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THE MATCHING OF A FREE PISTON STIRLING ENGINE COUPLED WITH A FREE PISTON LINEAR COMPRESSOR FOR A HEAT PUMP APPLICATION

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ABSTRACT

Because of its inherently high efficiency, power transmission simplicity, lack of seals, and elimination of oil lubrication, a Free Piston Stirling Engine coupled with a Free Piston Linear Compressor appears to be an attractive candidate for a heat activated engine driven heat pump application. Unlike a conventional kinematic drive system, this mechanical arrangement comprises a thermodynamically coupled three free-body dynamic system, consisting of a working fluid displacer, a power piston, and a compressor piston. The relative motions of the three bodies are constrained only by the working fluid, and they are dependent variable functions of load and operating parameters. This leads to a unique load matching problem over the range of ambient temperature conditions.

This paper discusses the analytical approach for dynamically matching the engine and compressor. Based on this study, a prototype engine/compressor assembly was designed with a flexibility for tuning the dynamic components in order to optimize the performance. Preliminary test data is presented and compared with analytical predictions.

APPLICATION

Conventional heat pumps provide an energy efficient means for effecting space heating and air conditioning. Heat activated heat pumps potentially offer the consumer alternate features which enhance the product's utility. The prime mover waste heat can be recovered and utilized to augment the heat pump heating capacity, and thereby increase the heating coefficient of performance. Because little or no auxiliary heat would be required to meet the building load demand, the product could be especially attractive to regional markets with high heating to cooling ratio requirements.

The General Electric Company, in a cooperative effort with the American Gas Association (AGA), the Department of Energy, and the Gas Research Institute (GRI), is currently developing a heat activated heat pump. A detailed discussion of the heat pump implementation program has been given in a previous paper (Reference 1). The primary prototype product goals include: a cooling mode design point capacity of three tons, a cooling mode COP that is competitive with the performance of an electric air conditioner, and a minimum design life comparable with that of conventional space conditioning equipment. Because of its inherently high efficiency, mechanical simplicity, quiet, clean combustion characteristics, and hermetically sealed load feature, a free piston Stirling engine (Reference 2), driving a free piston linear refrigerant compressor was selected for the prototype heat activated heat pump system prime mover/load configuration. Figure 1 shows a design layout of the engine/compressor system.

