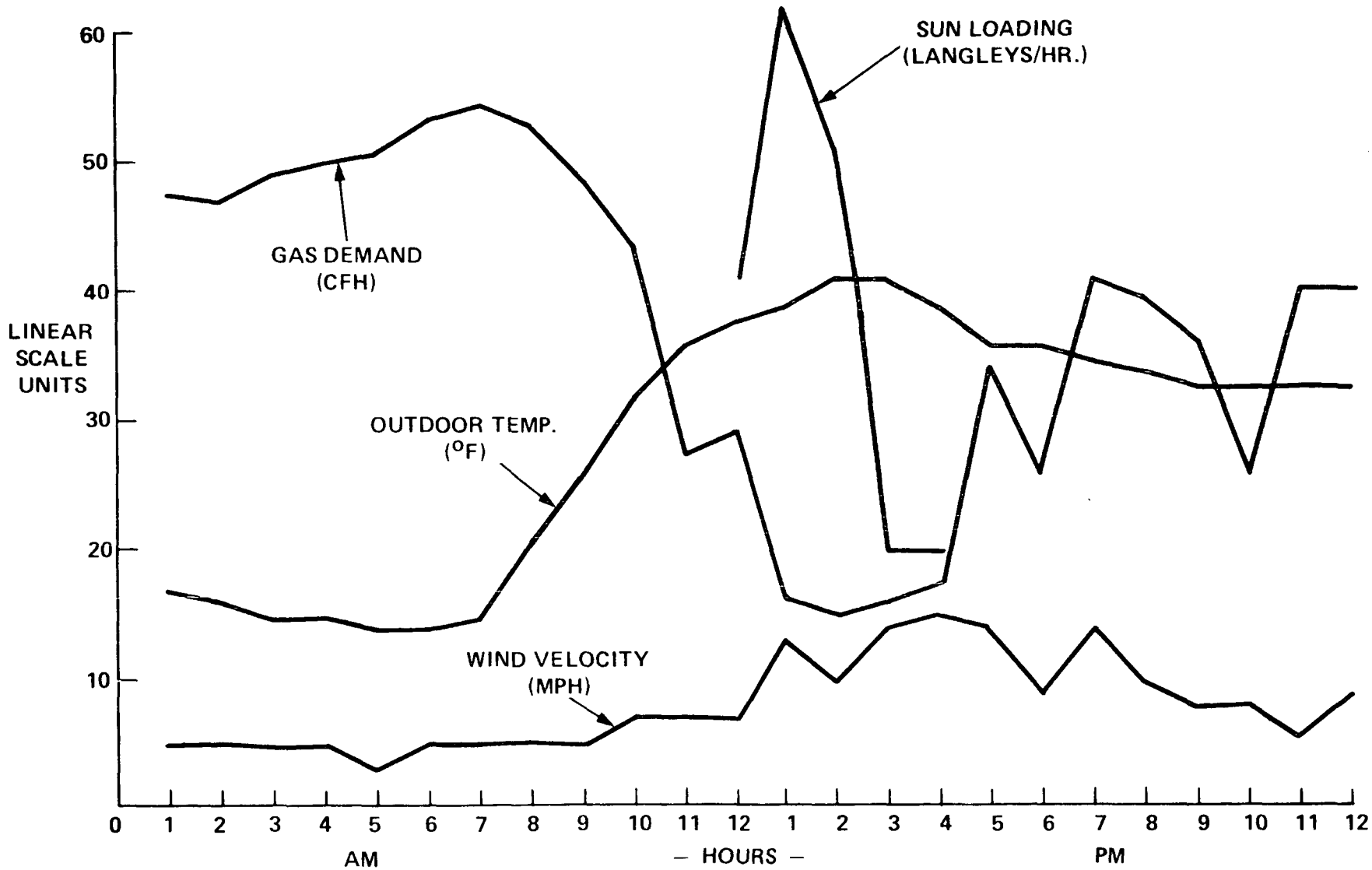


Figure 6-5. PECO Test Site Sample Data
(February 11, 1980)

