GE DOCUMENT NO. 77SDS4254
SEPTEMBER 1977

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

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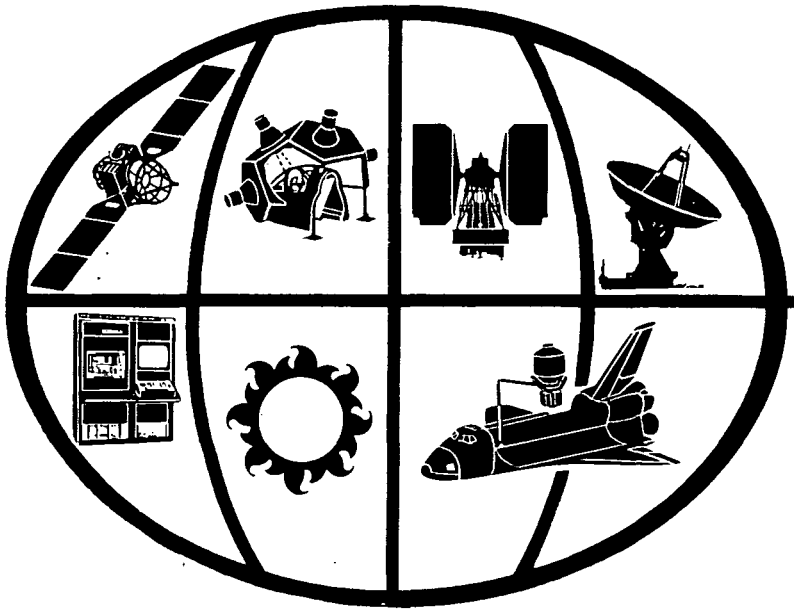
SEMI-ANNUAL REPORT
NO. COO/2911-1

JANUARY THROUGH JUNE 1977



space division

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PREPARED FOR:

THE AMERICAN GAS ASSOCIATION
AND
THE ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
CONSUMER PRODUCTS AND TECHNOLOGY BRANCH

BY

GENERAL ELECTRIC

ADVANCED ENERGY PROGRAMS

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 AGA on a multi-phase program for the development of a Gas Heat Pump (GHP). The Energy Research and Development Administration (ERDA) joined General Electric and the American Gas Association in the joint program in 1976.

Phase I of the program was completed in December, 1975, and Phase II was initiated in February, 1976. This report documents the work accomplished during the first 6 months of the Phase II program.

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 determine, through an analytical effort, if a gas heat pump product could evolve into a viable business venture. An evaluation 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 system 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

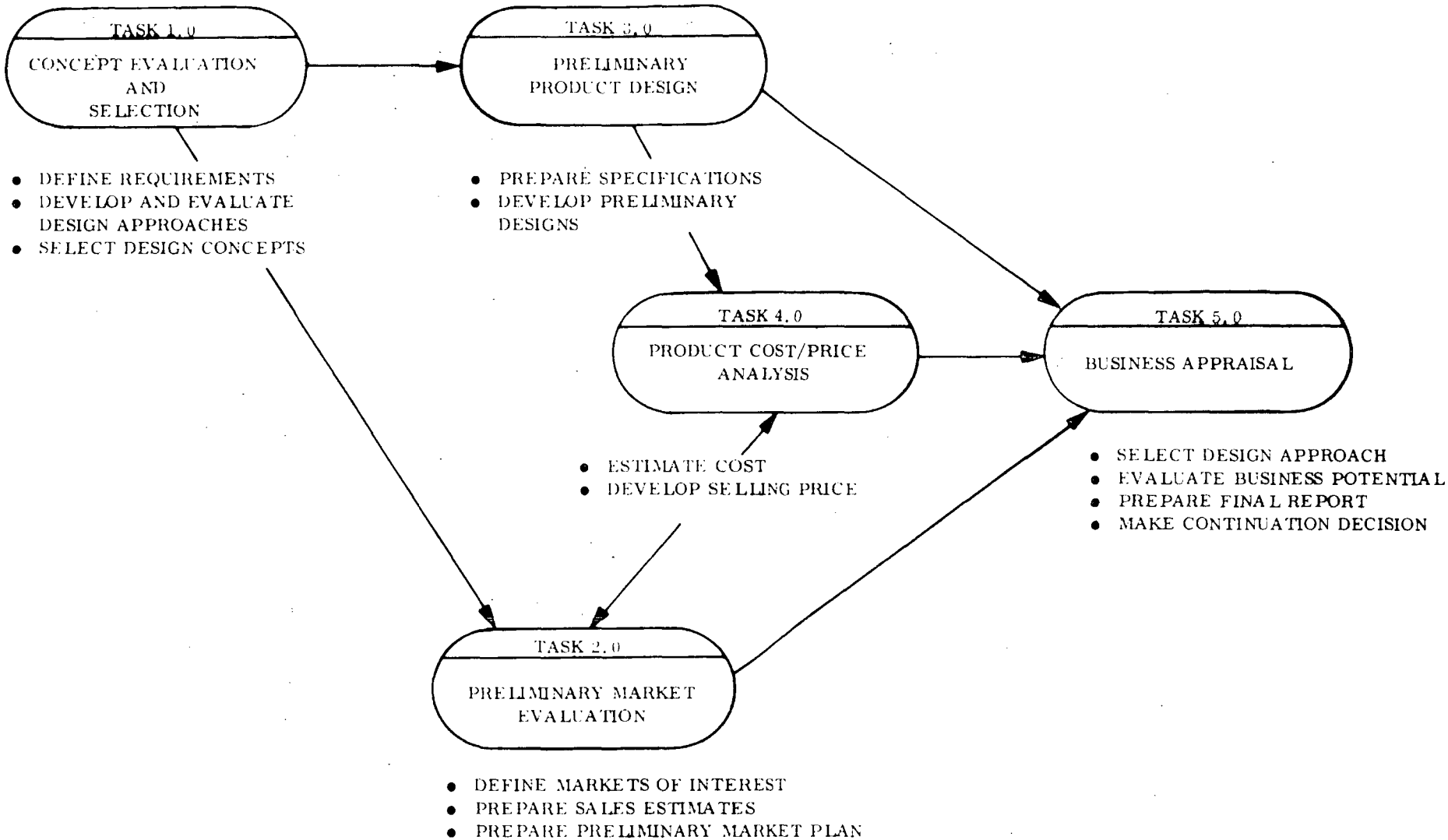


Figure 1-1. Phase I Flow Diagram

representatives. As a result of Phase I, 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.

Two generations of prototype hardware will be developed during the Phase II program. Prototype 1 will be the first attempt to test, on a system level, the Stirling/Rankine GHP with heat pump configured hardware. This system will be comprised of hardware which is primarily developmental in nature. The Prototype 2 system will consist of more mature hardware

1.1.3 PHASE II PRIOR ACCOMPLISHMENTS

The Prototype 1 GHP will be a 3-ton, split system which will provide approximately 64,000 Btu/hr in the heating mode at the 45°F ARI rating temperature. Other system and component level requirements and goals were specified in the system design effort - Task 1.0. The system performance goal is to obtain coefficients of performance (COP) of 1.6 and 0.85 for the heating and cooling modes, respectively, excluding the system electrical power requirements; the electrical power required for the fans, blowers, and pumps shall be less than 1500 watts. Based on these heat pump performance levels, the following subsystem performance goals have been established:

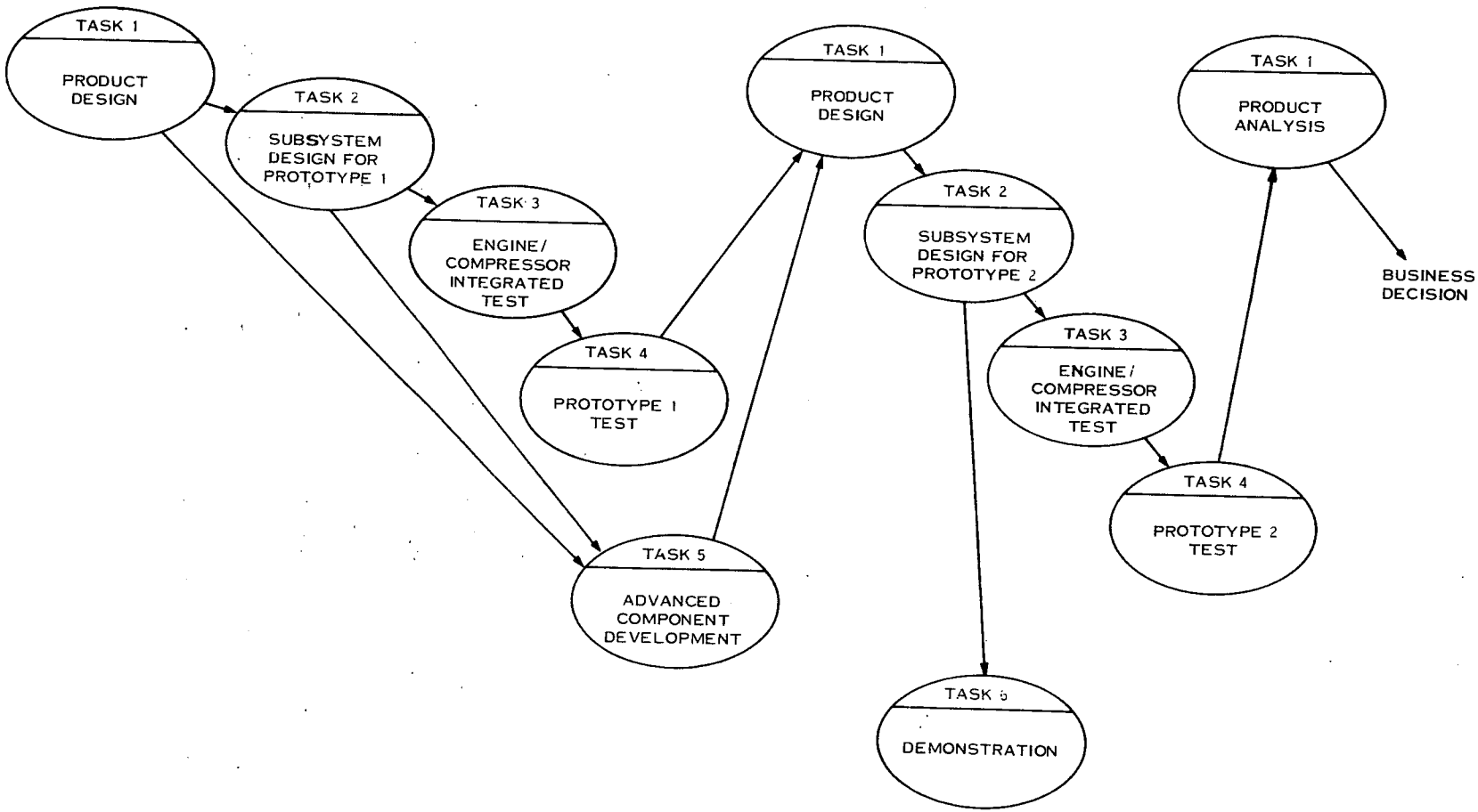


Figure 1-2. Phase II Flow Diagram

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

The component level accomplishments performed prior to this reporting period are discussed in the following sections.

1.1.3.1 Combustor

The development of the low pressure drop, high energy density radiation cooled transpiration burner was completed. The bench testing of the combustor enabled the burner to be characterized in terms of its design parameters - air preheat, temperature, percentage of excess air, porosity, thickness, and base material thermal conductivity of the flame holders; also, it enabled the design for the Protype 1 combustor to be selected. The cylindrically shaped flame holder employing a metal weave fabricated from Hoskins 875 alloy was tested over a wide range of parameters:

- A turn down ratio of 3 to 1 (60,000 Btuh firing rate to 20,000 Btuh)
- Excess air of 0% to 30%
- Pre-heat of $0^{o}F$ to $1000^{o}F$

Flashback did not occur, and the flame was uniformly distributed over the burner surface under all of these operating conditions.

1.1.3.2 Engineering Research Compressor

A test program to characterize a double acting inertia compressor was undertaken in order to acquire data to assist in the design of the Prototype 1 compressor. Parts and piston configurations from a standard heat pump compressor were used;

this expidited the Prototype 1 design and allowed evaluation of the approach for prototype implementation.

The compressor was initially tested using air as the working fluid. In this test, the compressor was mounted on a vibration test facility. This testing confirmed the GE design approach for an inertia compressor; the data on air corresponded well with predicted values. Subsequently, the compressor was characterized employing Refrigerant-22 as the working fluid. A variable speed crank-arm mechanism driven by an electric motor provided the excitation stroke, and a 5 ton single package heat pump acted as the test load for the compressor. The compressor was fully instrumented including two linear velocity displacement transducers which made it possible to measure the dynamic pressures as a function not only of time but also as a function of the compressor piston displacement. These measurements facilitated the interpretation of the test data and provided the means to verify analytical prediction techniques. The compressor was performance mapped as a function of pressure ratio, excitation stroke, and frequency; an isentropic efficiency approaching 75% was achieved.

1.1.3.3 Engine/Compressor Assembly

A significant effort was devoted to performing an analytical study to investigate the dynamic interactions between the Stirling engine and the inertia compressor. A design of the three body resonant system evolved which operates stably over the entire heat pump ambient temperature operating range of interest. Sufficient latitude was provided in the engine design parameters - displacer rod diameter, the displaced gas spring volume, the power piston weight, and the charge pressure - so that the engine/compressor performance can be adjusted during the testing phase of the program.

The detail design was completed during the prior Phase II reporting periods and fabrication of selected engine components was initiated.

1.1.3.4 Controls

The controls for Prototype 1 were designed, components were selected and parts were received; assembly of the control box was started. The Prototype 1 design incorporates a time delay relay implementation of the start sequence. Although the implementation may eventually be electronic, the relay approach allows increased test flexibility at this early stage of development. The controls are designed to interface with standard GE-ACBD (Air Conditioning Business Division) control components. The gas control selected was a unit recently developed by Fenwall for GE-ACBD applications. Safety controls incorporated in Prototype 1 include engine temperature, high/low refrigeration pressure, low cooling fluid flow and engine over-stroke protection.

1.1.3.5 Outdoor and Indoor Units

The outdoor unit configuration for Prototype 1 is illustrated in Figure 1-3. The unit is characterized by three compartments housing the various assemblies - the combustor/engine/compressor assembly, the refrigerant and hydronic outdoor coils and blower assembly, and the piping compartment. Significant in this configuation is the vertical orientation of the combustor/engine/compressor assembly in the rear compartment. Prototype 1 was also configured to provide easy access to heat pump components in order to facilitate servicing and component exchange at the expense of overall size. Design of the outdoor unit was completed, all components were fabricated, and assembly of the outdoor package was initiated during 1976. The indoor unit is a conventional GE Air Handler (WE960C) modified to incorporate the waste heat coil. The particular

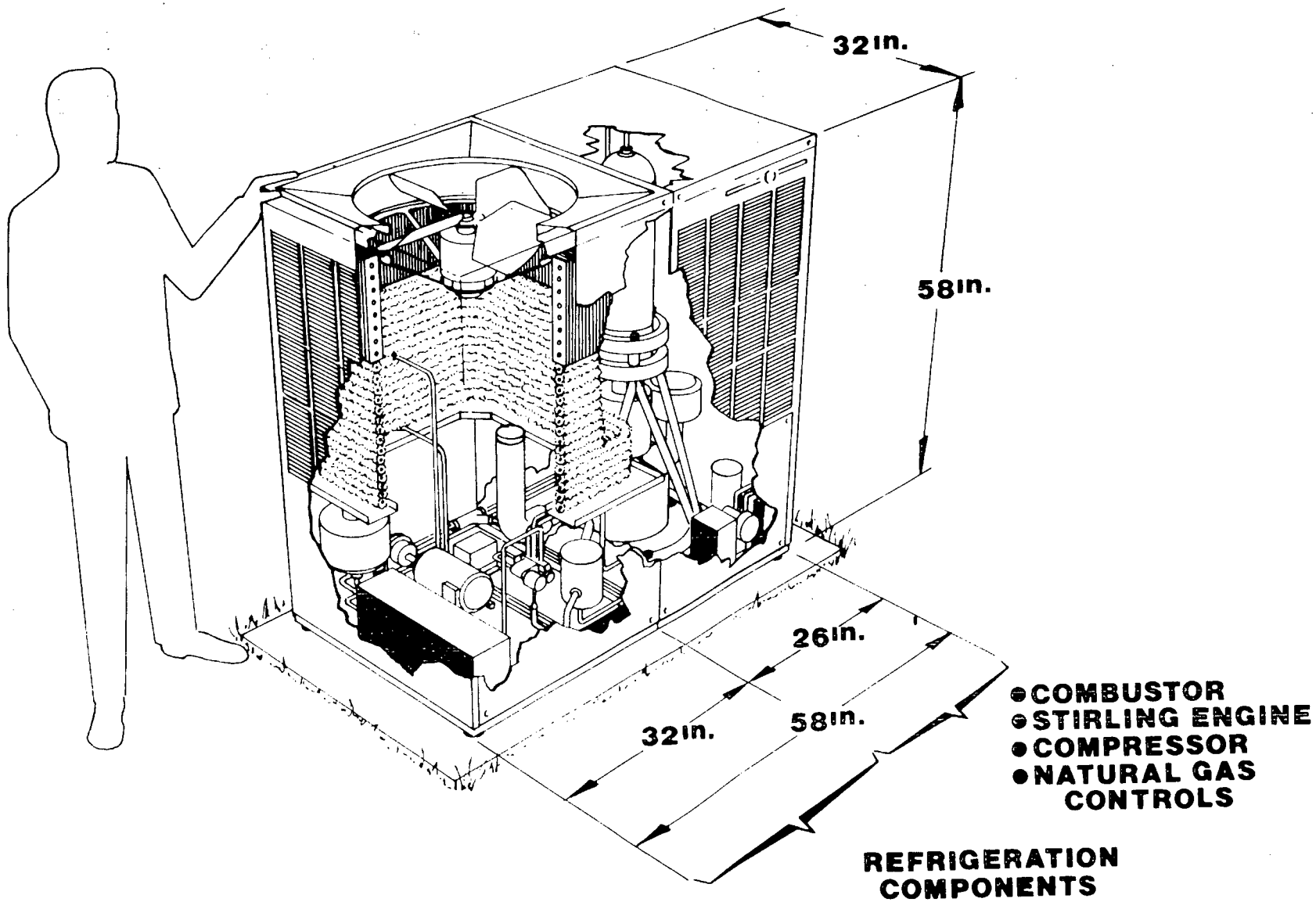


Figure 1-3. Prototype 1 Configuration

model selected has a 5-ton capacity which was modified to permit flexiblity when testing the 3-ton GHP. Drawings for the unit were completed and the unit was procured. Modifications to the evaporator were completed at GE-ACBD in Trenton, New Jersey. All other indoor unit components were received, and assembly of the package was started.

1.2 PROGRESS SUMMARY

The work effort during the third 6-month period of Phase II concentrated on the following activities: performance characterization of the combustor assembly; a detailed parametric evaluation of the three-body, engine/compressor dynamic system and initiation of the fine tuning of the system; the selection of vendors and fabrication of the Prototype 1 combustor, engine, and compressor hardware; and completion of the assembly of the control box and the outdoor and indoor packages. The subsystem level accomplishments are discussed in greater detail in Paragraphs 1.2.1 through 1.2.4. A summary of the design status of these components is presented in Table 1-1.

Table 1-1. Subsystem Status

Subsystem	Status
Combustor	• Combustor Assembly Performance Characterized
	• Fabrication of Prototype 1 Units Underway
Engine	• Fabrication Underway
Compressor	• Fabrication Underway
Controls	• Assembly Completed
	• Check-out Underway
Outdoor and Indoor Units	• Assembly Completed

1.2.1 COMBUSTOR

The combustor assembly was performance characterized in order to verify the Prototype 1 design. A wide range of firing rates, excess air levels, and working fluid flow rates were tested which simulated various operating conditions. The combustor design was proven to have high performance while providing quiet, safe, and reliable operation. The design goals as summarized in Table 1-2 were all either met or exceeded. The Prototype 1 combustor design was subsequently released for fabrication after the verification of the developmental design was proven.

Table 1-2. Combustor Design Point Performance

	Goals	Predicted From Test
Overall Efficiency	80%	86%
CO Emissions	0.04%	0.04%
Pressure Head	2 inches H_2O	1 1/2 inches H_2O

1.2.2 ENGINEERING RESEARCH COMPRESSOR (ERC)

Significant understanding of the operation of the inertia compressor was obtained during this reporting period, and as a consequence, there presently exists a high level of confidence in the satisfactory operation of the Prototype 1 design. Several operational conditions -- including start-up and compressor stall -- were demonstrated, and the Prototype 1 design was modified accordingly. Additionally, special tests were conducted which investigated the influence of oil flow rate on performance and which ascertained the magnitude of each of the losses which contributed to the overall compressor inefficiency. Also, using the results of the testing, the compressor simulation model was upgraded so that the performance and dynamics of the compressor can be accurately predicted. As a

result of these activities, there is a better understanding of the compressor operation, and the developmental risk once associated with the inertia compressor has been greatly reduced.