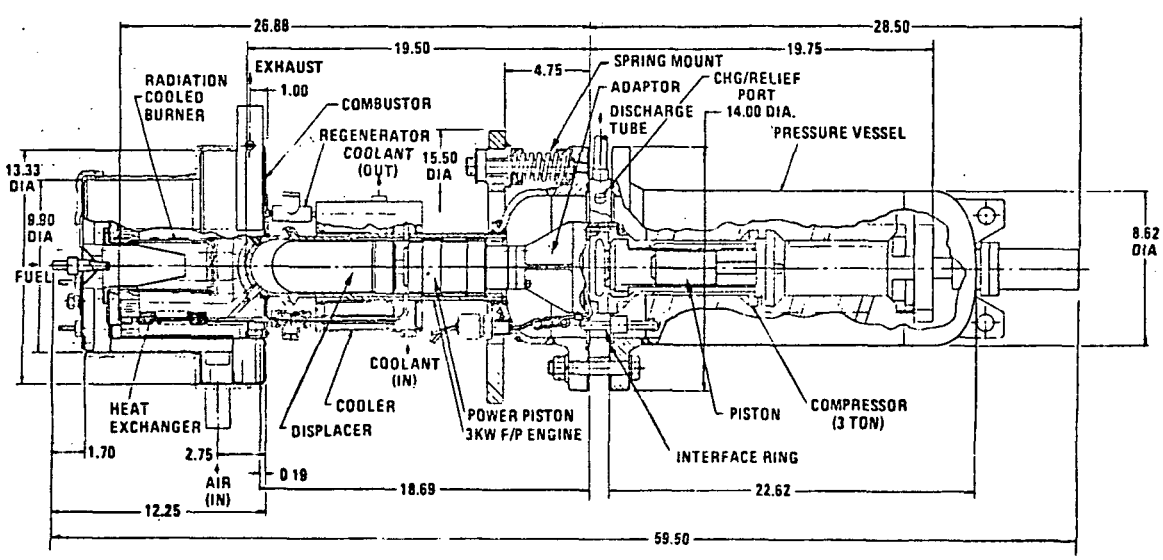

Fig 1 - Combustor/free piston Stirling engine/compressor assembly

ENGINE/COMPRESSOR MATCHING

The proper matching of a prime mover and a load is a design problem common to most power plant system analyses. The matching of a free piston system is characterized by the non-mechanical linkage between dynamic components and, therefore, is uniquely sensitive to load variations. Common positive drive systems can be controlled, for example, by constraining the developed strokes and frequencies. In contrast, a free piston system relies on a proper balance of the thermodynamic and dynamic forces which interact through the three dynamic component motions. These moving elements consist of the compressor inertia mass, the power piston, and the displacer. The movement of the inertia mass relative to the compressor housing is actually an extended member of the power piston upon which the engine pressure forces act. The power piston also seals the oscillating pressure working fluid volume from a nearly constant-pressure bounce space gas volume. It is the power piston motion which oscillates the compressor housing so as to induce a relative motion between the housing and the compressor inertia mass. Finally, the relatively light displacer mass functions in such a fashion so as to periodically shuttle the working space gases through the hot side and cold side engine heat exchangers, thereby phasing the rates at which heat is absorbed and rejected from the engine working fluid. The displacer motion is affected by a gas spring action between the power piston and the displacer rod. Figure 2 illustrates the dynamic component coupling and the main feedback paths through which these component functions are interrelated.

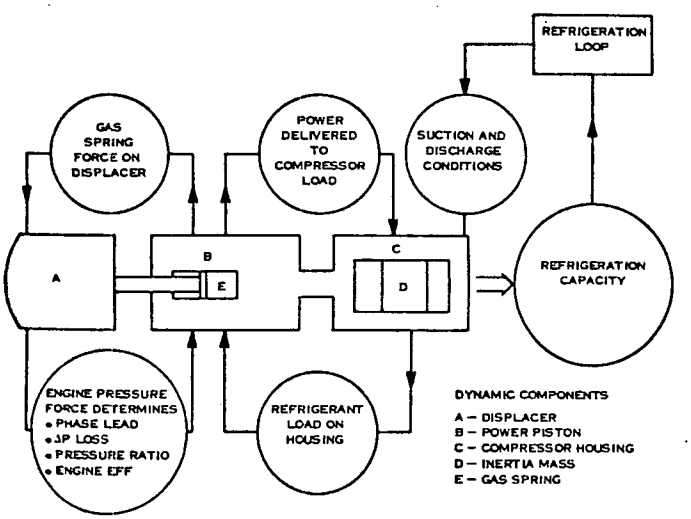

Fig 2 - Compressor-engine coupling

Designing for a proper force balance at a specific load condition is a complex problem in itself. However, adapting the free piston engine and inertia compressor to the heat pump application requires designing a system which would be stable over a range of variations. The anticipated load variation is shown in Figure 3. This plot indicates the range of compression ratios and pressure forces which will be encountered by the free piston compressor. The refrigerant compressor spring rates were derived from compressor natural frequency test