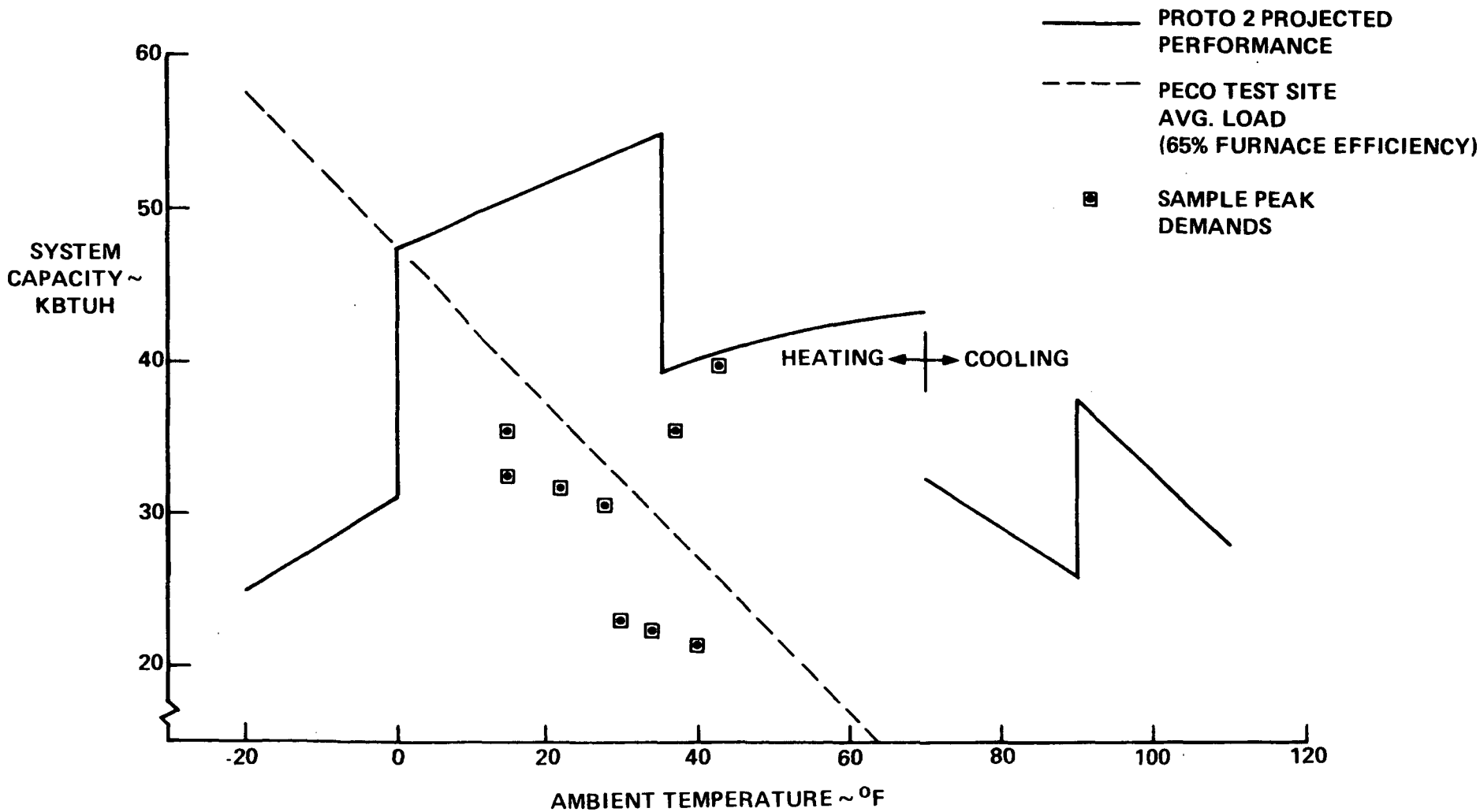


Figure 6-6. Proto 2 Capacity Match to Test Site Load

APPENDIX A

TEST PLAN SUMMARY FOR MATERIAL DEVELOPMENT - BEARING/SEALS

APPENDIX A

TEST PLAN SUMMARY FOR MATERIAL DEVELOPMENT - BEARING/SEALS

A.1 INTRODUCTION

This test plan describes the functional and evaluation tests to be conducted on a number of candidate materials for bearing/seals applications in the Stirling Engine of the HAHP Program. The tests are to be conducted in a Wear Test fixture where engine conditions of temperature, pressures, stroke, frequency and helium environment will be maintained. The previous semi-annual report (July through December, 1979) contains a description of the Wear Test fixture. The tests are to be conducted according to the test plan to minimize the effect of extraneous conditions on the test results. Very exacting measurements are required throughout the test series and shall be made using the same techniques and procedures in order to enhance consistency and to reduce operator error.