1.2.3 ENGINE/COMPRESSOR INTERACTIONS

The analytical effort to better understand the dynamic interactions between the free piston Stirling engine and the inertia compressor continued during this reporting period. An in-depth frequency response parametric analysis was completed which enabled the construction of performance maps for both the engine and compressor. Employing the information generated from this parametric study, the fine-tuning of the Prototype 1 engine/compressor design was initiated; it will be completed early in the 3rd quarter of 1977.

1.2.4 AIR HANDLERS

The assembly and test preparation of both the indoor and outdoor units were completed during this reporting period. These activities included the fabrication of both the refrigeration and hydronic loops, component assembly, installation and checkout of the required instrumentation and leak checking of the final assemblies. Testing is scheduled to start during the third quarter of 1977.

SECTION 2

PROGRAM PROGRESS

SECTION 2

PROGRAM PROGRESS

2.1 COMBUSTOR

2.1.1 SUMMARY

During the second half of 1976, a combustor was fabricated and assembled. The thrust of the effort during this reporting period was to bench test the combustor assembly in order to define its performance characteristics.

The combustor was fully instrumented so that the flow rates could be accurately controlled and so that operating pressures and temperatures could be measured. Wide ranges of firing rates, excess air levels and flow rates of the working fluid were tested in order to simulate different operating conditions.

Water instead of helium was utilized as the cold side fluid in these bench tests in order to simplify test procedures. This is an acceptable approach since the heat transfer mechanism at the heater head is combustion products side limited. The effective overall heat transfer coefficient will, therefore, be similar whether either helium or water is employed as the working fluid.

The combustor has proven to be a quiet, safe and reliable unit while providing outstanding performance; all design goals[1)] have been met or exceeded. Table 2-1 summarizes the anticipated combustor performance based on the data which evolved from the present series of combustor tests.

[1)] The combustor design goals are as follows:

- Combustion efficiency = 80% at 41,400 BTU/Hr capacity based on the lower heating value of natural gas
- CO Emission = 0.04% of flue gas
- Pressure head required for air blower = 2 inches H_2O
- Heat input to working fluid = 33,100 BTU/Hr
- Heater head heat exchanger effectiveness = 90%

Table 2-1. Combustor Performance at the GHP Design Point Operating Condition

Combustion Efficiency = 86%
Heater Head Heat Transfer Effectiveness = 93%
Air Preheated Temperature = 850°F
Preheater Heat Transfer Effectiveness = 68%
Exhaust Temperature = 630°F
Over-all Air Side Pressure Drop = 1.2" H_2O
CO Emission Level = 0.04%

Detail test results are presented in Paragraphs 2.1.4 to 2.1.9. The objectives of the combustor testing and a description of the combustor hardware test facility and instrumentation were presented in detail in Reference 1; they are, however, briefly summarized in the next two sections.

2.1.2 TEST OBJECTIVES

The main objectives of the combustor test were:

1. The verification of the component performance and combustion efficiency predictions.
2. The definition of component operating characteristics
3. A modification of the Prototype 1 design, where required, in order to meet the combustor design goals.

2.1.3 TEST EQUIPMENT & PROCEDURES

Figure 2-1 shows the test facility which was used to characterize and validate the combustor design. A cutaway drawing of the combustor assembly showing the flow paths of the various fluids is presented in Figure 2-2.

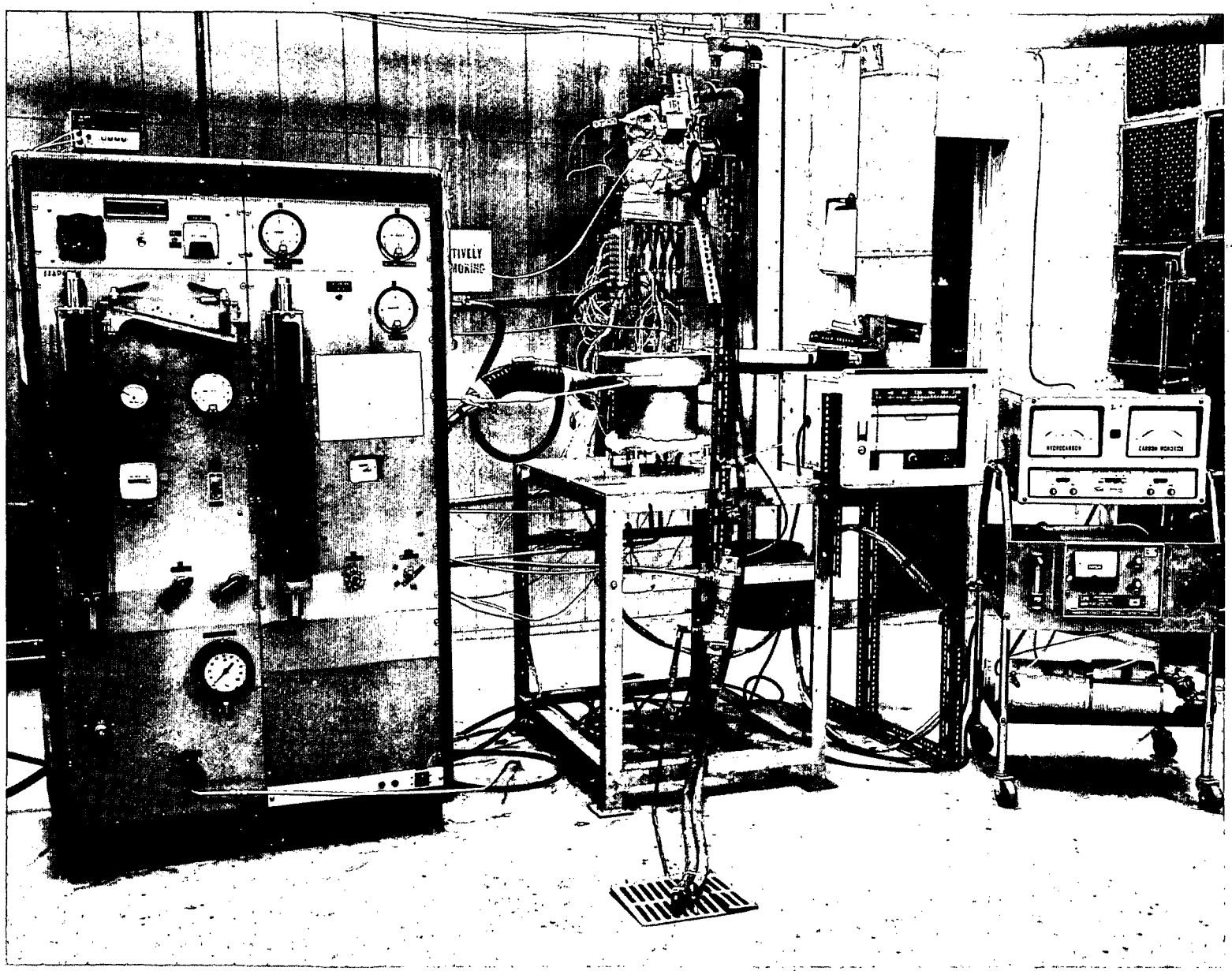


Figure 2-1. Combustor Test Facility

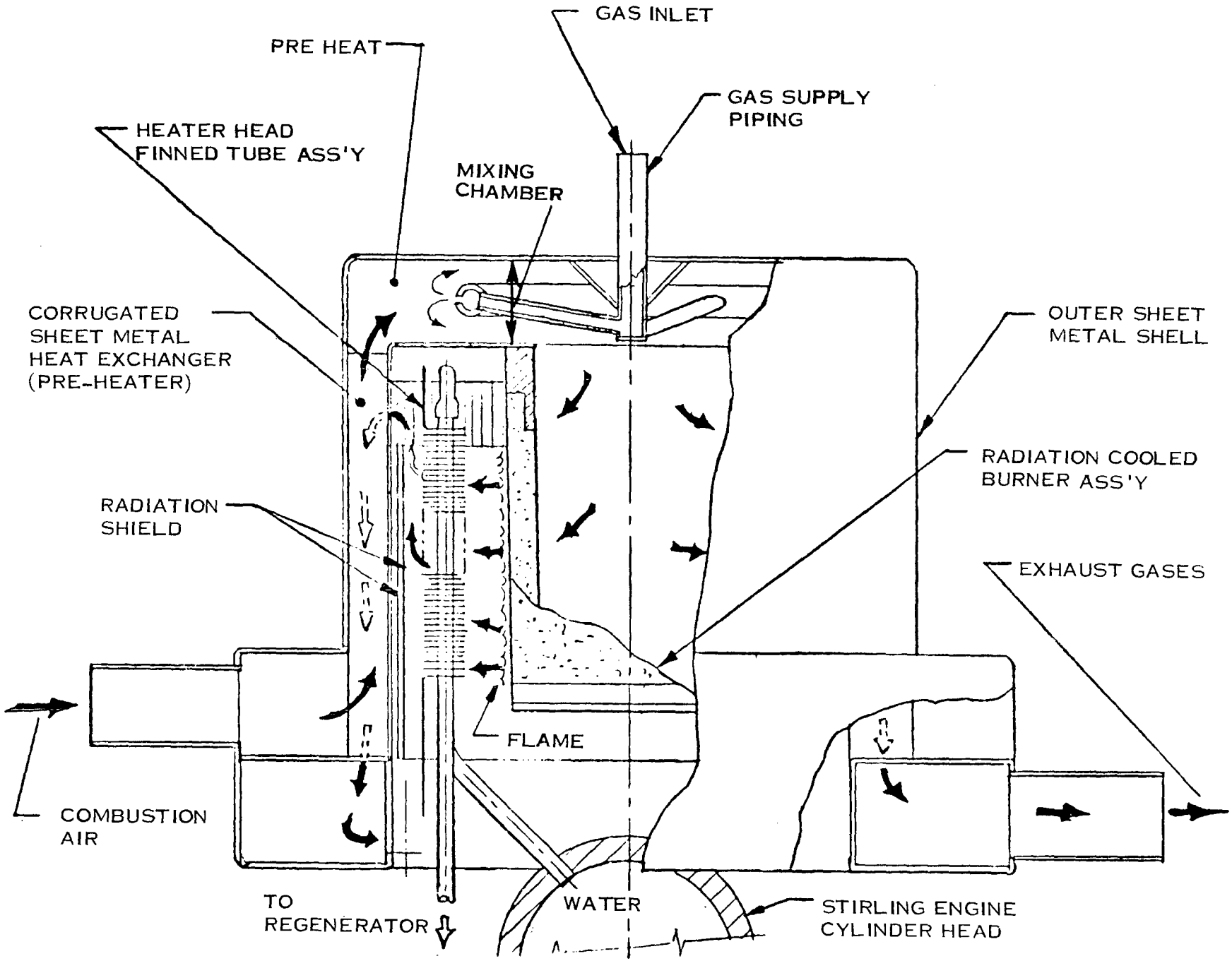


Figure 2-2. Sketch of Flow Paths in Combustor Test

The testing covered the following ranges of operating conditions:

1. Firing rate = 20,000 to 60,000 BTU/hr
2. Excess air level = 0 to 30%

Pressure and temperature measurements at various locations along the fluid flow paths were recorded, and a total of 28 thermocouples were attached to the heater head fins in order to investigate the uniformity of the heat distribution axially along the heater head.

A flue gas analyzer was employed to measure the carbon dioxide (CO_2), carbon monoxide (CO), and unburned hydrocarbon (UHC) emissions at the exhaust. The measured CO_2 emissions at the exhaust were also used to determine the true excess air level. The results of this series of combustor tests are presented in the following paragraphs.

2.1.4 HEAT EXCHANGER EFFECTIVENESS

A non-dimensional heat transfer parameter - the heat exchanger effectiveness - is employed to evaluate the performance of the combustor and preheater heat exchanger designs.

The effectiveness is defined for the combustor configuration as:

$$\text{Effectiveness} = \epsilon = \frac{T_{h_1} - T_{h_2}}{T_{h_1} - T_{c_1}} \qquad (1)$$

where,

T_{h_1} = temperature of the hot side fluid (combustion products) at the heat exchanger entrance

T_{h_2} = temperature of the hot side fluid at the heat exchanger exit

T_{c_1} = temperature of the cold side fluid (water) at the heat exchanger entrance

This equation is the ratio of the actual heat transfer rate in the exchanger to the thermodynamically limited maximum possible heat transfer rate. The subscript h refers to the hot flow (combustion gas) and, c, to the cold flow (water). As suggested by Kays and London (Reference 2) for a given heat exchanger, ϵ is a function of the flow capacitance ratio c_{min}/c_{max} (which approahces zero in the present case) and the heat transfer area number, $NTU=AU/c_{min}$. C is the product of the specific heat and massflow rate, and AU is the product of heat transfer area and the effective over-all heat transfer coefficient. The subscripts min and max refer to the smaller of the two capacitance terms and the larger of the two capacitance terms, respectively.

Based on equation (1), the heat exchanger effectiveness for both the heater head heat exchanger and the preheater can be calculated from the measured temperatures of the air, flue gas and coolant water. The results are plotted in Figure 2-3 as a function of NTU. The predictions given by Kays and London for crossflow (heaterhead) and counterflow (pre-heater) heat exchangers are also plotted for comparison. Figure 2-3 shows a measured effectiveness which is somewhat lower than what theory predicts for the heater head. This divergence from theory can be attributed to an axial variation in the mass flow rate distribution of the combustion products along the heater head heat exchanger. (See Section 2.1.6 for a more detailed discussion of the heater head temperature distribution.) The test data does indicate, however, that at the GHP operating condition where NTU will have a value between 3 and 4, an effectiveness of 91 to 98% for the heater head can be achieved for the Prototype 1 combustor design. The preheater effectiveness for the range of the GHP operating conditions (NTU's of 1.5 to 2.0) will be 60 to 70%; this effectiveness is equal to or better than the 60% which is predicted from theory for a counterflow heat exchanger.

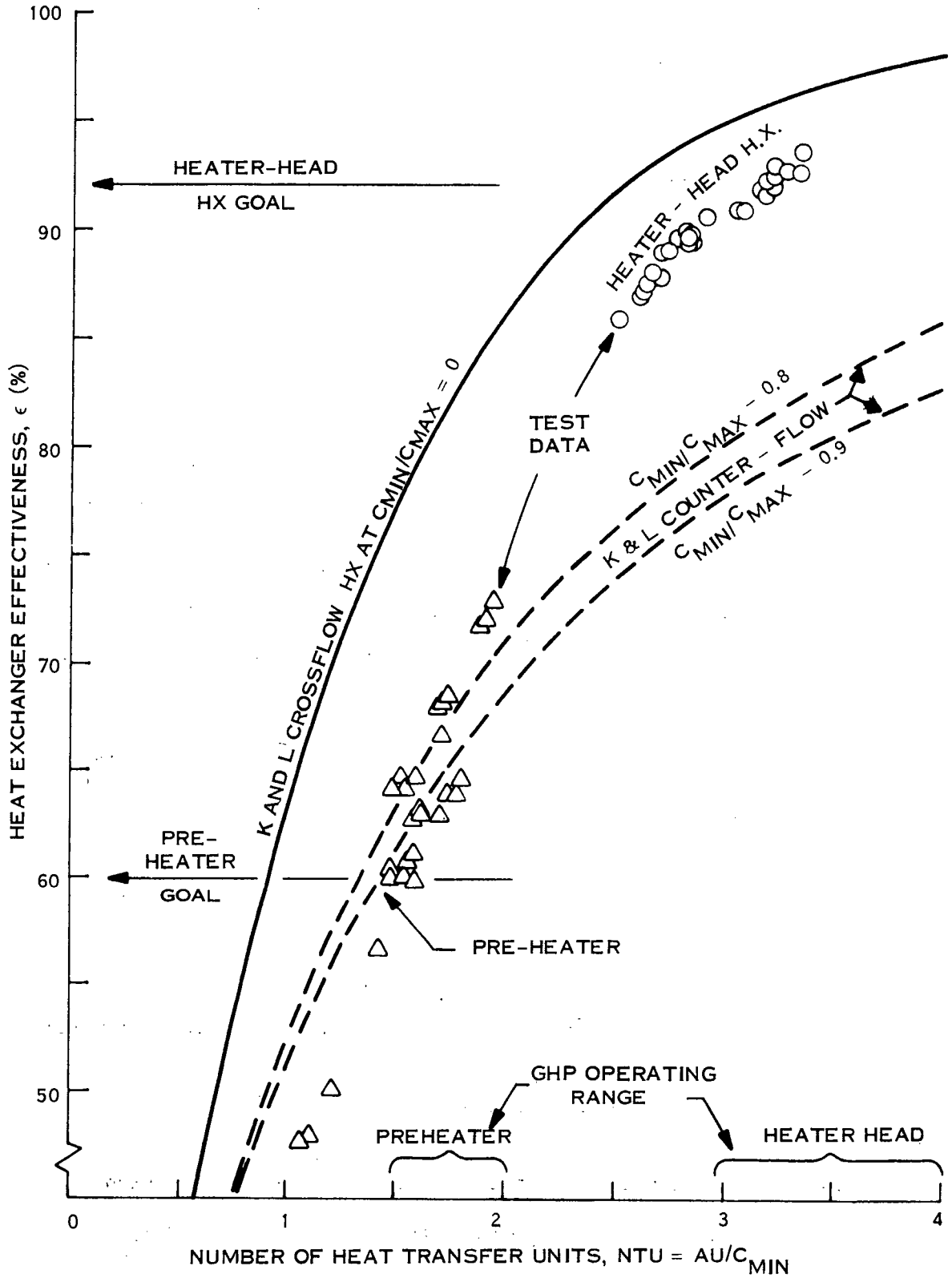


Figure 2-3. Combustor Heat Exchanger Effectiveness

Since both the heater head and preheater effectivenesses are higher than the specified design goals, the overall efficiency of the combustor can, therefore, be expected to be higher than the 80% efficiency requirement imposed from product requirement considerations.

2.1.5 COMBUSTION EFFICIENCY

The overall combustion efficiency is defined as:

$$\eta \text{ comb} = \frac{\text{energy transferred to the working fluid}}{\text{total heat input based on the Lower Heating Value of the natural gas.}} \qquad (2)$$

This efficiency can be determined experimentally from the enthalpy change of the working fluid and from the flow rate of the natural gas. However, fluctuations in the water flow rate attributable to variations in the water supply pressure did not permit an accurate determination of the energy input into the working fluid. The combustion efficiency is more accurately determined from the CO_2 content and the gas temperature of the exhaust products. The flue gas analysis based on a lower heating value of the natural gas (930 Btu/cu ft) results in the following realtionship:

$$\eta \text{ comb} = 1 - \left\{\frac{T_{exh} - T_{room}}{35358\ (\%CO_2)}\right\} \qquad (3)$$

where:

T_{exh} = temperature of the combustion products measured at the combustor exhaust

T_{room} = ambient temperature in the test cell

Figure 2-4 presents the overall combustion efficiency as a function of the exhaust gas temperature for a 15% excess air level; the corresponding fin temperature at downstream side and the air preheat temperature are also plotted in this figure.