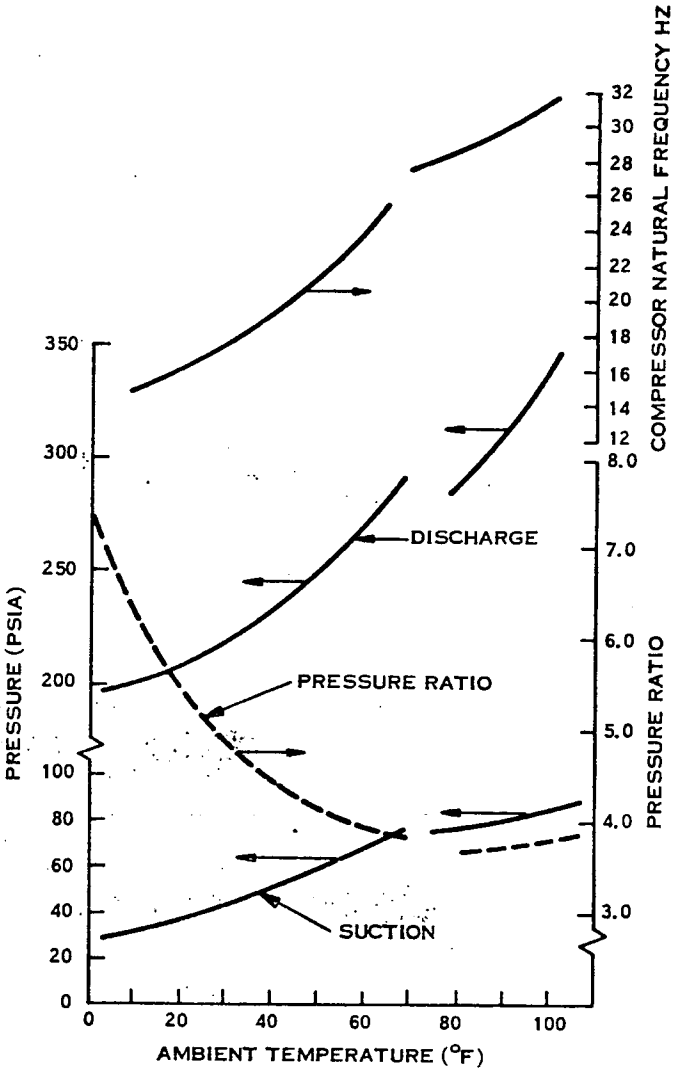

Fig 3 - Compressor pressure characteristics

data. These frequencies, which were determined as a function of the load characteristics during the testing of an engineering inertia compressor prototype, are also included in the figure.

DYNAMIC ANALYSIS

The goal of the engine design task was to identify parameters which establish the required periodic motion of the dynamic components over the load range. To this end, a simplified free piston engine/compressor model was examined in terms of a frequency response analysis. The analysis is an extension of Rauch's two-mass model (Reference 3) modified to reflect a three-mass dynamic system. The basic model, which is shown in Figure 4(b) employs linear springs and dampers to model the compression, expansion, and the isothermal heat exchanger pressure drop as schematically shown in Figure 4(a). Those parameters that are descriptive of the modeled spring rates are then, in effect, the design terms which directly determine the resultant engine/compressor dynamics. These terms were identified as the tuning parameters and are listed in Table 1.