A.2 MATERIAL AND EQUIPMENT REQUIRED

1. Wear Test fixture assembly - GE Dwg. 919E291
2. Drive mechanism - GE Dwg. 848E901
3. Instrumentation
   a. Model 500 Electrolytic Hygrometer
   b. Figure A-1 (Carr, SK-WT-002) set-up
   c. Figure A-3 (Carr, SK-WT-004) readout
4. 0-1 0.0001/div indicating micrometer with ball tip

   1-2 0.0002/div indicating micrometer
5. Flow bench - EAC1 Environmental Air Control Assembly in Room 23B35
6. Gram scale - Mettler #G37944 and Mettler #G37901
7. Rod/sleeve test parts - Table A-3
8. Rings/liner test parts - Table A-4

9. Gas flow meter - Cat. No. 11-164050

10. Electronic timer-Textronic DC 503

11. "Free Fall" test fixture assembly, GE Dwg. 147D9959

12. Two channel oscilloscope (intermittent use)

13. G.R. Strobo-tach (initial set-up only)

A.3 SAFETY REQUIREMENTS

No smoking shall be permitted in the immediate area while Genesolv D is being used to clean test parts. Bag all specimens to maintain cleanliness.

A.4 INSTRUMENTATION

The instrumentation used to monitor the test parameters are tabulated in Tables A-1 and A-2 with locations noted in the tables.

Table A-1. Set-up Instrumentation

Readout/Units	Identification/Location
PPM of H_2O	Moisture content of He supply
In. Vacuum	Evacuation vacuum, Item 3, Figure A-1
SCIM	Nitrogen flow thru rod/sleeve clearance, Data Sheet 10.6
Microsec	"Free Fall" time of piston and rings in liner, Data Sheet 10.7

Table A-2. Operational Instrumentation

Readout/Units	Identification/Location Shown in Figure A-2
P_1 - psig	Regulated helium supply pressure
P_2 - psig	Top part of test cylinder assembly
P_3 - psig	Bottom part of test cylinder assembly
T_1 - ^oF	Set point temperature - mid point of cylinder wall
T_2 - ^oF	Temperature - mid point of seal housing