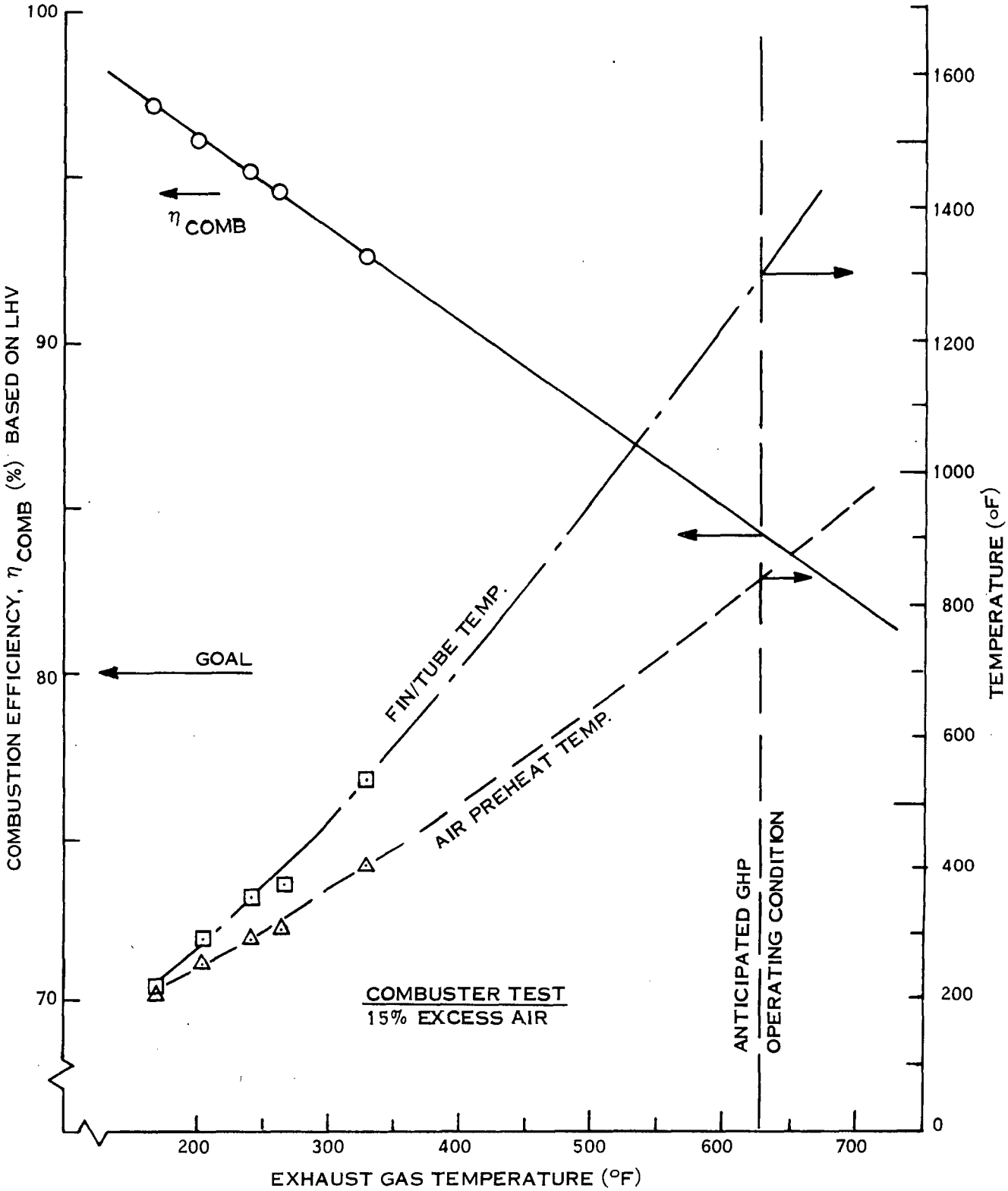
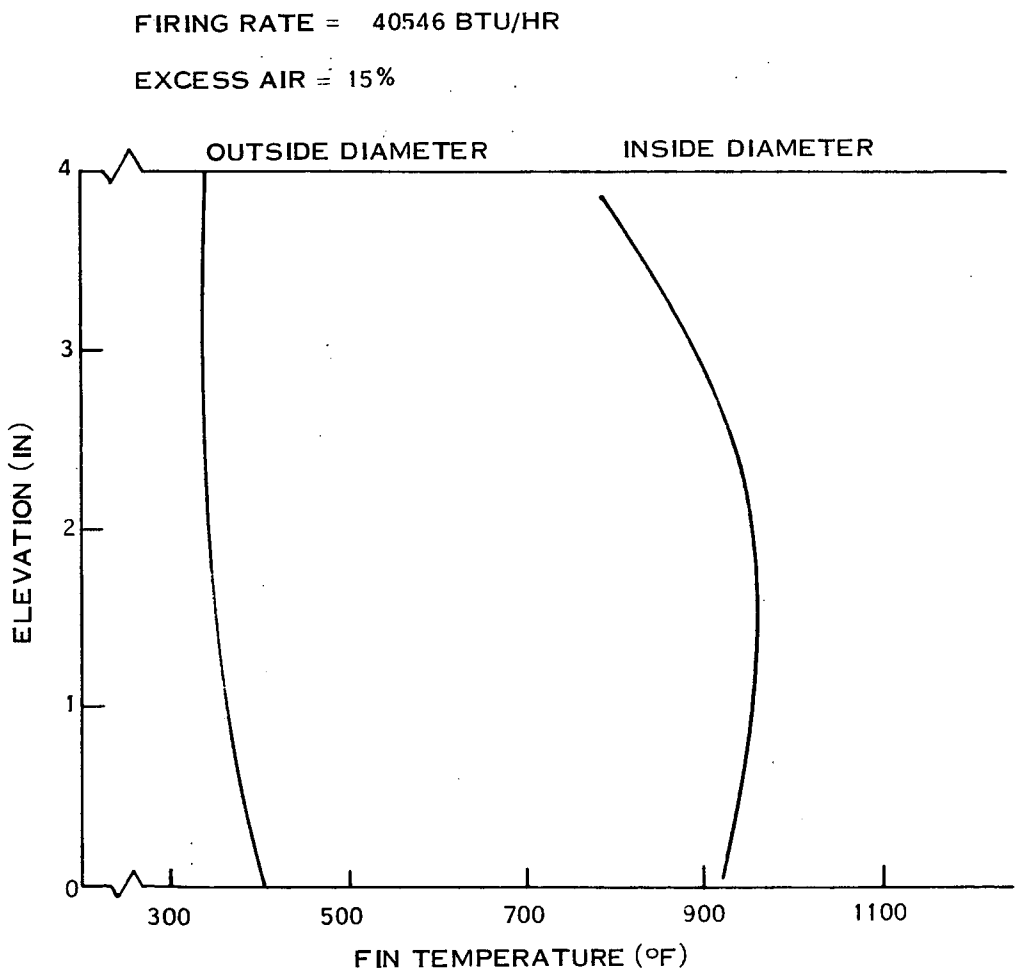


Figure 2-4. Overall Combustion Efficiency of Prototype 1 Configured Combustor

At the anticipated gas heat pump operating condition where the hot side helium temperature will be approximately 1200°F, the heater tube temperature is expected to be 1300°F. Therefore, as can be seen from the data presented in Figure 2-4, an 84% combustion efficiency can be realized. The air preheat and the exhaust temperatures are expected to be 840°F and 630°F, respectively. The combustion efficiency which is projected from the results of the combustor testing is higher than the design goal; this is consistent with the higher heat exchanger effectiveness which was discussed in the previous section.

2.1.6 HEATER HEAD TEMPERATURE DISTRIBUTION

It was shown in Section 2.4 that the effectiveness of the heater head, as determined from the test data, varies from that predicted for an ideal cross-flow heat exchanger. This may be attributable to the non-uniform axial temperature distribution across the heater head. Figure 2-5 presents a typical fin temperature distribution for the Prototype 1 heater head. The temperature of the fin tip facing the flame is minimum at the highest elevation; i.e., where combustion gas traverses the longest path to reach the pre-heater entrance. (The preheater entrance is located at zero elevation.) If the flow path resistance varies axially along the heater head, then, the flow rate of the combustion products will vary along the heater head. As a result, there will be a variation in the axial temperature distribution along the heat exchanger as well as a corresponding variation in the transfer of energy to the engine working fluid. The heater head effectiveess can be enhanced by impoving the uniformity of the flow distribution of the combustion product.



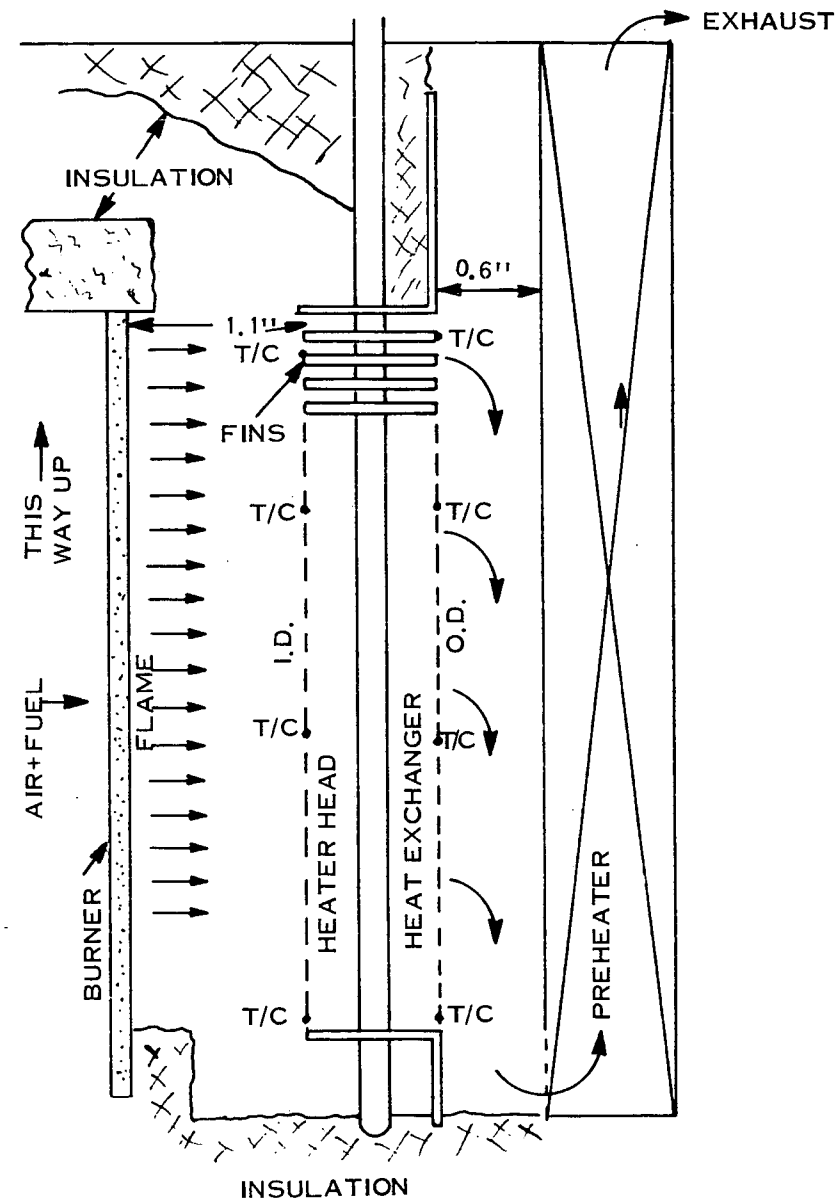


Figure 2-5. Typical Heater Head Fin Temperature Distribution

2.1.7 PRESSURE DROP

The minimization of the air side pressure drop across the combustor is one of the major goals of the combustor. Satisfying this requirement results in a low-cost air blower to supply the low pressure head at the required flow rate.

The measured air side pressure drop across the combustor is presented in Figure 2-6 as a function of the exhaust temperature for a 40,000 Btu/hr firing rate and for a 15% excess air level. The total pressure drop is less than 1.2 inches of water at an exhaust temperature of 630°F - that predicted for the GHP design point operating condition. This pressure drop is well below the design goal of less than 2 inches of water.

2.1.8 EXHAUST EMISSIONS

The CO emission levels initially measured exceeded combustor goals (see Reference 1). Significant reduction in CO emissions were achieved by improving the air fuel mixture uniformity. THE CO emission levels which were obtained during the series of combustor testing are plotted in Figure 2-7 for various firing rates and excess air levels. The target CO emission level of 0.04% was met at the design points of 40,000 Btu/hr and 15% excess air. It is further noted that, when the combustor is eventually incorporated into the outdoor unit, the CO emission as measured at the exit of the outdoor fan will be several hundred times less than that at the combustor exhaust. The actual emission levels of the combustor will be equal to or less than that measured in the present test when the combustor has been integrated into the engine/compressor assembly. These lower emissions will result since the fin temperatures will be higher when helium is employed as the working fluid rather than the water used for calorimetry.

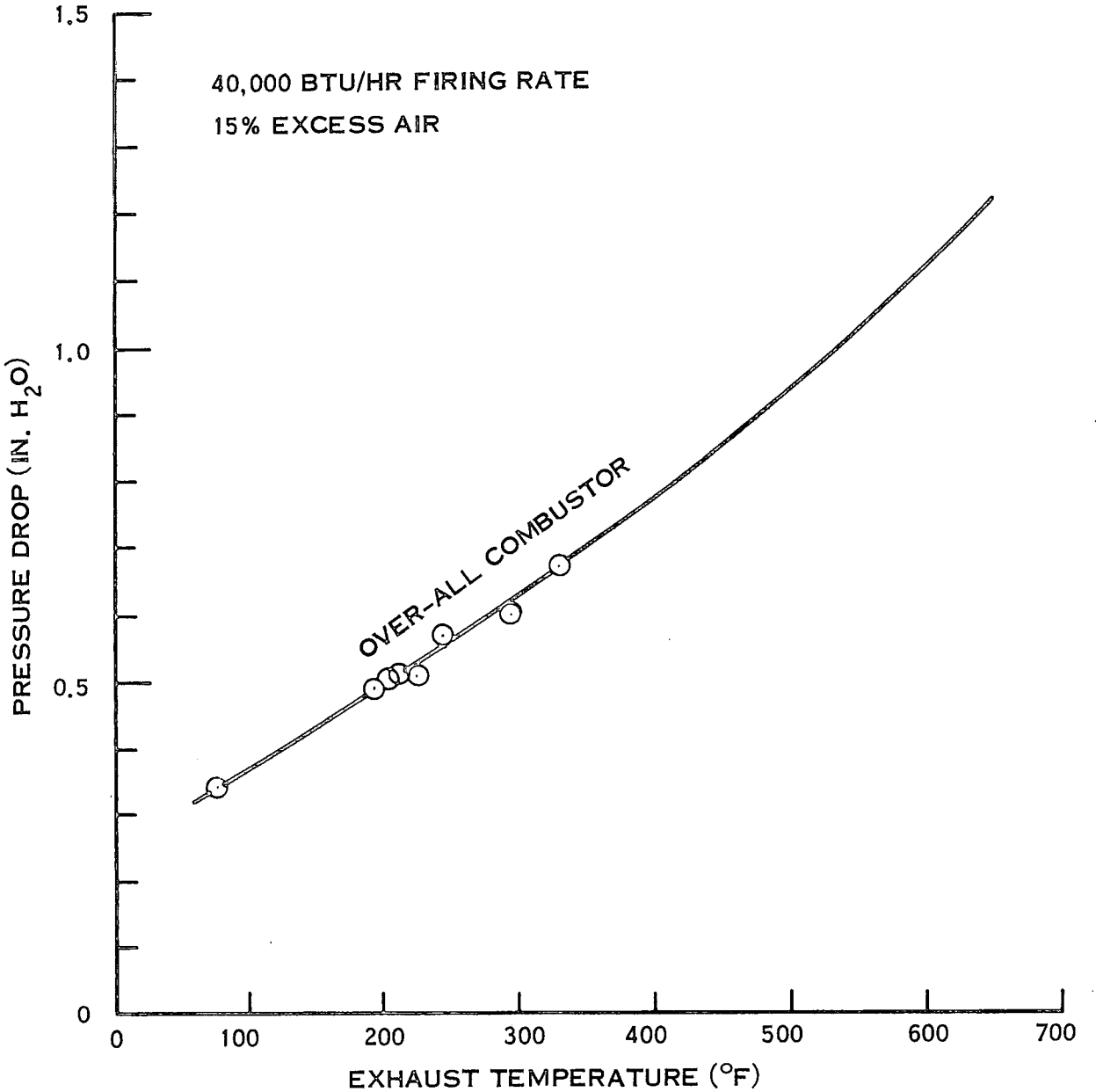


Figure 2-6. Air Pressure Drop Across the Combustor

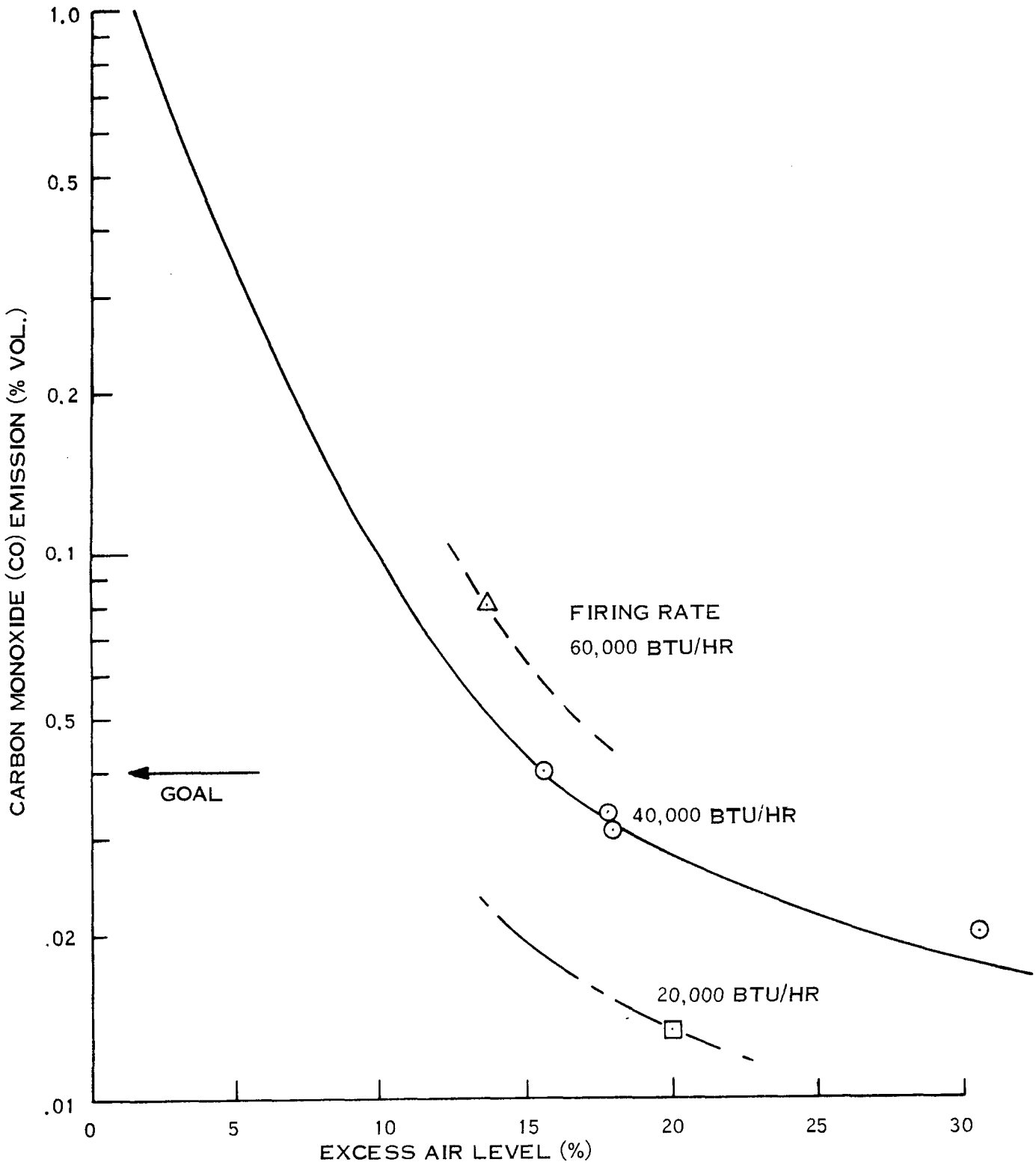


Figure 2-7. Carbon Monoxide Emissions at Various Firing Rates

2.1.9 IGNITION AND RELIABILITY

The combustor design requirements specify the provision that the combustor provide for safe and reliable operation. Testing has identified the proper location for the spark plug and the flame detector in relation to the transpiration burner. Subsequently, ignition has been 100% successful, and the flame, once ignited, has been sustained throughout the entire operating period.

No audible noise can be detected during combustor operation except that of the air blower. "Combustion noise" using the transpiration burner is non-existent.

After having accumulated over 200 hours of operation, no noticeable degradation has been found on any of the combustor or heater head parts. Even though the heater head fins were exposed to temperatures in the range of 900° to 1300°F, no severe oxidation of the metal has been encountered.

2.1.10 CONCLUSION

The Prototype 1 combustor promises to be a high performance, quiet, safe, and reliable unit. The present series of tests has successfully demonstrated that all major design requirements of the combustor can be met.

2.2 COMPRESSOR

2.2.1 ENGINEERING RESEARCH COMPRESSOR TESTING

The objectives of the Engineering Research Compressor (ERC) test program were as follows:

1. Verification of an inertia (free piston) type refrigerant compressor design concept in the 3 ton size range.

2. Generation of empirical data for use in defining and substantiating the Prototype 1 inertia compressor design features.