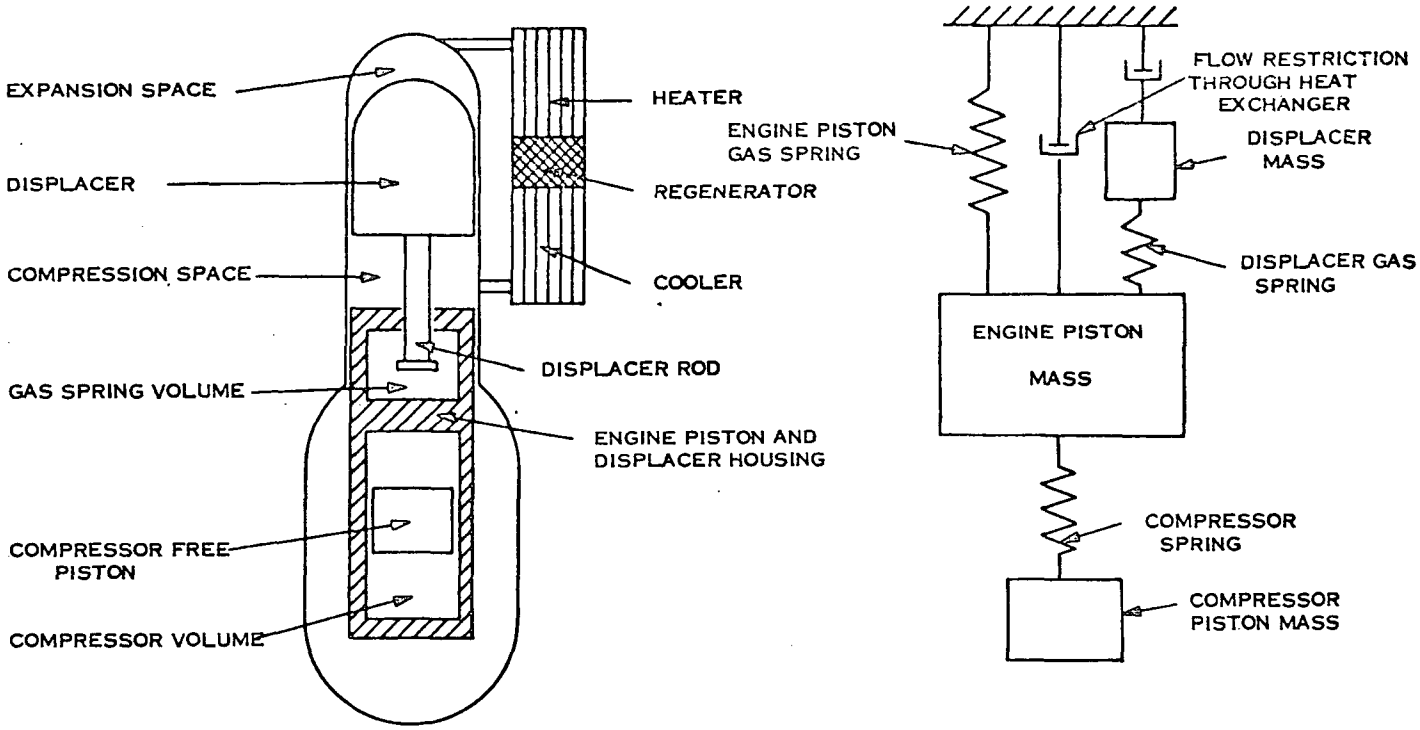

Fig 4 - Three-body dynamic model simulating a free piston Stirling engine/free piston compressor

Table 1 - Engine Tuning Parameters

ENGINE CHARGE PRESSURE	DISPLACER ROD DIAMETER
ENGINE PISTON MASS	DISPLACER MASS
ENGINE TOTAL VOLUME	DISPLACER GAS SPRING VOLUME
ENGINE BORE SIZE	COMPRESSOR INERTIA MASS
	COMPRESSOR BORE DIAMETER

The frequency response characteristics of the engine/compressor model were studied in order to determine the operating characteristics for a given design and a given refrigerant loading. Typical results are shown for two different compressor loadings in Figure 5. A maximum energy analysis dictates that the engine/compressor system will operate near the resonance frequency of the engine piston. Therefore, the compressor loading typified in Figure 5a will result in a first harmonic engine operation, while Figure 5b indicates a compressor loading which would induce a dominant second harmonic engine operation at a higher frequency. The calculations of the dynamic parameters associated with the resonance point analytically determine the engine and the displacer strokes, the operating frequency, and the displacer/piston phase angle. From these values, the power output and efficiency can be approximated with close agreement to a thermodynamic model computer analysis for the same design configuration.