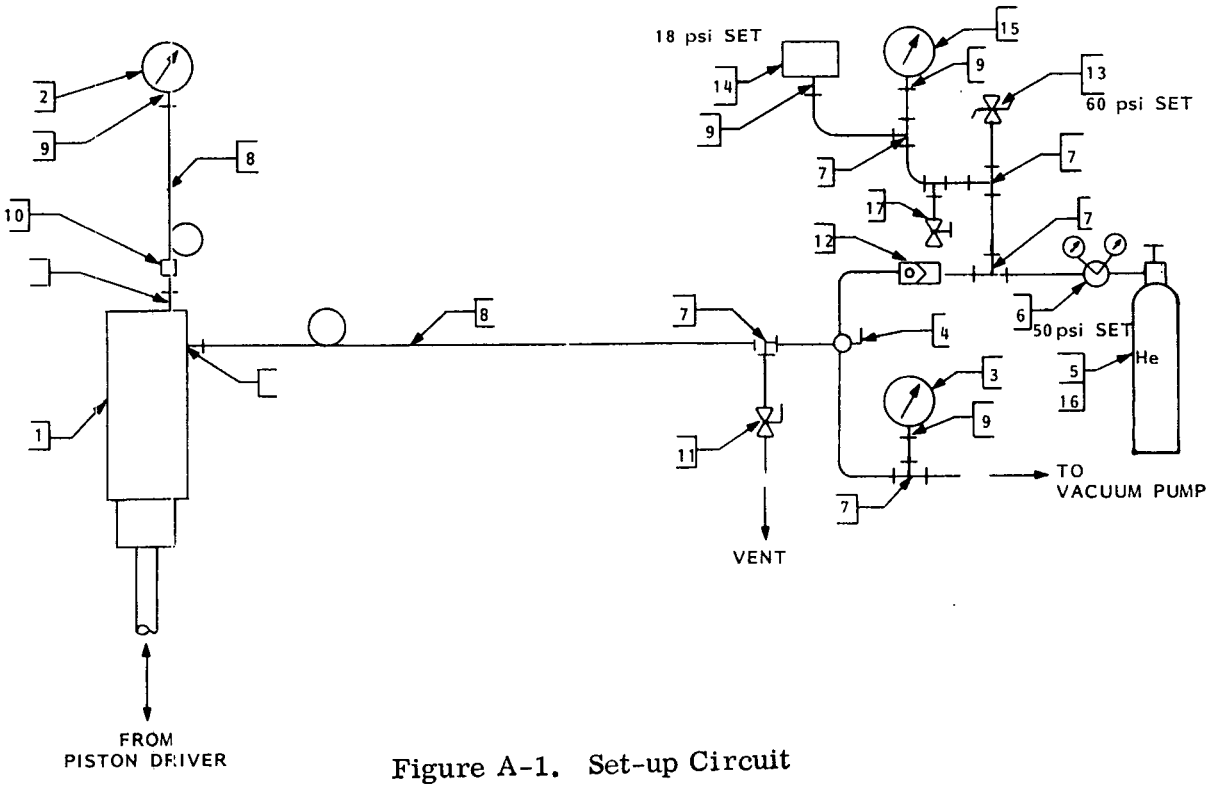


Figure A-1. Set-up Circuit

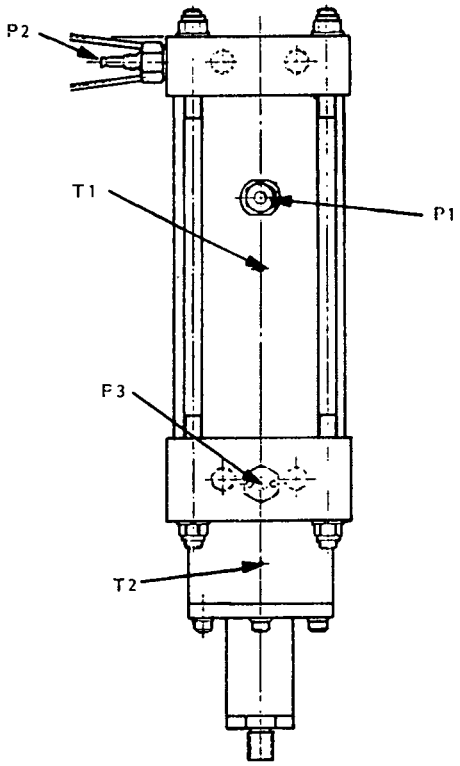


Figure A-2. Instrumentation on Cylinder Assembly

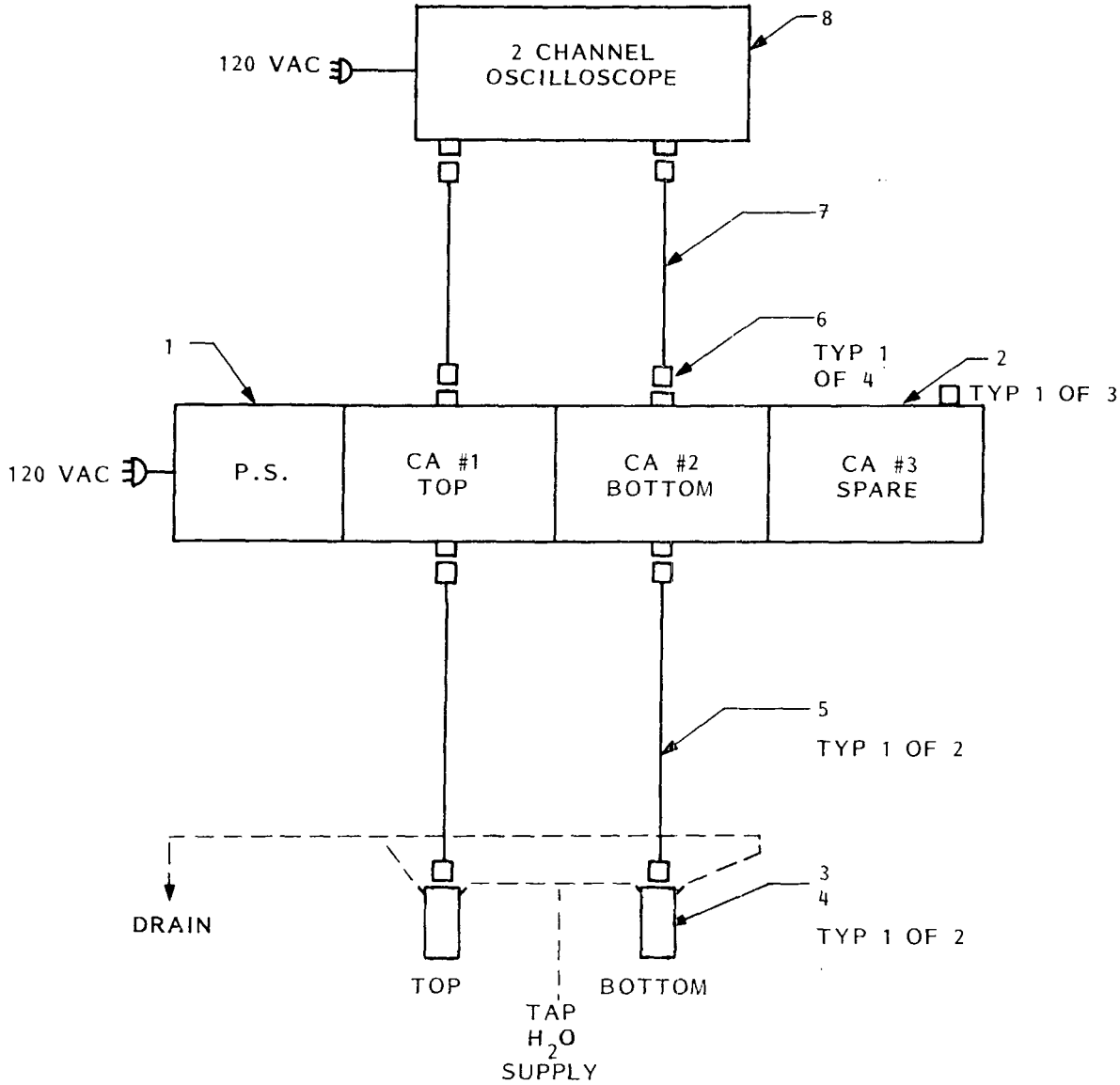


Figure A-3. Pressure Readout Circuit

A.5 INSTRUMENTATION ACCURACY

The test facility shall provide the following measurement accuracies:

A.5.1 TEMPERATURE MEASUREMENTS

Measurement Type	Required Accuracy
Thermocouple	$\pm 2.2^{o}C/4^{o}F$
Whiteback, indoor	$\pm 0.5^{o}C/1^{o}F$

A.5.2 PRESSURE MEASUREMENTS

Measurement Type	Required Accuracy
Gauge	± 1/2% F.S.
Piezoelectric transducer	± 10 psi

A.5.3 FLOW RATES

Fluid Measured	Required Accuracy
He	± 2%
N_2	± 2%

A.5.4 HEATING SET POINT TEMPERATURES

The temperatures for the heating set point shall be controlled to within ± 0.5°C/1°F.

A.6 TEST PREPARATIONS - PRE-TEST

A.6.1 TEST PART MEASUREMENTS

Measure and record all dimensional characteristics specified on Data Sheets 10.1, 10.2 and 10.5 and the weights specified on Data Sheets 10.3, 10.4. All weights are to be measured on the Mettler #G37944 scale, except rings, which are to be weighed on Mettler Milligram scale #G37901.