3. Establishment of preliminary material compatibility data.

4. Evaluation of performance and operational characteristics unique to a free-piston inertia type compressor

A thorough description of the ERC hardware, test instrumentation, and test facility is presented in Reference 1. The assembly drawing of the ERC is presented in Figure 2-8. The results of the testing are presented in the following paragraphs.

2.2.2 TEST RESULTS

2.2.2.1 Compressor Start-up

Early in the test program it became evident that the ERC, a vertically mounted, double-acting, free piston compressor, had unacceptable start-up characteristics because of its particular free piston configuration. That is, if permitted to stand idle for several minutes, the piston assembly would drift downward until the back face of the top piston rested on the upper face of the mid-housing seal assembly. When startup of the compressor was attempted, a low pressure or suction would form in the small volume between these faces. This resulted in a sufficient pressure unbalance across the total piston assembly such that the piston could not break loose - i.e., move upward - in order to begin compressing the refrigerant. This start-up problem was resolved by the incorporation of a small check valve into the suction line that supplied gas to the entrapped volume between the piston and mid-housing faces[2]. Each successive piston

[2] This entrapped volume is designated the gas spring.

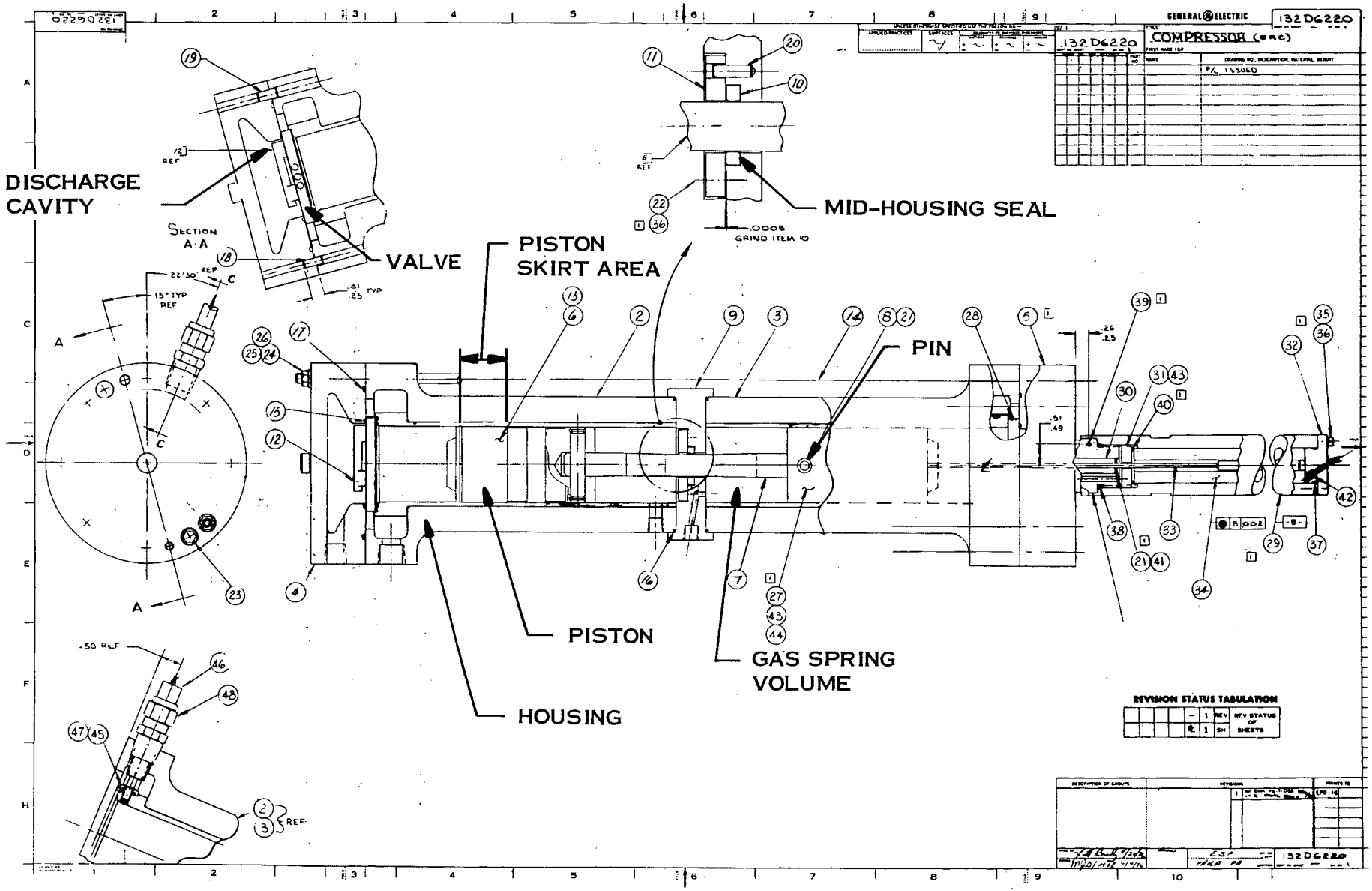


Figure 2-8. Engineering Research Compressor (ERC)

stroke draws an added charge of gas into the low pressure volume during start-up, thereby, permitting the piston to initiate its upward motion ("walk-up") and to center itself so that pumping can begin. The check valve operates only during start-up since steady state operation results in pressure levels greater than suction pressure inside the gas spring valve. A typical start-up sequence is shown in Figure 2-9 after the above feature was incorporated into the ERC design.

2.2.2.2 Oil Flow Rate

Tests were completed to evaluate the effects of oil flow rate on compressor performance. No signficant performance changes were observed for variations in oil flow rate ranging from a minimum of 0.2% to a maximum of 1.1% by weight of total R-22 mass flow. Adequate lubrication was demonstrated with this oil flow rate as evidenced by satisfactory operation without a related failure throughout the test program. Also, post-test disassembly and examination showed no evidence of inadequate or marginal lubrication of the ERC components. On the basis of this test, an oil flow rate of 0.7 $\pm$ 0.5% by weight (oil to R-22) will be specified for Prototype 1 compressor testing.

2.2.2.3 Material Compatibility

The compatibility of operating a cast iron piston in a cast iron liner in an inertia type compressor was substantiated as evidenced by the satisfactory operation of the ERC. Post-test examination of these parts also showed no evidence of excessive wear or galling of parts.

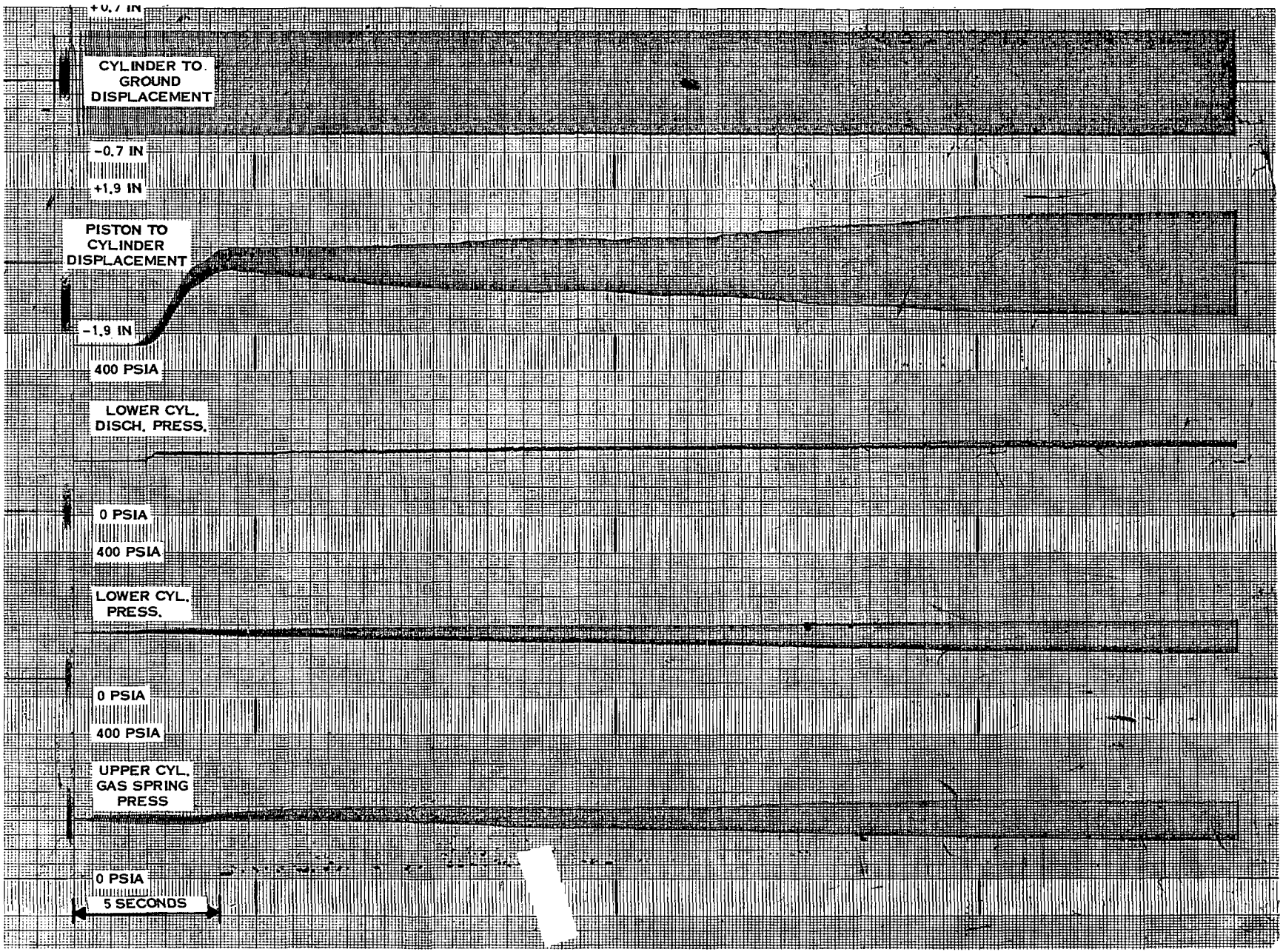


Figure 2-9. Typical ERC Start-up with Check Valve Incorporated

2.2.2.4 Compressor Stall and Restart Characteristics

Tests were conducted to evaluate the performance of the ERC when the driving function is shifted off the resonance point. This was accomplished by increasing the refrigerant loop condenser pressure and, correspondingly, the compressor pressure ratio to that point where the compressor stalled; stall did occur as predicted by analysis. Furthermore, the compressor fully recovered from stall when the condenser pressure was returned to normal.

2.2.2.5 Efficiency Loss Evaluation

A series of tests were conducted to evaluate various efficiency losses due to friction, leakage, and discharge overpressure.

2.2.2.5.1 Friction Tests

"Free-fall" tests were conducted on an ERC piston within its cylinder (open-ended) to evaluate and compare friction losses for several different piston configurations. The following configurations were tested: (1) the original piston design with full skirt area and a piston ring, (2) the same piston without the piston ring, and (3) the piston with and without the piston ring but with approximately one-half the skirt area removed. Each of these configurations was subjected to static leakage tests in addition to operational tests in the ERC in order to determine performance variations. The results of these tests showed that piston friction losses could be reduced by approximately one-half while maintaining the same volumetric efficiency, by removing the piston ring from the original piston configuration. Friction losses were also reduced by cutting back area; however, this was accompanied by an even greater increase in the leakage losses.

2.2.2.5.2 Connecting-Rod Seal Leakage

The connecting-rod seal consists of a "floating" cast iron ring mounted inside the compressor mid-housing; sealing is achieved by having a small running clearance between the seal inside diameter and the connecting-rod outside diameter. This floating seal design provides some forgiveness for minor misalignemnts. An o-ring face type seal was later incorporated into the design so that the overall seal leak path would be appreciably reduced while not further restricting the allowable ring movement. Static leak tests confirmed a reduction in seal leakage by 35 to 40% by incorporating the o-ring back-up.

Reference is made to Figure 2-10 for a pictorial representation of the relative efficiency improvements identified by the above noted test series.

2.2.2.6 Free Piston Modelling

Test data taken throughout the ERC test program were factored into a free piston, compressor model. Excellent corrrelation is obtained between predictions and actual compressor performance. Figure 2-11 presents a typical measured versus predicted relationship for compressor isentropic efficiency as a function of frequency. Other parameters such as refrigerant flow rate and piston to housing phase angle are consistently predictable with similar accuracies; these correlations are presented in Figure 2-12 and 2-13, respectively.

2.2.2.7 Gas Spring Evaluation

The analyses supporting matching of engine and compressor characteristics which was performed in the prior reporting period determined that the gas spring between the piston backface and the housing mid-section (the gas spring volume)