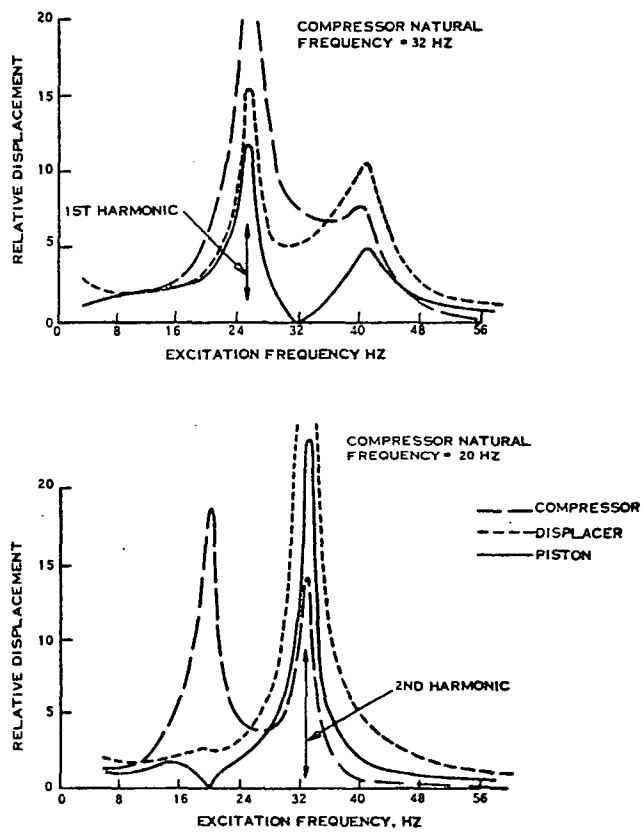

Fig 5 - Frequency response characteristics

DYNAMIC ANALYSIS RESULTS

Many configurations were simulated with this type of an approach, and the results were evaluated. The data was catalogued in the type of plot format shown in Figure 6. This particular plot indicates the variation in the engine piston stroke and the displacer-piston phase angle for a wide range of compressor load natural frequencies and for several displacer gas spring volume sizes. All of the other engine parameters were held constant. However, parametric variations in compon-

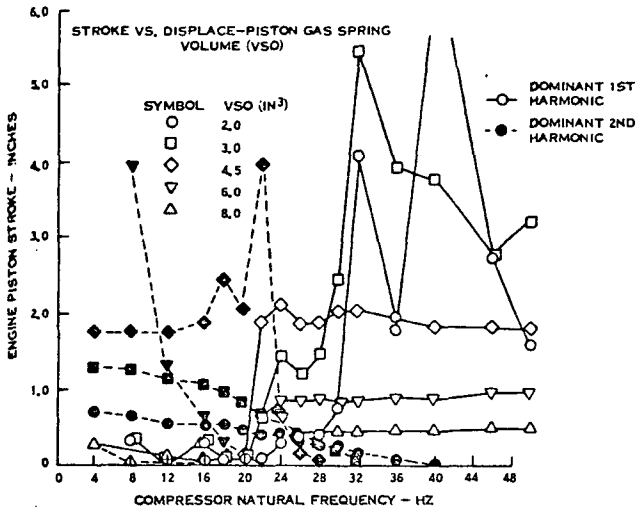

Fig 6(a) - Engine stroke versus displacer gas spring volume and compressor natural frequency

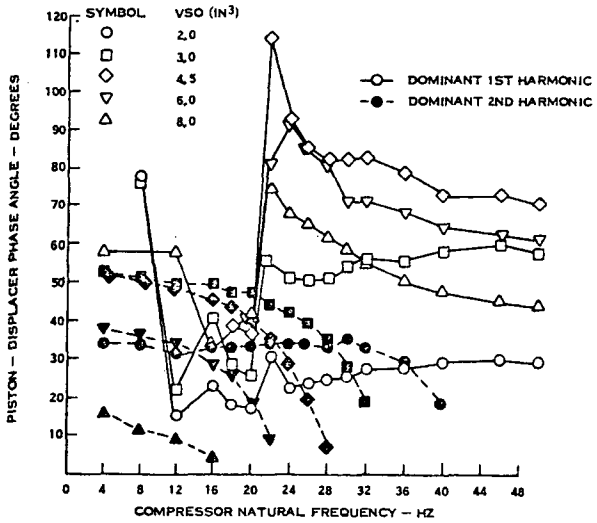

Fig 6(b) - Phase angle versus displacer piston gas spring volume and compressor natural frequency

ent stroke, frequency, and phase angle were studied for all of the design variables listed in Table 1. This study leads to the identification of those parameters which could most readily effect a favorable dynamic response.