A.6.2 PART FIT AT OPERATING TEMPERATURE

Match up all test sets shown on Tables A-3 and A-4 and position parts in their normal operating relationship. Place matched sets in oven where temperature can be controlled and make provision to maintain nitrogen atmosphere around parts while they are at elevated temperature.

Heat part sets to 275°F in 50°F increments and check for freedom of fit after temperature stabilizes at each increment. Note fit on Data Sheets 10.3 and 10.5.

Table A-3. Rod/Sleeve Tests

Test Sequence	Rod / Sleeve	GE Dwg. 147D9951 G#	Comments
1	6061-T6 + TUFRAM + MoS_2 / 4340 + Cr + MoS_2	1	Control, MoS_2 in grease form applied by lab, remove excess.
2	6061-T6 + TUFRAM + MoS_2 / 440C + MoS_2	2	MoS_2 in grease form applied by lab, remove excess.
3	6061-T6 + Rulon LD / 440C	4	
4	440C + Rulon LD / 440C + Xylan	7	
5	440C + Rulon LD / Nitronic -60	8	
6	440C + Rulon J / Nitronic -60	10	
7	6061-T6 + Koplon / 4340 + Cr	3	
8	440C + Rulon LD + MoS_2 / CI + MoS_2	9	MoS_2 in grease form applied by lab, remove excess.

Table A-4. Rings/Liner Tests

Test Sequence	Seal & Rider / Liner		Comments
	Materials	Parts Numbers (GE Dwgs)	
1	K-30C (2) / 440C + Cr	295A8326P1&225B1238P3 / 225B1392 P2	Control
2	K-30C (2) / 440C	295CA8326P1&225B1238P3 / 225B1392 P1	
3	Envex 1228 (2) / 440C + MoS_2	295A8325P2&225B1239P5 / 225B1401 P2	
4	K30C (2) / 440C + Xylan	259A8326P1&225B1238P3 / 225B1401 P1	
5	K-30C (2) / Nitronic-60	295A8326P1&225B1238P3 / 225B1393 P1	
6	Envex 1115 & K-30C / #1 above	295A8325P1&225B1238P3 / See comments	This test reuses #1 liner after grit blast + MoS_2 grease application
7	Vespel 211 & K-30C / #3 or 4 above	295A8325P4&225B1238P3 / See comments	* Remove deposited ring material from previously used #3 or 4 liner
8	Rulon LD & K-30C / #2 above	295A8325P5&225B1238P3 / See comments	* Remove deposited ring material from previously used #2 liner
**9	K-30C (2) / #8 above	295C8326P1&225B1238P3 / See comments	Resize liner #8 I.D. & Coat with baked on MoS_2

* Burnish or otherwise remove high spots without abrasive.

** Optional to be determined later.

A.6.3 CYLINDER ASSEMBLY - ROD/SLEEVE TESTS

This test uses all parts shown on GE Dwg. 147D9950 except Piston Assembly, GE Dwg. 225B1389G1 is to be replaced by Sleeve Plug, GE Dwg. 225B1459P1.

Select appropriate rod/sleeve set from Table A-3.

Clean all parts that go into the cylinder assembly in Genesolv D and lay them out on a clean sheet of paper in the EAC1 Environmental Air Control Chamber located in Room 23B35.

Apply MoS_2 grease to parts as identified in Table A-3.

Position sleeve in Seal Housing, GE Dwg. 18C3002P1, matching punch mark on flange of sleeve with mark on seal housing. Build up balance of cylinder assembly, but replace the He seals, spacers, O-rings, etc. with Leak Test Spacer, GE Dwg. 295A8549P1.

Position assembly on work table so that test rod points upward. Conduct leak rate measurement according to Data Sheet 10.6 and record specific values.

Remove leak test spacer and complete the assembly installing He seals, O-rings, spacers, etc. and Upper End Cap, GE Dwg. 225B1457P1, of rod shield.

Complete wear test fixture assembly per GE Dwg. 919E291.

Move assembly to test stand, apply control thermocouple to mid-point of Shell Assembly, GE Dwg. 188C3003G1, and another T/C to mid-point of Housing Seal, GE Dwg. 188C3002P1. Install Hi-Flex heaters, clamp and insulate.