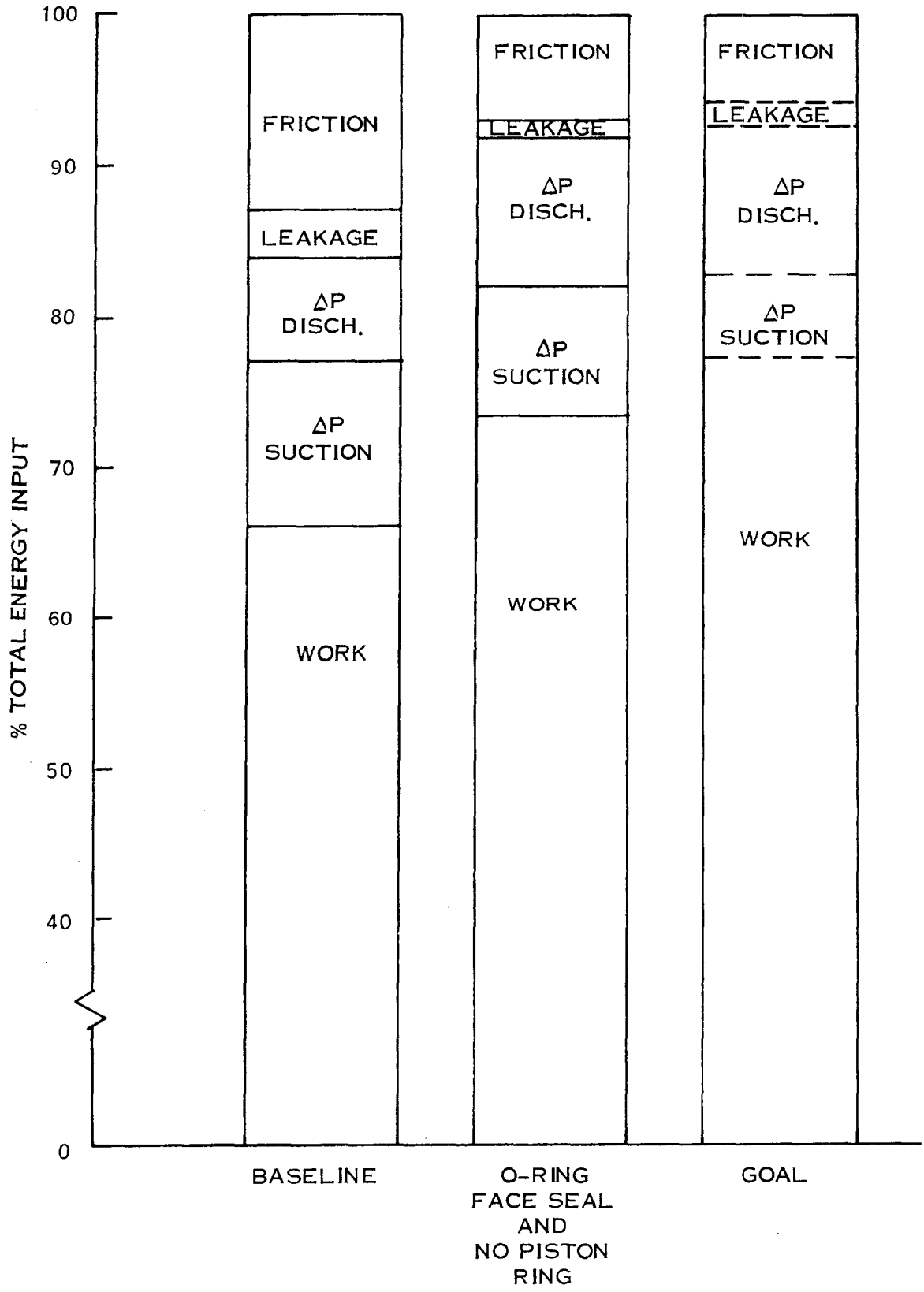


Figure 2-10. Typical ERC Energy Input Apportionment

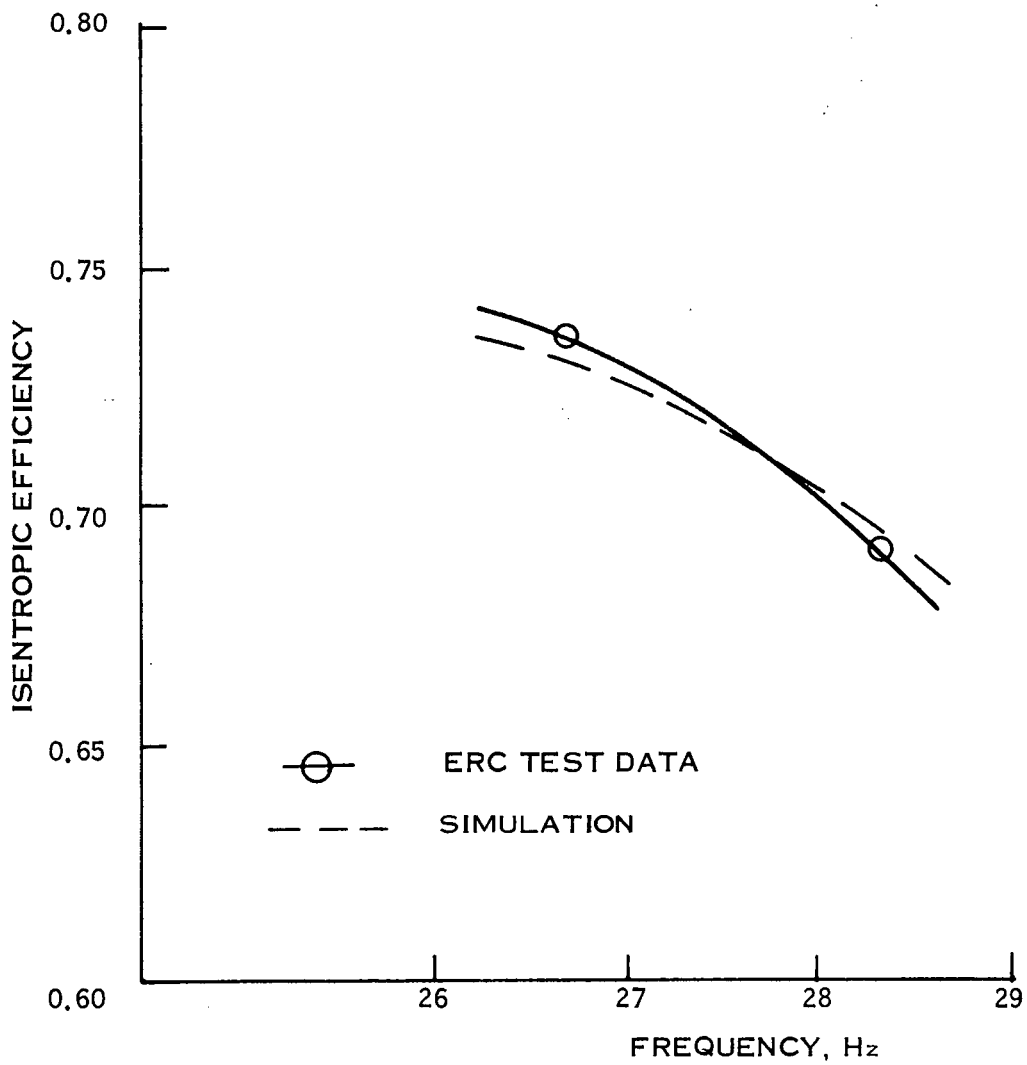


Figure 2-11. Typical ERC Efficiency

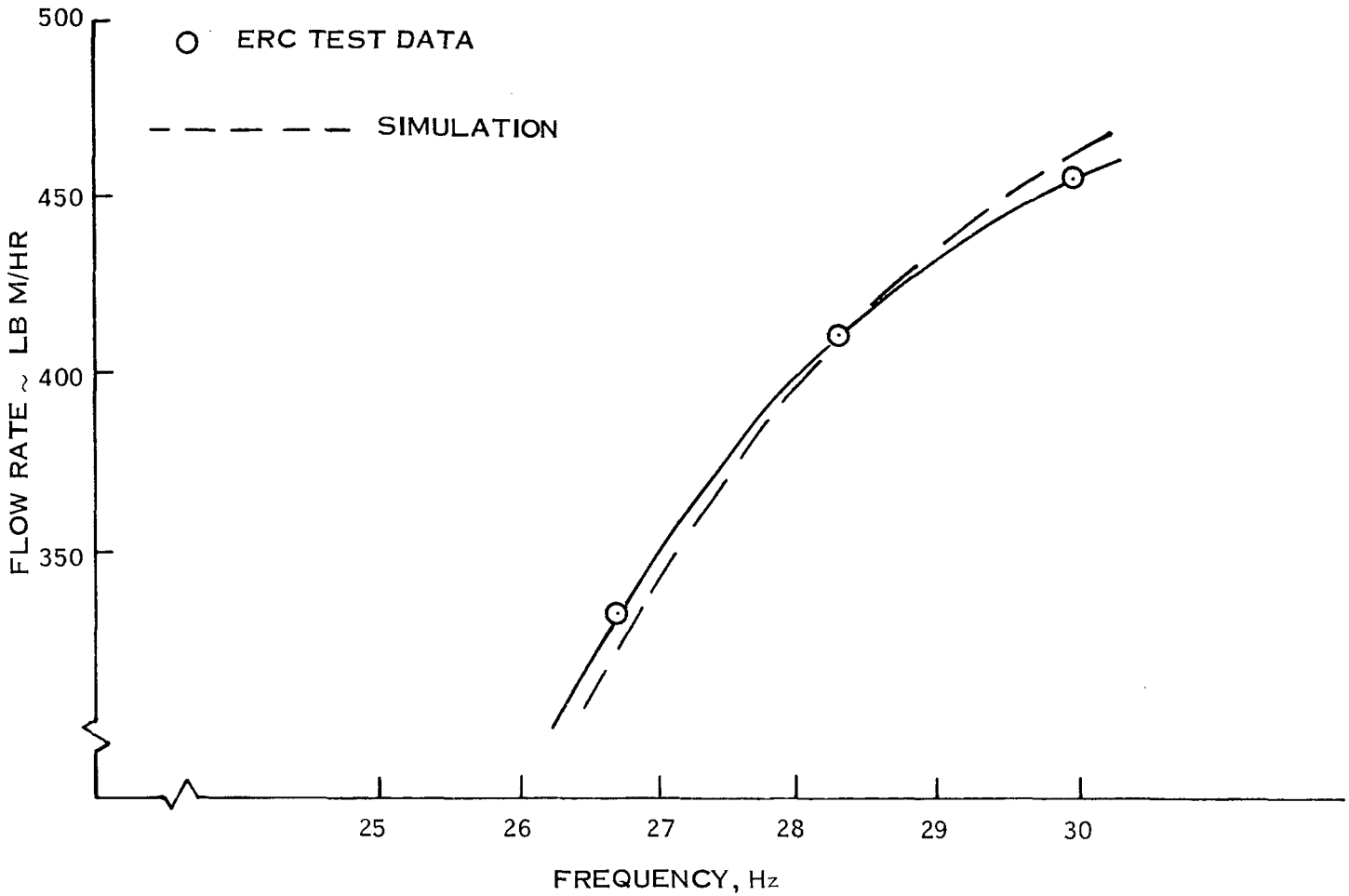


Figure 2-12. ERC Refrigerant Flow Rate

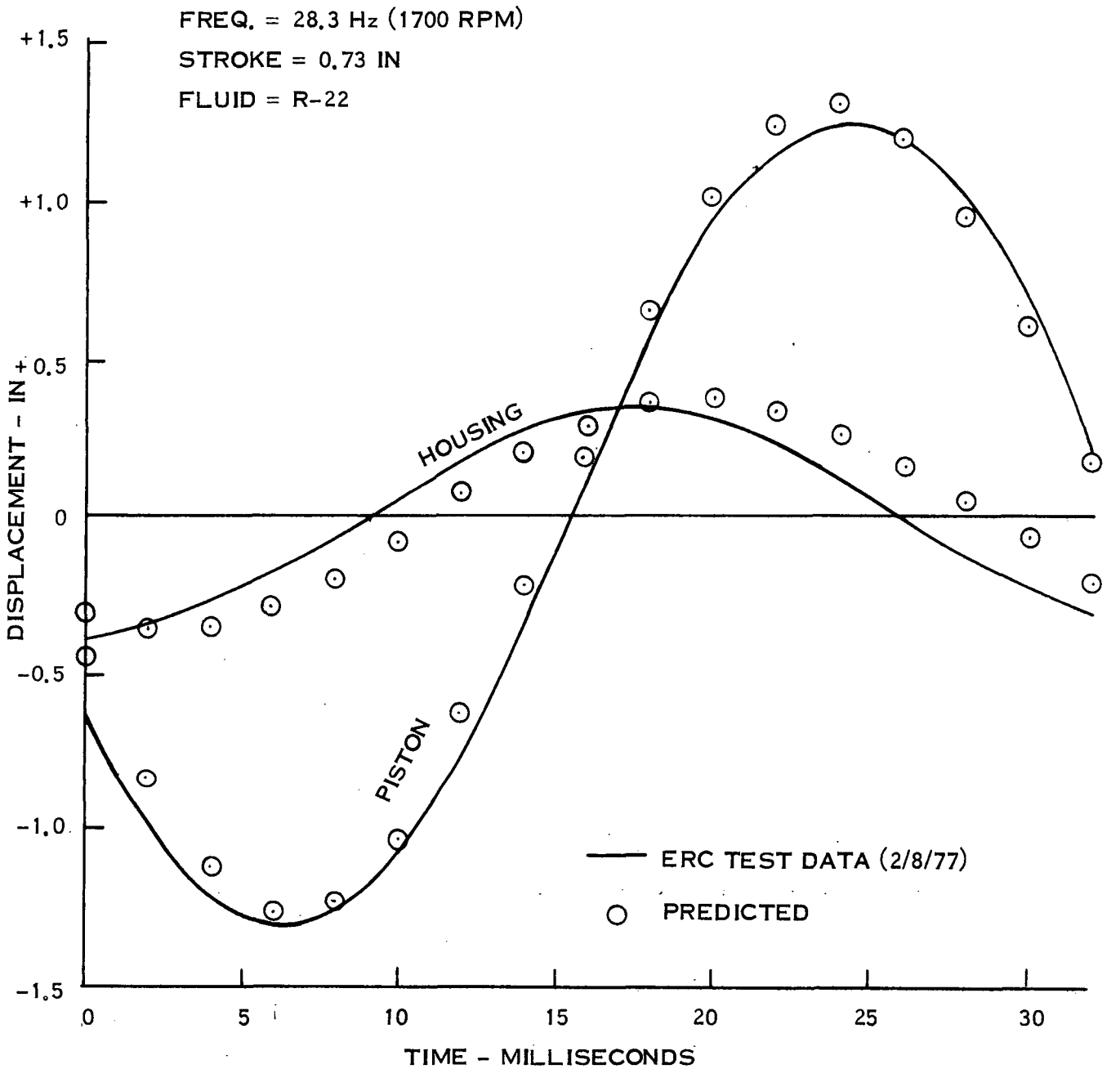


Figure 2-13. ERC Phasing Relationships

should be "soft" (low spring rate) if stable engine operation were to be achieved. A "soft" gas spring can be incorporated in the ERC design by increasing the volume in this chamber by 20% from the original design; this enlarged volume will reduce the gas spring constant by 10%. Tests were performed and completed on the ERC during the present reporting period to evaluate compressor performance with a "soft" gas spring configuration. The test results verified the predicted performance.

2.2.2.8 Piston to Piston-Pin Design Change

A design change was completed to reduce the piston-pin induced stress levels in the piston. This change was incorporated because of the development of a hairline crack in the ERC piston in the section between the piston-pin hole and the back face of the piston (see Figure 2-14). The crack was observed during an interim tear-down inspection following the development tests relating to compressor start-up (reference Paragraph 2.2.2.1). It is noted that the piston assembly absorbed many shock loads during this test sequence for the piston repeatedly impacted the mid-housing when start-up was not successfully achieved.

An analysis of the initial design configuration showed that the piston stress levels were well within accepted design limits for normal operation. However, to provide addtional margin for anomalous operating conditions, the piston-pin was relocated to increase the piston support area and thus reduce stress levels by a factor of 3 to 1.

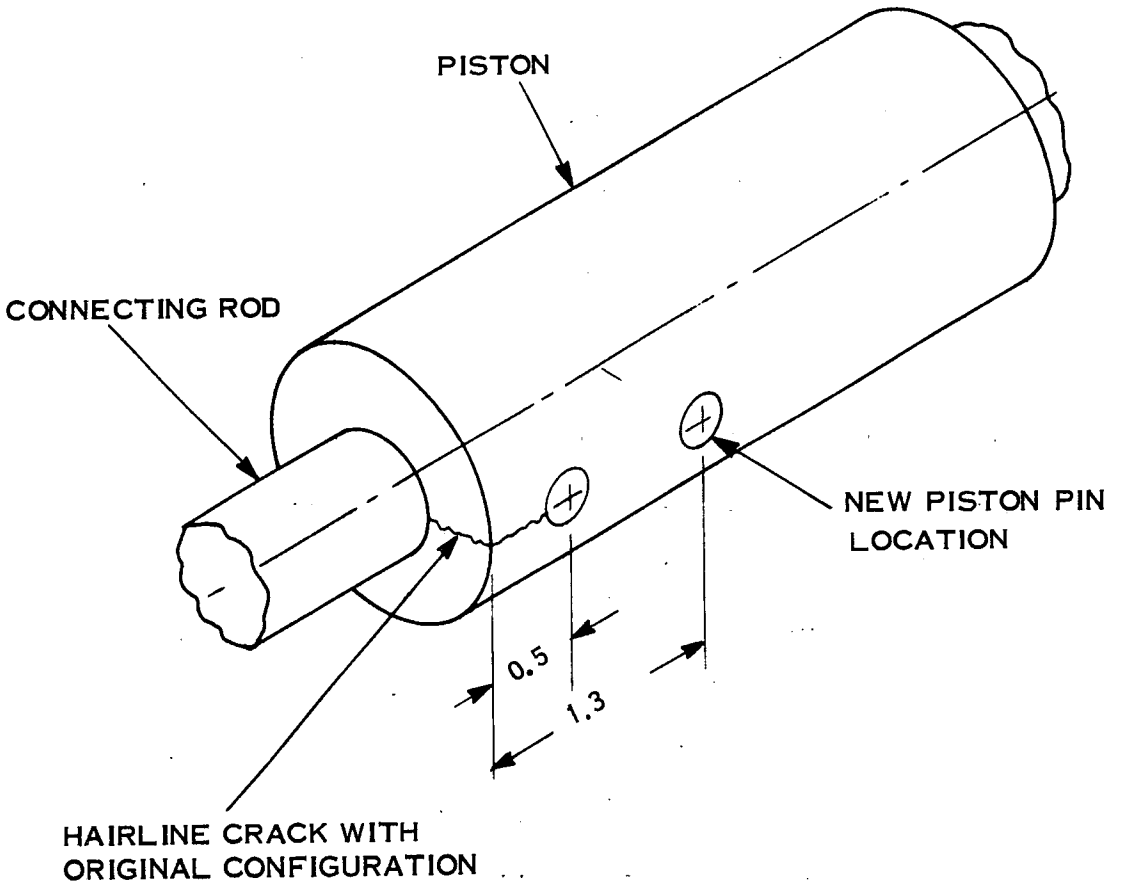


Figure 2-14. Piston to Piston-Pin Design Change

2.3 ENGINE/COMPRESSOR INTERACTION

2.3.1 SUMMARY

Prior analytical effort in this area had been directed toward a sensitivity analysis of the various engine design parameters. This work showed a strong need for a more basic understanding of free piston machinery performance characteristics. To this end, an in-depth frequency response parametric analysis was completed. The results of this study enabled the construction of performance design maps for both the engine and compressor. Based on this entire analytical effort, fine tuning of the engine/compressor design can be completed.

2.3.2 PARAMETRIC STUDY

In order to meet the requirements of the heat pump system for efficiency and capacity, a high performance engine/compressor design is required which is well matched over the entire operating range. A sensitivity analysis had been performed earlier in order to establish the variation in the engine characteristics as a function of the various design parameters (see Reference 1, Page 2.12). A significant result of this study was that certain parameters have a controlling influence on one or more of the engine performance characteristics. These parameters were identified as tuning parameters. Table 2-2 lists these features as well as their performance effects. Because of the wide variation in engine performance attainable through manipulation of the tuning parameters, Prototype 1 was designed with considerable flexibility. In this way the hardware can be "fine tuned" for high efficiency with a proper compressor/load match. The following example demonstrates the wide flexibility allowed by the Prototype 1 engine/compressor design philosophy: Engine power

output, at a constant engine heat exchanger temperature can be nearly doubled by changing the displacer rod diameter from 0.90 inches to 1.10 inches. (Note, however, that the other engine parameters will also change.) Other tuning parameters have an equally strong influence on frequency, stroke, and efficiency. Until this time, one of the major difficulties in specifying a "fine tuned" engine design was that, as one parameter was varied, the effect of changing one or more of the other parameters was uncertain. In other words, the effect of parametric variations in the design can be analytically determined only for small perturbations off the nominal and only for one parameter at a time. A thermodynamic model had been used to study and verify detailed performance. However, because of the computation time and cost required to produce a steady state solution, this approach allows only a limited number of design iterations to be simulated.

Table 2-2. Free Piston Engine Tuning Parameters

Parameter	Performance
Displacer Rod Diameter	Phase Angle
Displacer Spring Volume	Phase Angle
Charge Pressure	Frequency
Piston Mass	Stroke
Displacer Mass	Phase Angle

In order to augment the findings of the sensitivity analysis, a thorough parametric study was completed. The thrust of this effort was directed toward the fine tuning of Prototype 1 engine/compressor design parameters.

2.3.3 FREQUENCY RESPONSE ANALYSIS

A simplified free piston engine/compressor model was examined in terms of a frequency response analysis. The basic model, which is shown in Figure 2-15, employs linear springs and dampers to model the compression, expansion, and the isothermal heat exchanger pressure drop. The refrigerant compressor spring rates were determined from compressor natural frequency data that were derived as a function of load from the Engineering Research Compressor (ERC) test data. Accordingly, the compressor spring rate was changed to reflect a particular ambient day refrigeration loading. First, the frequency response characteristics were studied in order to determine the operating characteristics for a given design and a given loading. A typical result is shown in Figure 2-16. This particular plot indicates a first harmonic resonance at 27 Hz and a second at 46 Hz. In this case, the first harmonic is the dominant resonance, and, similar to an oscillator, the engine/compressor would operate at 27 Hz. The calculations of the dynamic parameters associated with the resonance point will closely approximate the engine stroke, frequency, and phase angle. The power output and efficiency can be determined from these values, with close agreement to a thermodynamic model computer analysis for the same design configuration.

Hundreds of configurations can be simulated with this type of approach, and the results evauated. The data was catalogued in the type of plot format shown in Figures 2-17, and 2-19. These particular curves indicate that the variation in stroke, and phase angle for a wide range of compressor natural frequences and for several displacer gas spring volume sizes. All of the other engine parameters were held constant. However, parametric variations were studied for ten design variables, including each of the tuning parameters listed in Table 2-2. Even though the trends in each parametric set are valid only if all the