Mount assembly into test stand and attach helium, heater power and temperature readout. Plug all transducer ports until ring/liner test series starts at a later date. Do not attach drive at clevis joint yet.

Switch 3-way valve shown on Figure A-1 to evacuate position and evacuate to 28 inch HG, at same time purge helium from supply to 3-way valve for 5 minutes.

A.6.4 CYLINDER ASSEMBLY - RING/LINER TEST

This test uses all parts shown on GE Dwg. 147D9950, except for the Sleeve, GE Dwg. 147D9951. Use Rod, GE Dwg. 147D9951G5 or G6 for entire test series.

Select appropriate rings/liner set from Table A-4.

Clean all parts that go into the cylinder assembly in Genesolv D and lay them out on a clean sheet of paper in the EAC1 Environmental Air Control Chamber located in Room 23B35.

Attach piston extension shaft to Piston Assembly, GE Dwg. 225B1389G1, in place of Piston Cap, GE Dwg. 225B1386G1, and conduct "Free Fall" test on the set of rings and liner as required on Data Sheet 10.7.

Attach Piston Cap, GE Dwg. 225B1386G1, to Piston Body, GE Dwg. 184C8299G1, and build cylinder assembly according to GE Dwg. 147D9950 except replace Sleeve, GE Dwg. 147D9951, with an additional Separator, GE Dwg. 295A8474P1 and associated O-ring. Use same exposed rod protector as employed in rod/sleeve tests, i.e., GE Dwg. 919E291P22, 23, 24 and 25.

Move assembly to test stand and install heaters and thermocouples as in the rod/sleeve tests.

Mount assembly into test stand and connect all instrumentation and service. Do not attach drive at clevis joint yet.

A.6.5 GENERAL PREPARATIONS

Sample each new bottle of helium using the Model 500 Electrolytic Hygrometer, and record moisture level. If moisture exceeds 10 ppm, notify the responsible engineer before continuing.

Evacuate thru helium supply line and charge with He. Set regulator to deliver 10 psig for rod/sleeve tests and 50 psig for the rings/liner tests. (The 50 psig value may require adjustment either up or down to produce the desired 125 psi P-P pressure during operation.)

A.7 CHECK-OUT PROCEDURE

Energize heaters and set temperature controller to 250°F, (mid point of large heater).

Observe that the test parts do not experience any binding when moved by hand thru the normal path of motion. Notify the responsible engineer if binding does occur.

A.8 TESTING

A.8.1 ROD/SLEEVE OPERATING CONDITIONS

Operating conditions for rod/sleeve tests are:

T_1 = 250°F

P_1 = 10 psig

A.8.2 RING/LINER OPERATING CONDITIONS

Operating conditions for rings/liner tests are:

T_1 = 250°F

P_1 = 50 psig (may require adjustment up or down to achieve desired value of P_2-P_3)

$*P_2 - *P_3$ = 125 psi (P - P)

* Peak to Peak Values

A.8.3 TEST SERIES

Conduct screening tests of 72 hours ± 4 on each of the material candidate sets in the order listed in Tables A-3 and A-4.

Conduct extended tests of 144 hours ± 8 on each of the material candidate sets selected by the development materials team from the screening test results. Conduct rod/sleeve tests separate from rings/liner tests as in screening tests.

Conduct combined extended test of 288 hours ± 8 on the full set of materials singled out through previous test results.

A.9 TESTING PREPARATION - POST TEST

Remove cylinder assembly from test stand and disconnect drive linkage at clevis joint. Clean all exterior surfaces with Genesolv D but make sure lubricant is not washed out of coupler. Also solvent is to be kept out of the cylinder assembly.

Take complete assembly to clean area identified in Section A.6.1 for disassembly.

A.9.1 ROD/SLEEVE SETS

Carefully observe alignment between the position marks on the rod and seal housing. If it changed from its initial set position, note new position on Data Sheet 10.5.