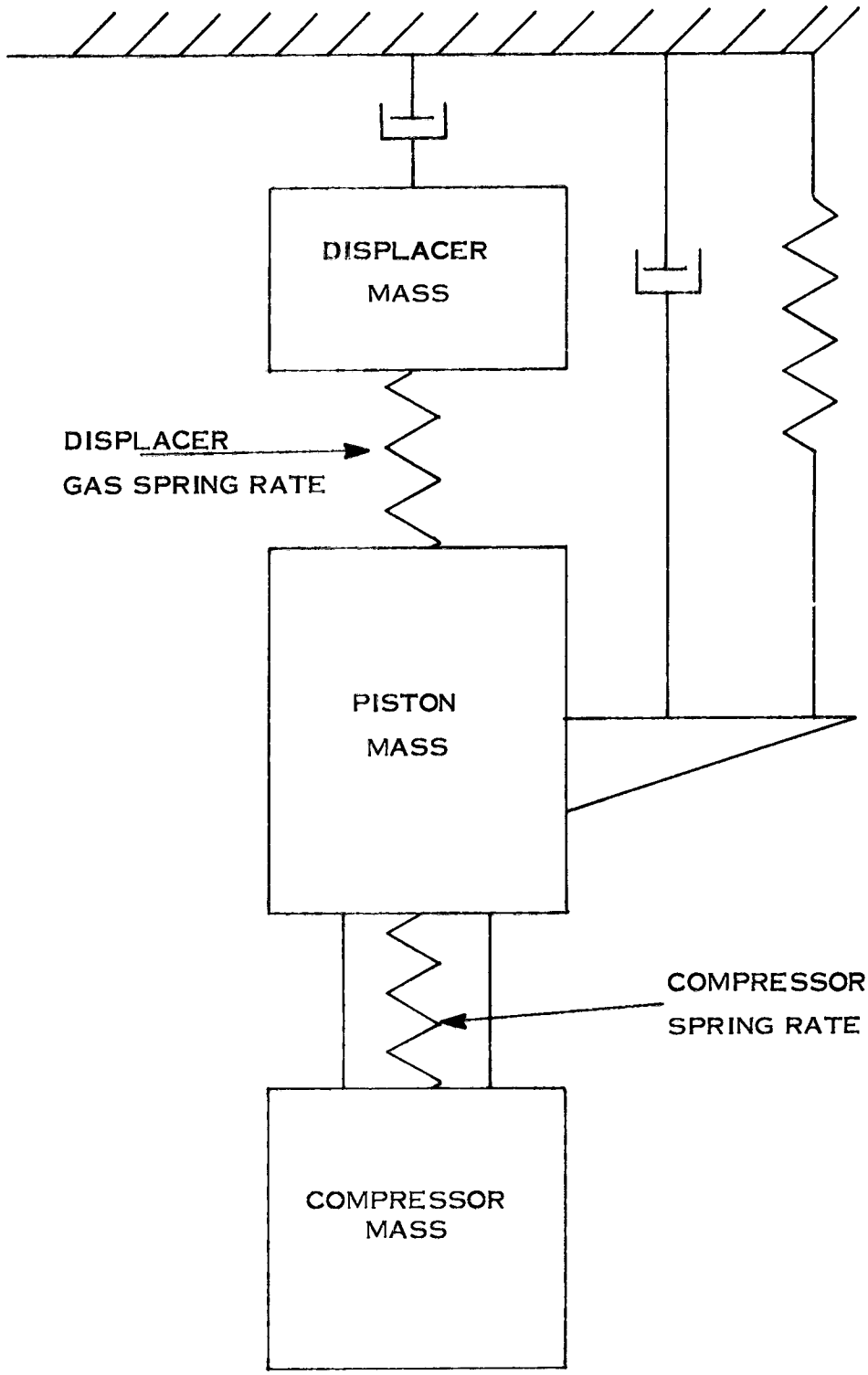


Figure 2-15. Three Body Dynamic Model

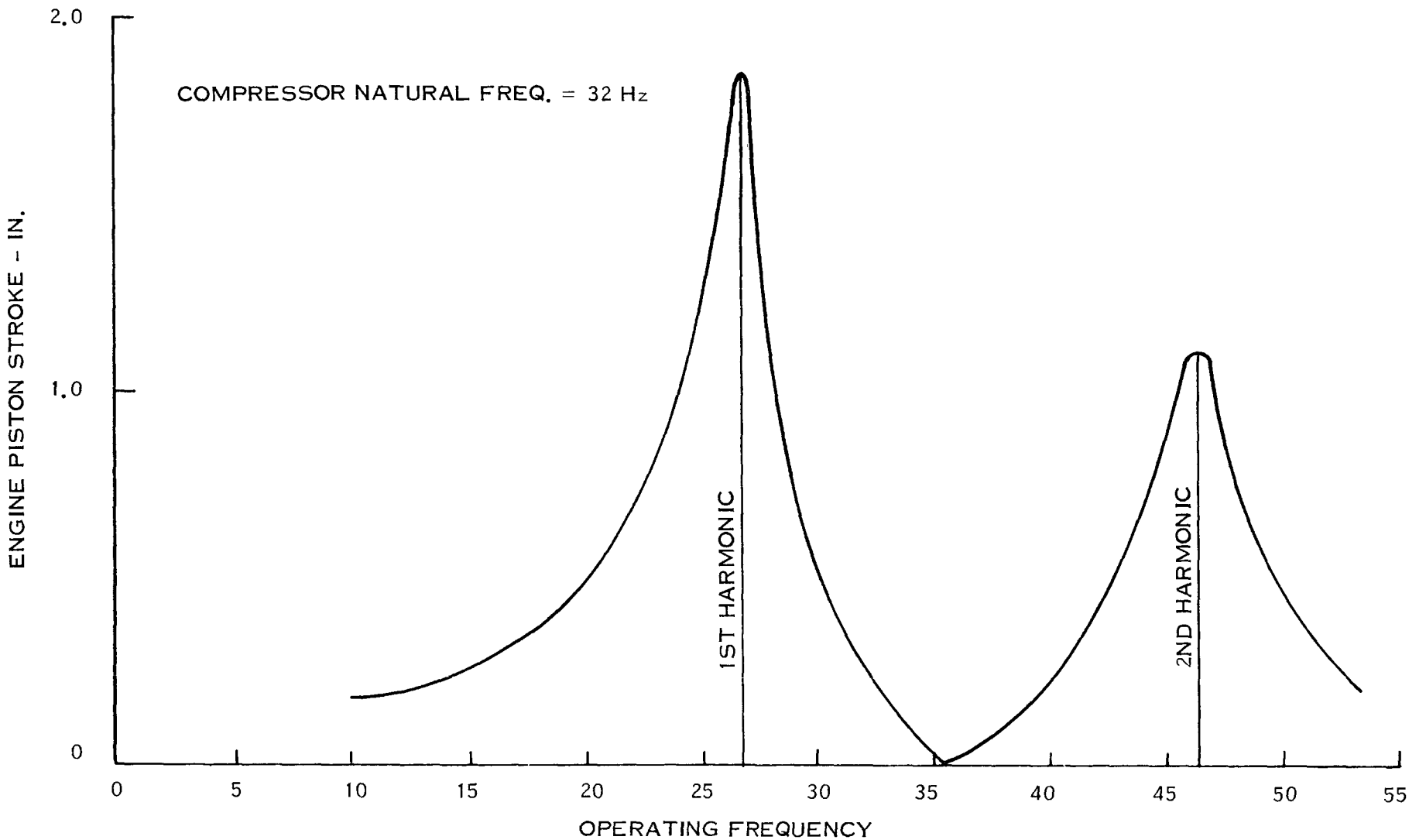


Figure 2-16. Frequency Response

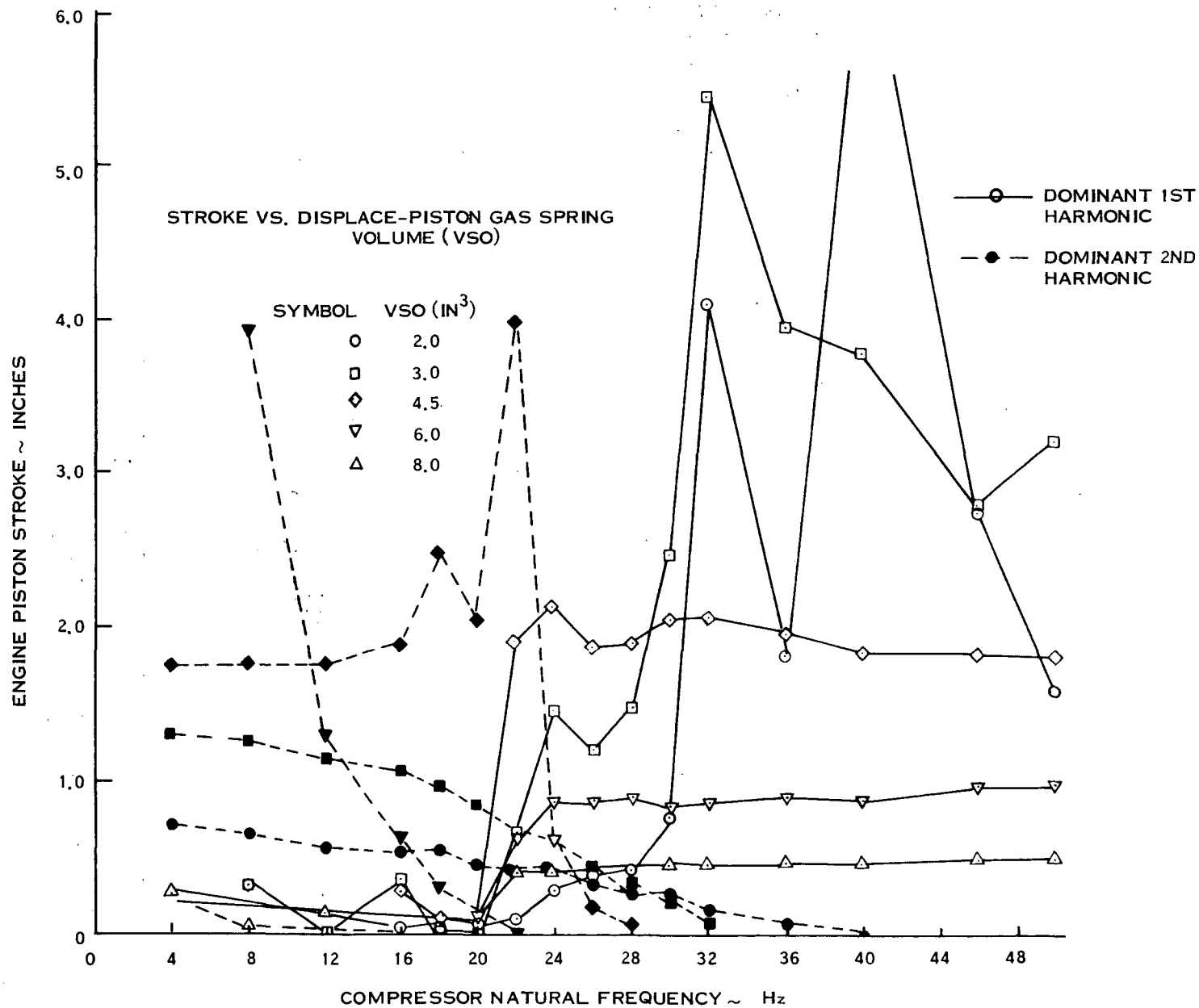


Figure 2-17. Engine Stroke vs. Displacer Gas Spring Volume and Compressor Natural Frequency

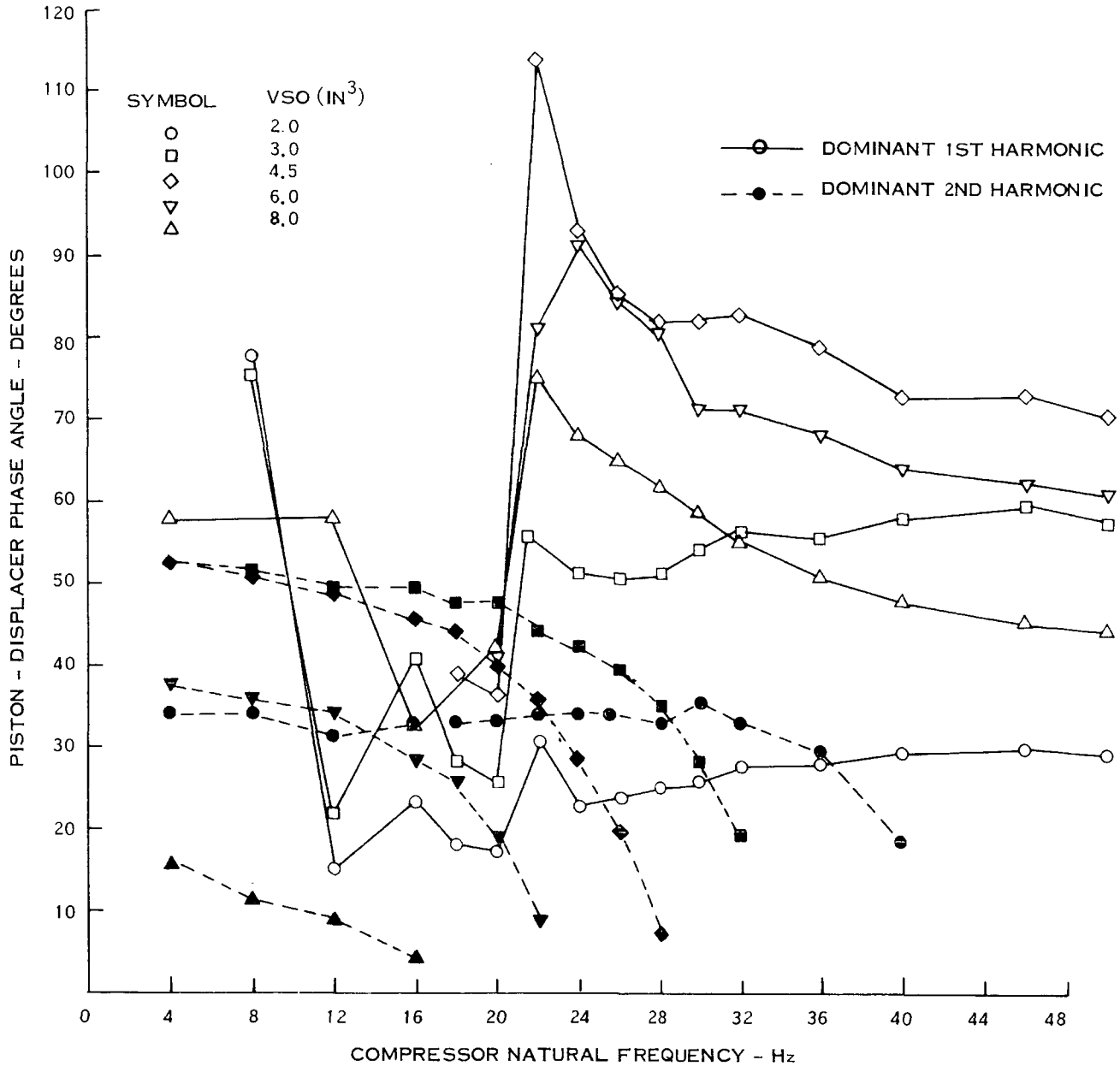


Figure 2-18. Phase Angle vs. Displacer - Piston Gas Spring Volume and Compressor Natural Frequency

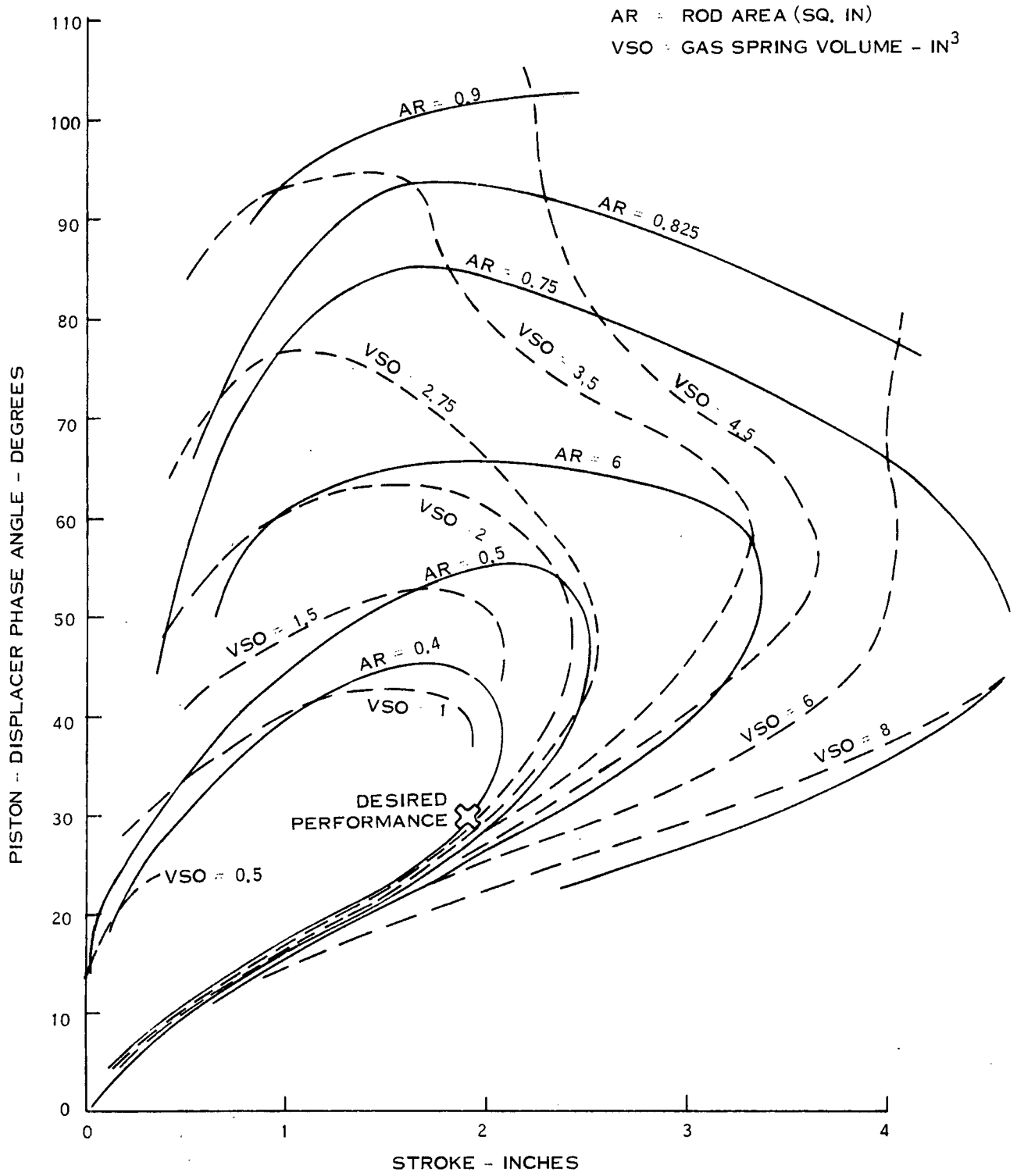


Figure 2-19. Performance Configuration Map

other variables are held constant, they are, nevertheless, invaluable in the following sense: the curves accentuate the proper approach for altering a design so that a favorable load response will result allowing the parameters with the most-significant influence to be readily identified. For example, Figure 2-17 indicates that increasing the displacer gas spring volume will have a large effect on engine stroke but that it has little effect on frequency. In fact, a thorough study of all the parametric variations showed that a most dramatic change in stroke and phase angle can be affected by only varying the displacer rod diameter and the gas spring volume. So, with the exception of these two, all of the other timing parameters can be fixed in order to attain an approximation of the desired performance at a given loading. Then, with only two variables remaining, a performance-configuration map can be generated from which a final configuration can be chosen. Figure 2-19 illustrates a typical performance-configuration map for a specified Prototype 1 design at a 95°F day loading. In order for this particular configuration to deliver 3 kW to the refrigerant load, a 1.9 inch stroke is required. Also, the demand for a high efficiency engine implies that the phase angle should be about 30 degrees (see Reference 1, Page 2-16). From Figure 2-19 it is evident that, in order to satisfy these conditions, the displacer gas spring volume should be set to a value of 1.75 cubic inches with a rod diameter of 0.71 inches (rod area = 0.4 in.^{2}). In fact, a thermodynamic performance analysis has verified that the Prototype 1 engine tuned in this manner will generate 3 kW power output with an accompanying high thermal efficiency.

The above discussion is concerned with tuning the engine for a given loading. However, Figures 2-17 and 2-18 show how the engine performance can appreciably change over a range of compressor load natural frequencies. This variation highlights the need for also "tuning" the compressor design. Here, tuning

infers an adjusting of the effective compressor spring rates in order to limit or control the excursions in engine performance. The Prototype 1 compressor design does provide for limited parametric freedom especially in the "soft" or centering gas spring design. The gas spring volume and the porting scheme employed to reference this volume can be changed so as to influence the freuency band width and/or the level (high or low) of compressor natural frequencies. Figure 2-20 illustrates the correlation between natural frequency and ambient temperature for the present Prototype 1 compressor design, and it also indicates the extended frequency range in which the Stirling engine would be compatible. This extended frequency operating range does allow the freedom for alternative design options. These "fine tuning" options will be evaluated in terms of the system performance impact on both the building load requirements and the product reliability.

2.4 AIR HANDLERS

2.4.1 INTRODUCTION

The air handler effort during this reporting period concentrated on the assembly of both the indoor and outdoor units and on preparing the units for component testing. These efforts included assembly of both the refrigerant and hydronic loops, component assembly, installation and checkout of the required instrumentation, and leak checking of the final assembly. The assembly and the test preparation of both the indoor and outdoor units have been completed and testing is scheduled to begin during the next reporting interim. A description of the above activities and the status of the hardware is presented in the following sections.

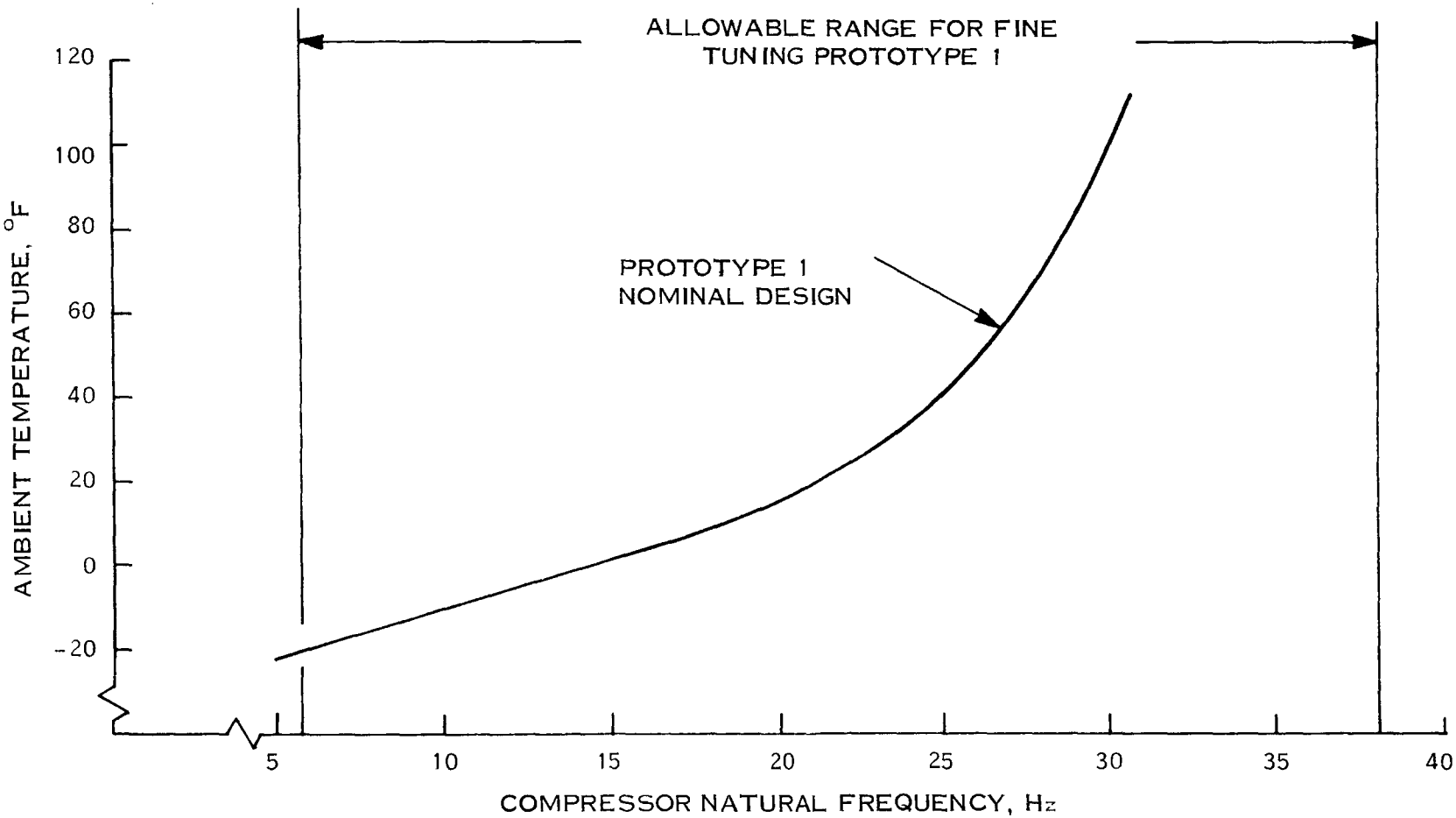


Figure 2-20. Compressor Natural Frequency as a Function of Ambient Temperature

2.4.2 INDOOR UNIT

The air handler indoor unit, which has been assembled, instrumented, and leak checked, is currently being prepared for component testing. The indoor unit is a standard unit fabricated by GE-CACBD, Trenton, N.J. except for the addition of a transition cap which houses the hydronic coil. The unit consists of an "A" coil (refrigerant), a blower, a hydronic coil, and a sheet metal housing. The "A" coil was removed from the unit and was modified by incorporating the following items:

1. Filter/Dryer
2. Helium Bleed Port
3. Pressure Taps
4. Immersion and surface Thermo Couples

The modified "A" coil is shown on Figure 2-21 and 2-22. A thermocouple grid was added to the upstream side of the "A" coil and is shown on Figure 2-22. The "A" coil was, subsequently, re-installed into the indoor unit as shown on Figure 2-23; this figure also shows the blower and the transition cap which houses the auxiliary coolant heat exchanger. Indoor unit instrumentation is shown on the Component Test Schematic illustrated in Figure 2-24.

2.4.3 OUTDOOR UNIT

The air handler outdoor unit was assembled, instrumented and leak checked during the present reporting period, and it, too is currently prepared for component testing. The outdoor unit is schematically shown in Figure 2-25, and the actual hardware is presented in Figure 2-26. This configuration utilizes a compartmentized approach for test flexibility and simplicity and allows excellent accessibility for component adjustment or replacement. It also

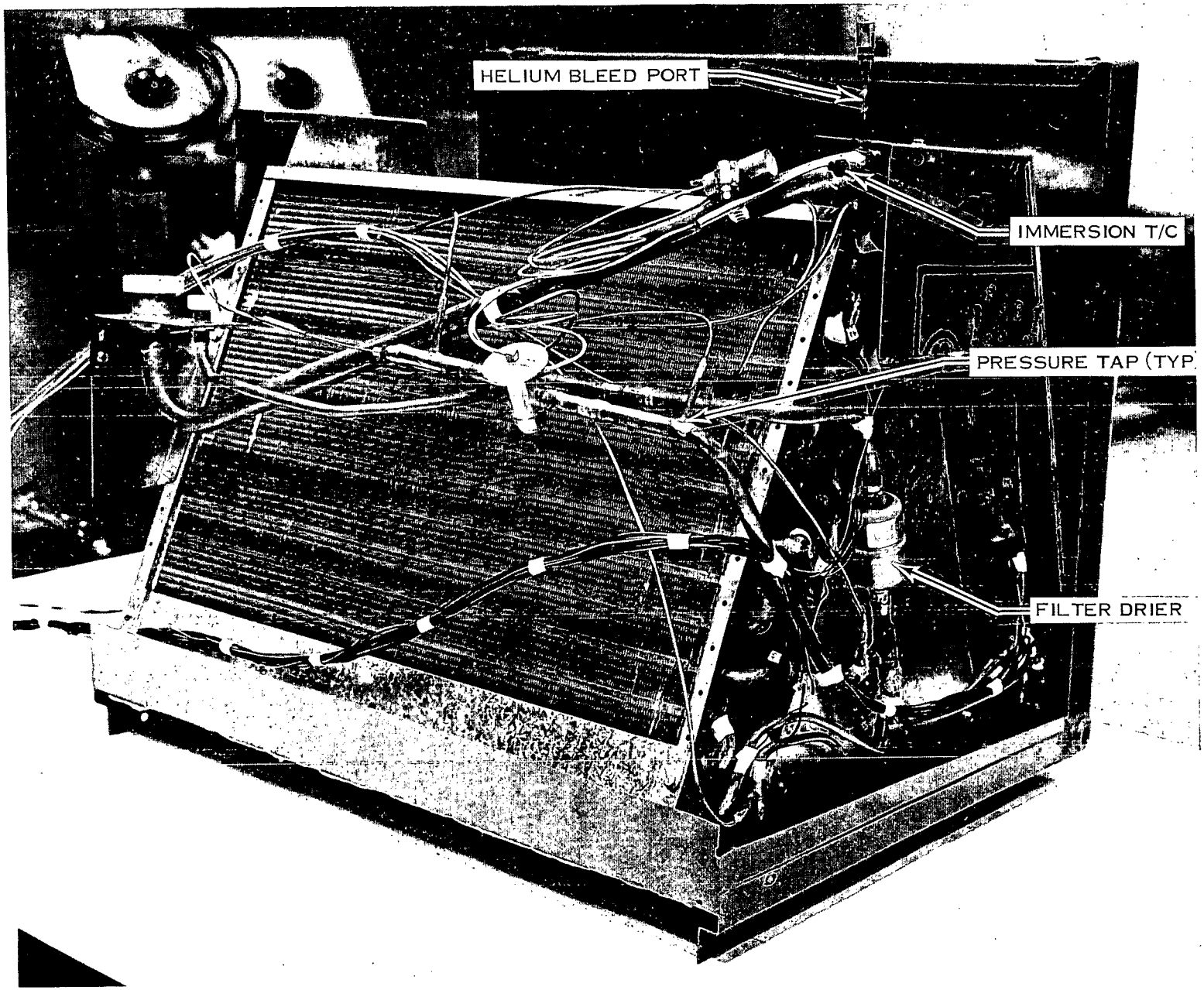


Figure 2-21. Indoor Air Handler "A" Coil and Modifications

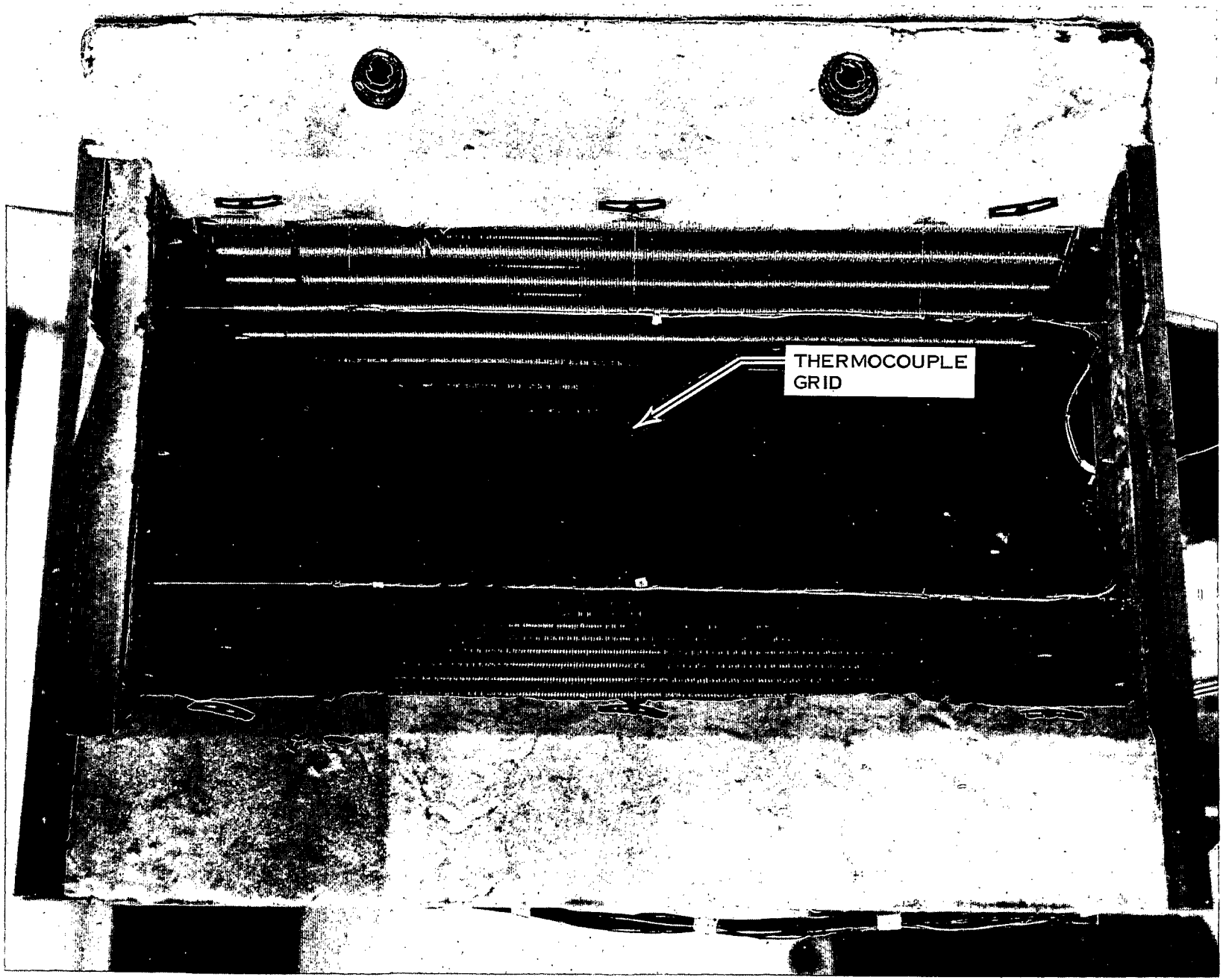


Figure 2-22. Indoor Air Handler "A" Coil Thermocouple Installation

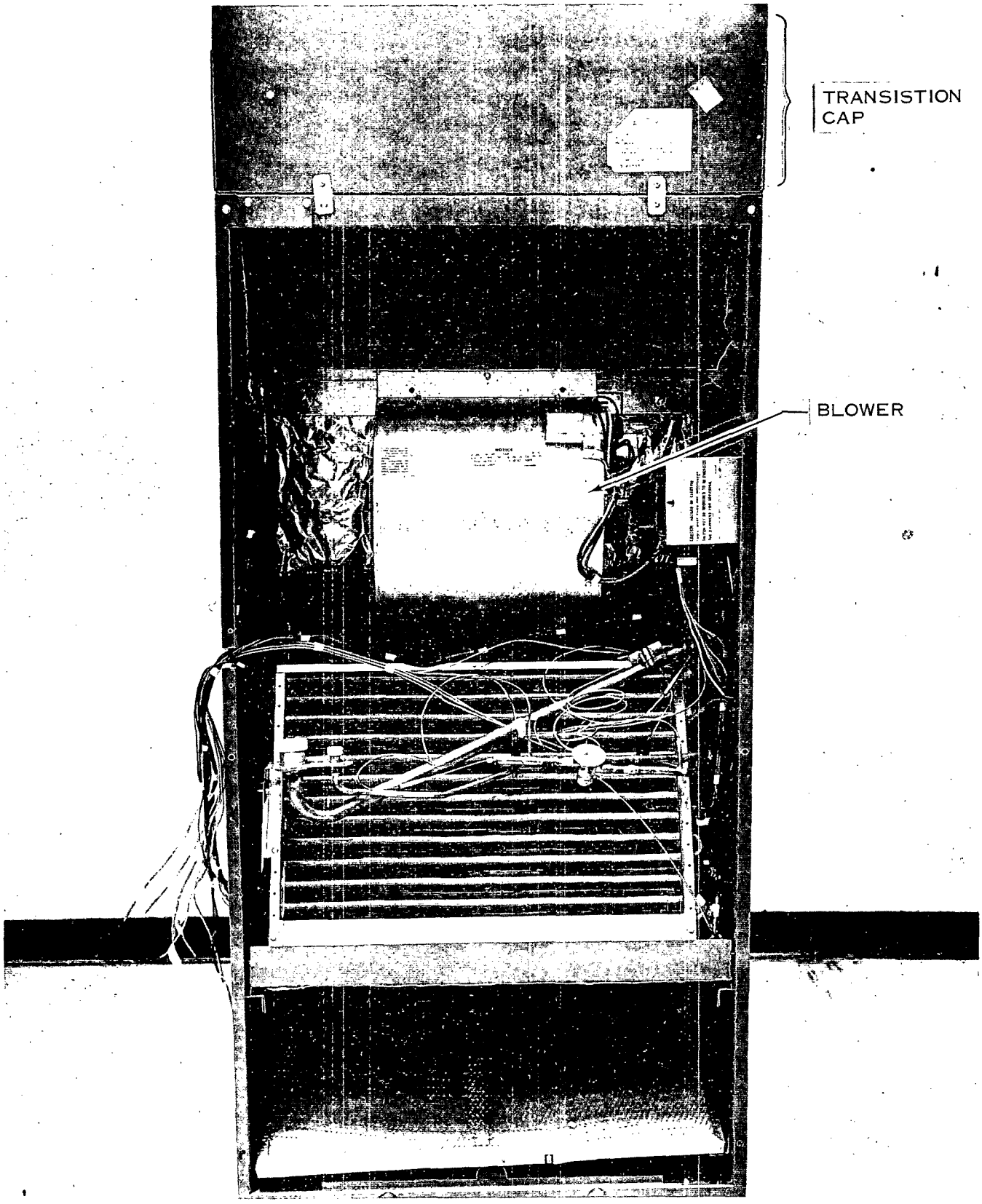
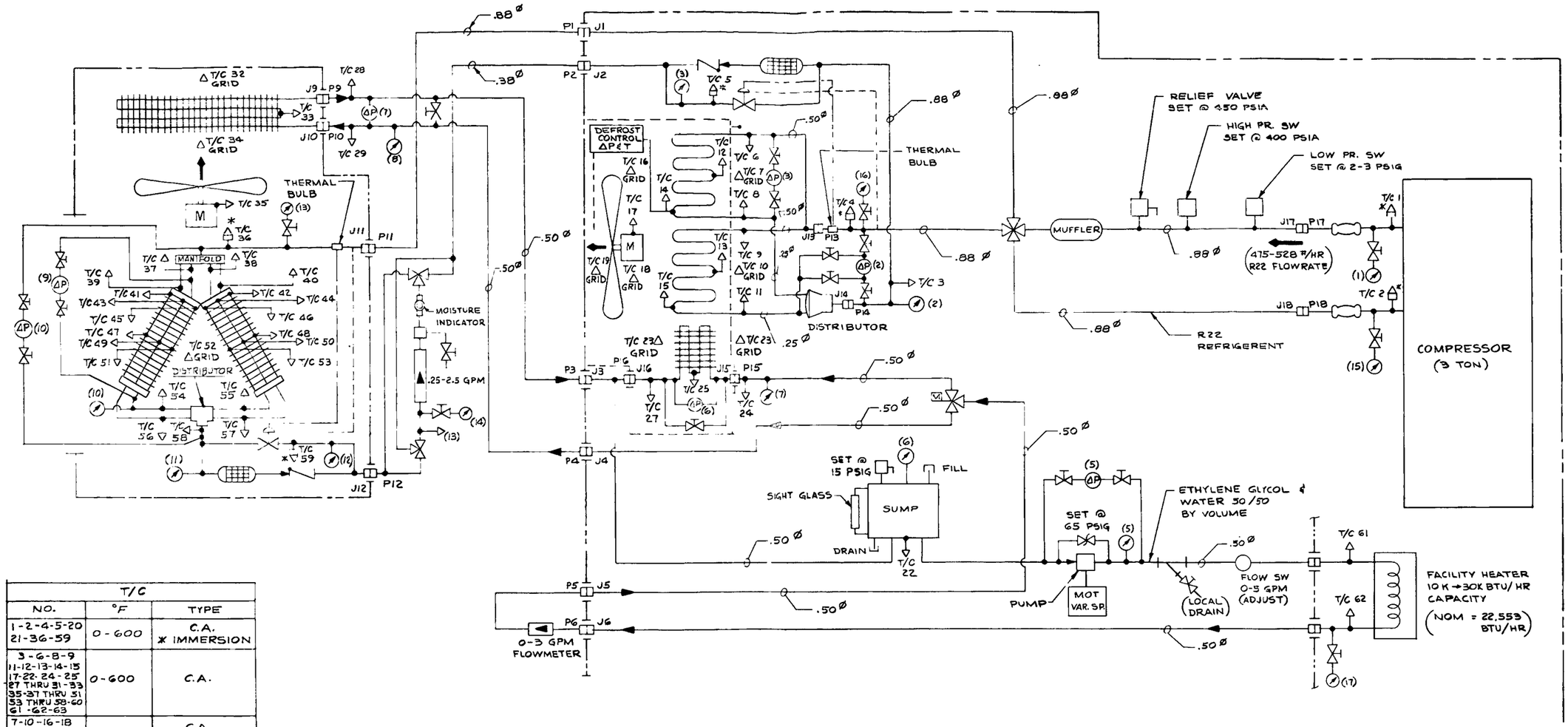


Figure 2-23. Modified Air Handler Assembly



T/C		
NO.	°F	TYPE
1-2-4-5-20 21-36-59	0-600	C.A. * IMMERSION
3-6-8-9 11-12-13-14-15 17-22-24-25 27 THRU 31-33 35-37 THRU 51 53 THRU 58-60 61-62-63	0-600	C.A.
7-10-16-18 19-23-26-32 34-52	0-600	C.A GRID