Remove the helium seals, spacers, etc. from the seal housing and replace with Leak Test Spacer, GE Dwg. 295A8549P1. (The complete removal of sleeve and helium rod seals may be accomplished if Sleeve Plug, GE Dwg. 225B145P1, is used in place of Piston Cap, GE Dwg. 225B1386P1.) This is accomplished by removing the Retainer Plate, GE Dwg. 225B1397P1 and pulling rod with other attached parts out of bottom of Seal Housing, GE Dwg. 188C3002P1.

Position assembly on work table so that rod points upward. Conduct leak rate measurement according to requirements of Data Sheet 10.6 and record values requested.

Remove rod and sleeve from cylinder assembly and note relationship of position marks on sleeve flange and seal housing on Data Sheet 10.4.

Remove all wear products, i.e. dust from rod and sleeve.

Weigh sleeve and rod and record values on Data Sheets 10.4 and 10.5 respectively. Bag and store for future I.D. and O.D. measurement and reuse (rod O.D. measurement will be made by engineer, sleeve I.D. measurement will be made by Incoming Inspection in Room M8408.)

Continue with next test set per Table A-3.

A.9.2 RINGS/LINER SETS

Remove Rods, GE Dwg. 295A8482P1 (4) from cylinder assembly and carefully remove Cylinder Shell, GE Dwg. 88C3003G1 and Liner, GE Dwg. 147D9950P12.

Remove Piston Cap, GE Dwg. 225B1386G1 from piston assembly and replace with Piston Cap/ Bolt Assembly shown on GE Dwg. 147D9959. Weigh this assembly and record on Data Sheet 10.7.

Place piston assembly inside the liner, center on "Free Fall" holding fixture and conduct "Free Fall" time measurements specified on Data Sheet 10.7.

Remove all rings from piston assembly, clean off all dust and wear particles, Weigh on Mettler Milligram G #37901 Scale. Also measure and record dimensions specified on Data Sheets 10.1 and 10.2. Bag and identify for future use.

Clean liner in Genesolv D, weigh and record value on Data Sheet 10.3. Bag and identify for future I.D. measurement by Incoming Inspection, Room M8409 and future tests.

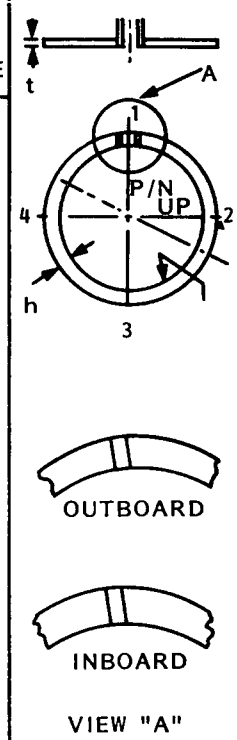
Continue with next test set per Table A-4.

A.10 TEST RECORDS AND EVENT LOG

The Data Sheet format is provided in this section for the Test Plan. The Data Sheets 10.1 to 10.7 shall be used as called for in Sections A.6.1, A.6.2, A.9.1 and A.9.2. Also, an event log is to be maintained in the Laboratory Log Book provided. All major events such as test start, completions of a test sequence, malfunction and all test interruptions shall be recorded.

Data Sheet No. 10.1, Figure A-4, shows a typical data sheet format to be used throughout this test program.

ITEM	DATA SHEET 10.0																			LAB TEMP	TIME
	OUTBOARD RING SEAL									INBOARD RING SEAL											
	WT	t_1	t_2	t_3	t_4	n_1	n_2	n_3	n_4	WT	t_1	t_2	t_3	t_4	n_1	n_2	n_3	n_4			
	TYPICAL DATA SHEET																				
1																					



1 WITH EACH SET OF MEASUREMENTS, RECORD DATE, ROOM TEMP & INSPECTOR'S INITIALS. PARTS TO REMAIN AT ROOM TEMP 4 HRS BEFORE MEASUREMENTS ARE MADE.

Figure A-4. Typical Data Sheet Format

GENERAL ELECTRIC

Space Systems Division

Headquarters: Valley Forge, PA ☐ Evendale, OH ☐ Huntsville, AL
☐ Houston, TX ☐ Sunnyvale, CA ☐ Beltsville, MD ☐ Lanham, MD
☐ Tacoma, WA ☐ Palmdale, CA ☐ Bedford, MA
☐ Washington, DC Area