PRESSURE GAUGE	
NO.	PSIG
1-2-3-10-11-12-13-14-15-16	0-400
4-5-6-7-8-9-17	0-100

ΔP GAUGE		
NO.	PSIG	OPERATING PSIG
2-3-9-10	0-30	400
5	0-100	100
6-7	0-30	100

Figure 2-24. Component Test Schematic

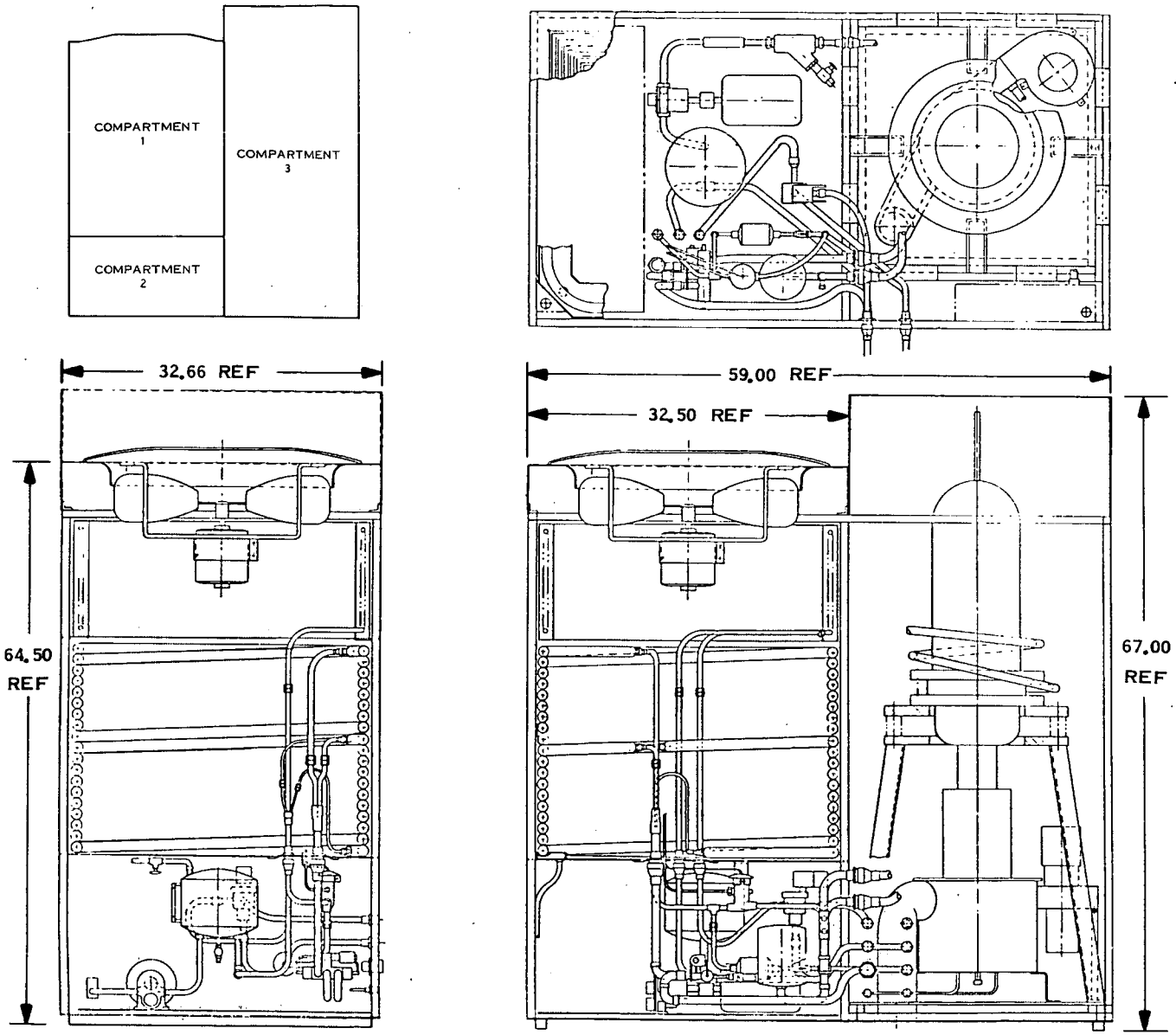


Figure 2-25. Prototype 1 Outdoor Unit Hardware Arrangement

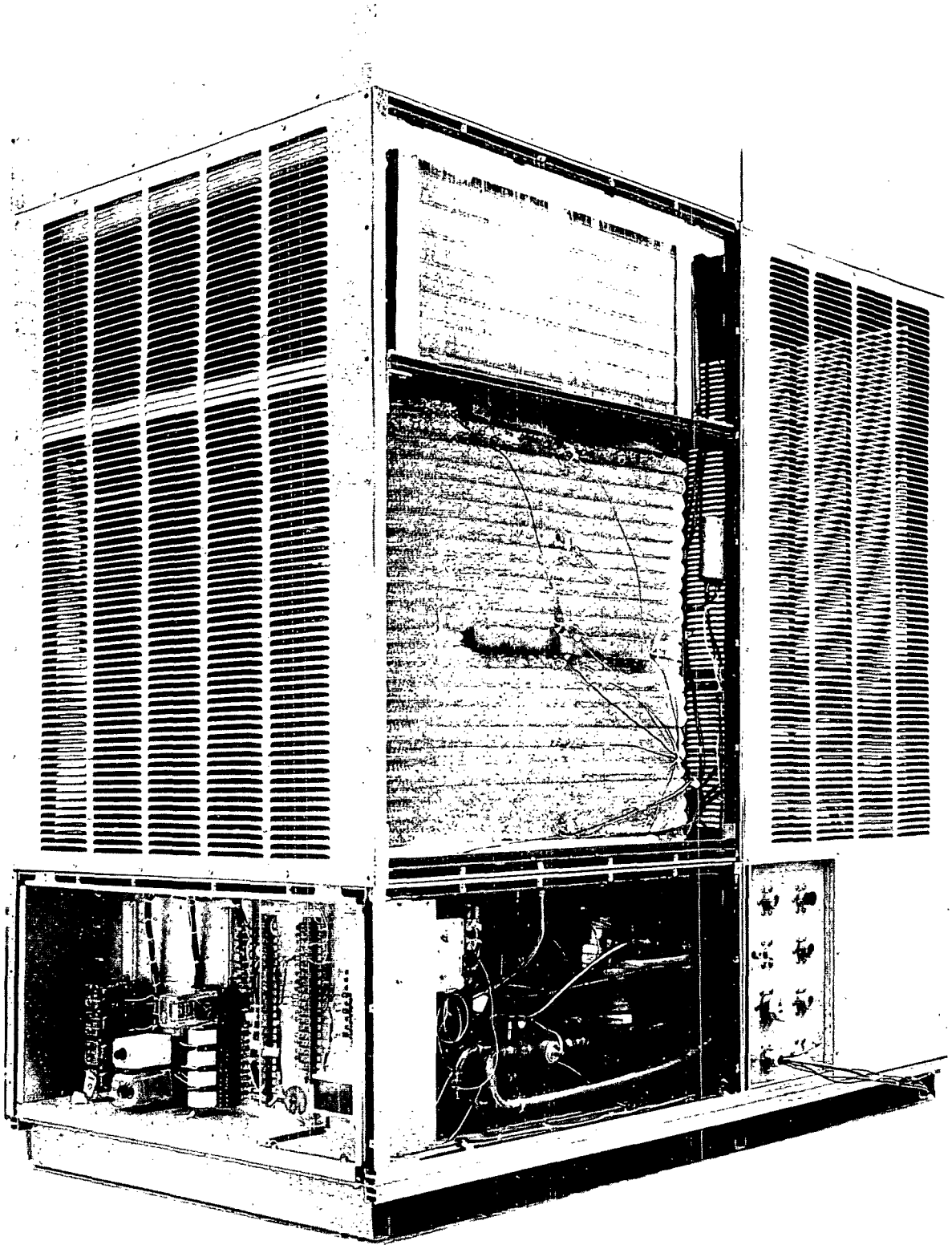


Figure 2-26. Outdoor Air Handler-Prototype 1 Assembly

employs low pressure drop, single bank heat exchangers with an upward discharge, top mounted fan system.

Two additional views of the assembled outdoor unit, less outside panels, are presented in Figures 2-27 and 2-28; the three compartments comprising the outdoor units are easily identified. They are:

1. The combuster/engine/compressor assembly compartment
2. The air handler compartment
3. The refrigeration and hydronic loop components compartment

As shown in Figure 2-28 a conventional, 3 ton electrically driven compressor is presently housed in the combustor/engine/compressor assembly compartment. This conventional compressor will be employed to simulate the Stirling engine/inertia compressor during the air handler component check-out and verification testing.

Details of the outdoor refrigerant and hydronic heat exchanger coils are also shown in Figures 2-27 and 2-28. Two coil sections comprise the refrigerant coil subassembly, one having 11 rows and the other having 12 rows. The refrigerant flow was distributed into two parallel paths in order to reduce the two-phase pressure drop; frontal areas and tubing lengths were selected to provide equal loads to each section. An internal view of the heat exchangers showing the two parallel flow paths is provided Figure 2-29. Also shown in this figure is a helium bleed port at the highest elevation in the refrigerant loops. The fully instrumented heat exchanger coils are shown in Figures 2-27 and 2-28. These thermocouples were installed as specified in the Component Test Schematic, Figure 2-24.

Figure 2-30 presents a view of the third outdoor unit compartment-that which houses the refrigeration and hydronic loop compartments. The piping interfaces

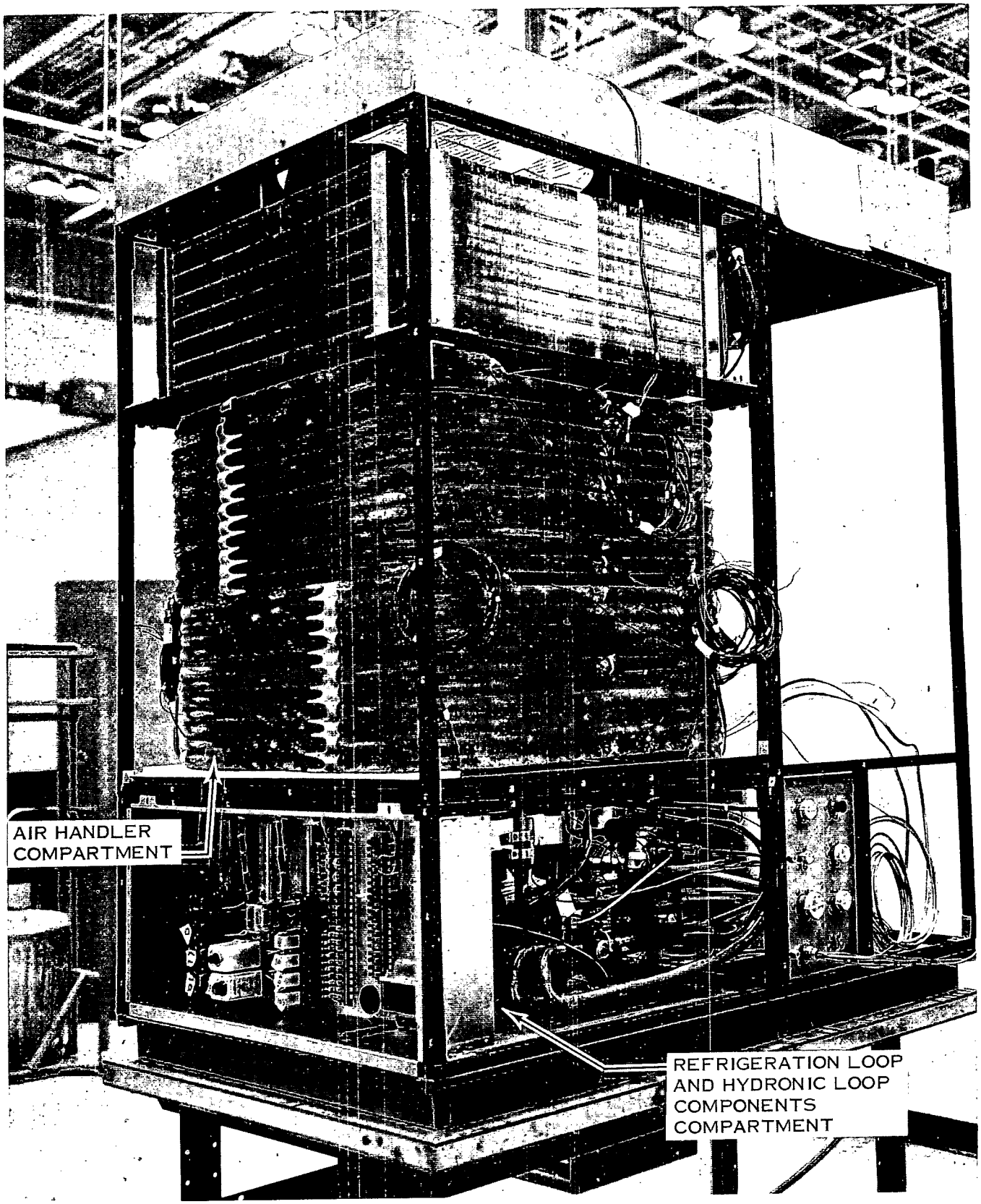


Figure 2-27. Outdoor Air Handler-Prototype 1 Assembly

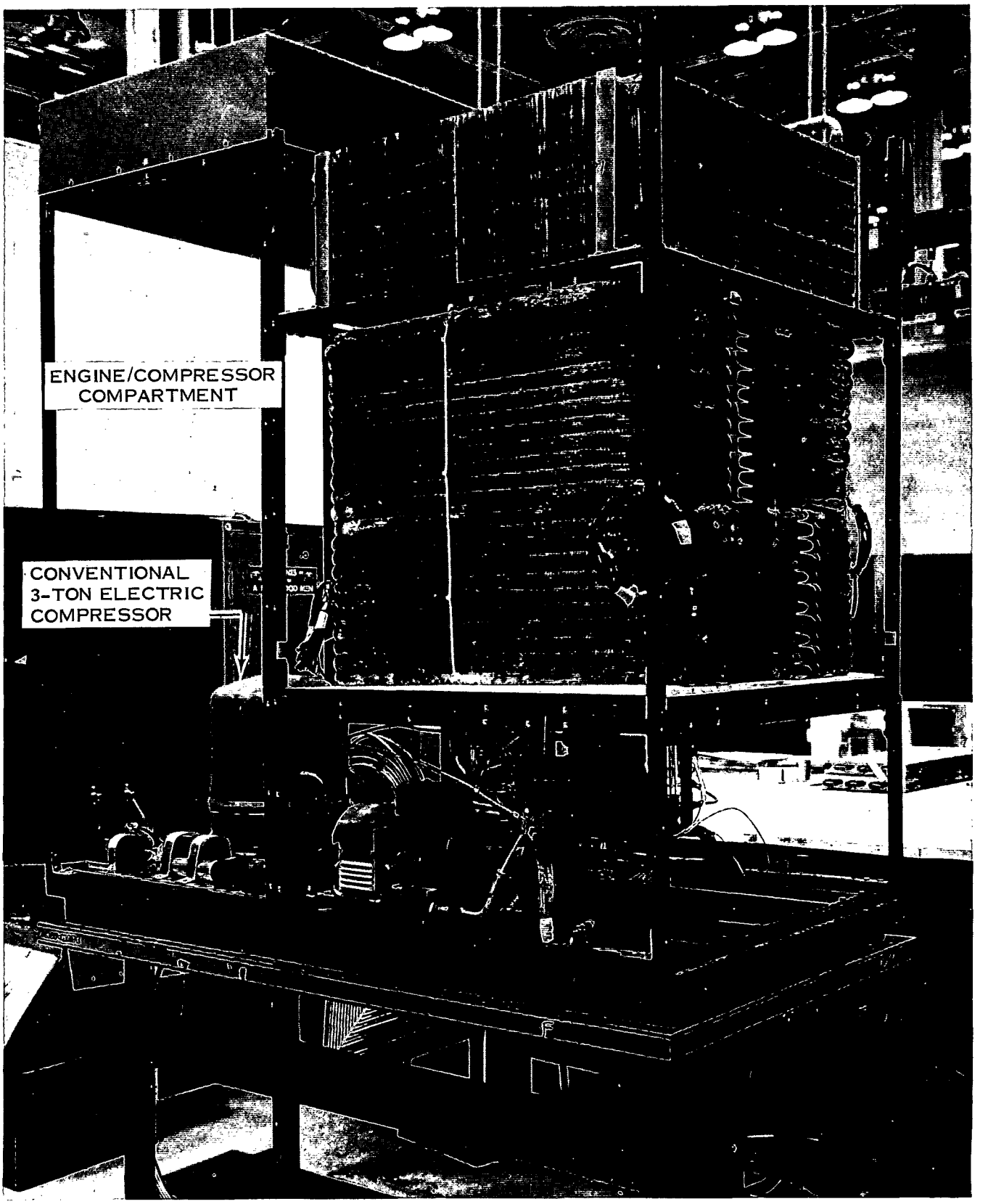


Figure 2-28. Outdoor Air Handler-Prototype 1 Assembly

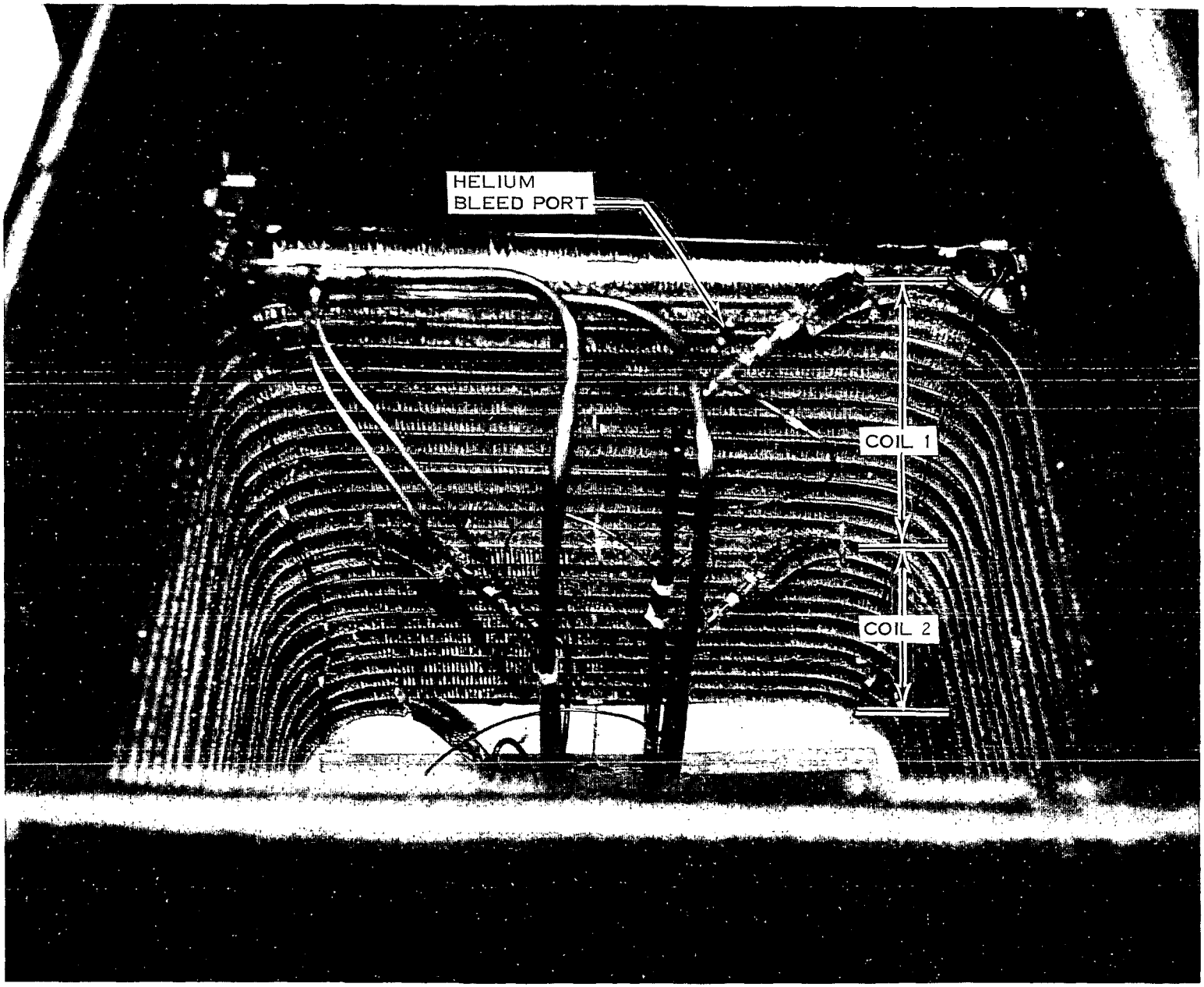


Figure 2-29. Outdoor Unit Refrigerant Heat Exchanger

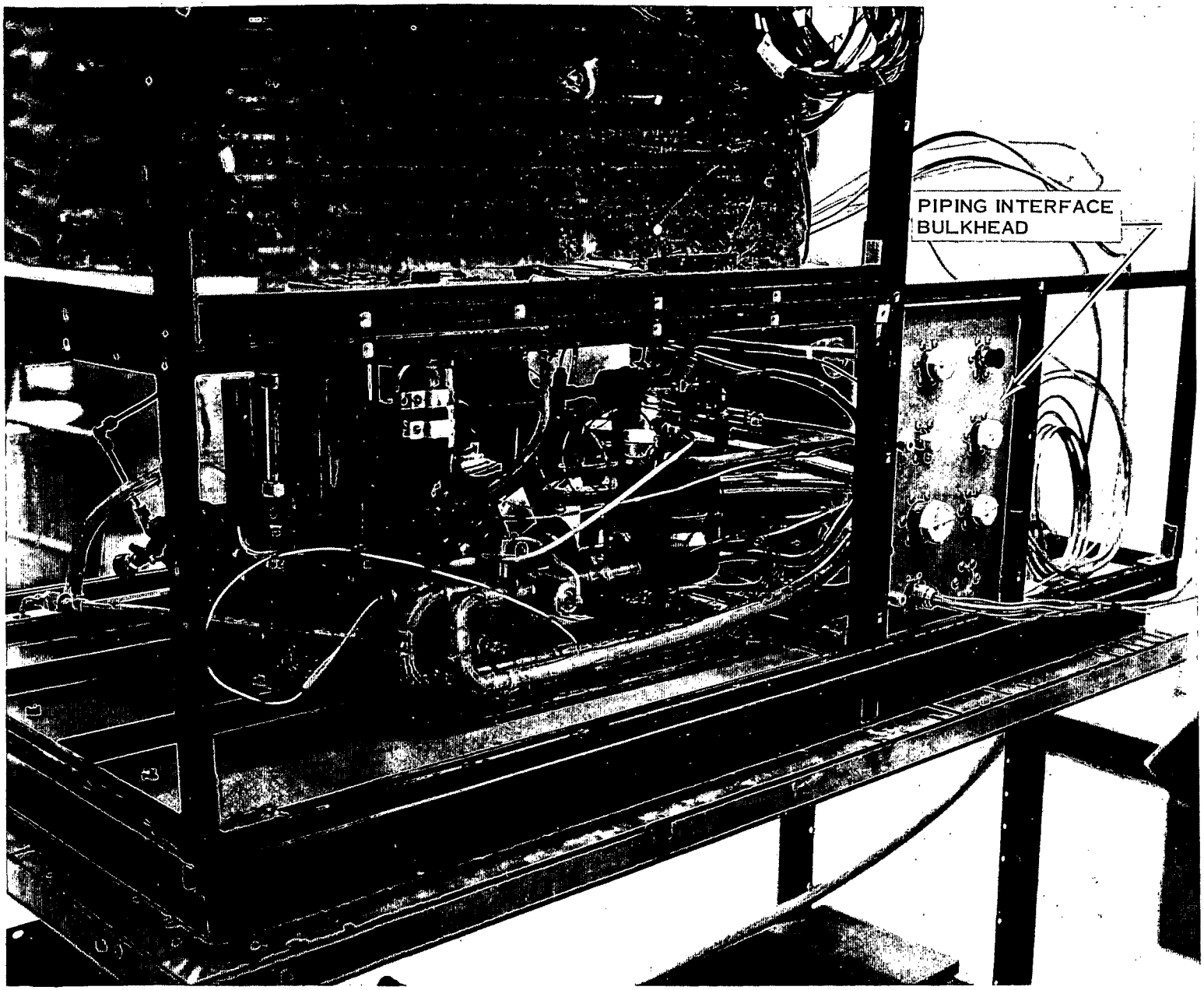


Figure 2-30. Outdoor Unit Interface to Indoor Unit

with the engine/compressor assembly and with the indoor coil are indicated in this figure.

2.5 REFERENCES

1. "Development of a Gas Heat Pump, July through December, 1976," General Electric Document No. 77SDS4219.

2. Keys, W.M. and London, A.L., "Compact Heat Exchangers", McGraw-Hill, Second Edition, 1964.

GENERAL ELECTRIC

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