A most dramatic change in engine piston stroke and phase angle can be affected simply by varying the displacer rod diameter and the gas spring volume. Figure 7 illustrates a typical performance-configuration map generated from the dynamic analysis. Lines representing constant displacer rod diameter and constant gas-spring volume are cross-plotted while all other engine design parameters are held constant. The importance of the proper selection of the displacer gas spring configuration, in order to obtain desirable engine stroke and phase angle, is clearly illustrated in this figure. It also highlights the effectiveness of the dynamic analysis as a valuable preliminary design tool for free piston machines.

With this information a free piston engine design was configured to meet the system constraints for power, efficiency, and load range acceptability. The descriptive design parameters for the prototype engine/compressor hardware are listed in Table 2.

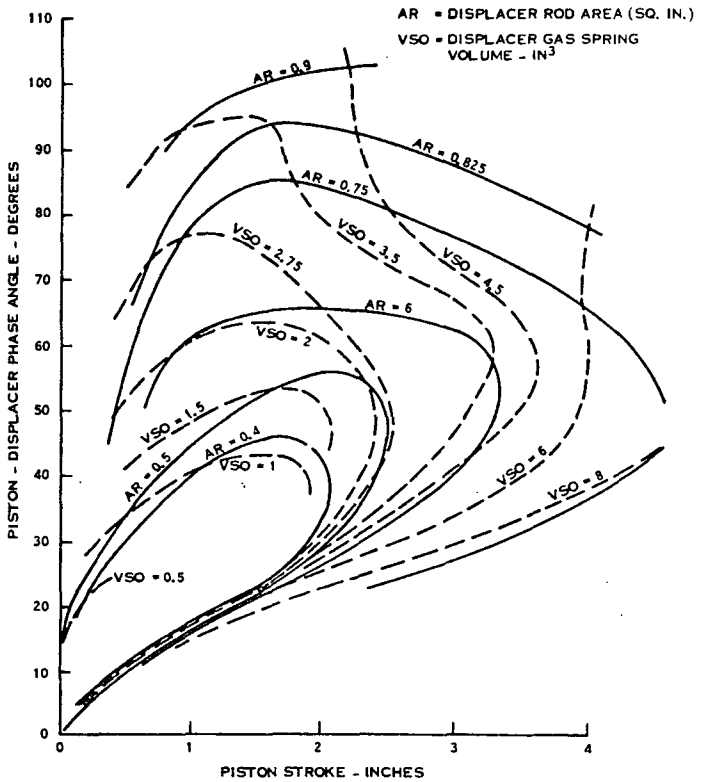

Fig 7 - Performance configuration map

Table 2 - Free Piston Stirling Engine Design Parameters

PARAMETER	NOMINAL
CHARGE PRESSURE (BAR)	60
BORE (DIAMETER) (INCHES)	2.986
HEATER HEAD VOLUME (INCHES)	4.256
REGENERATOR VOLUME (IN^3)	5.54
COOLER VOLUME (IN^3)	1.36
DISPLACER ROD DIAMETER (INCHES)	1.07
DISPLACER/PISTON GAS SPRING VOLUME (IN^3)	4.5
DISPLACER MASS (LB)	2.08
POWER PISTON AND COMPRESSOR HOUSING MASS (LB)	22.5
COMPRESSOR PISTON MASS (LB)	6
GAS SPRING REFERENCE PORT CLEARANCE (INCHES)	0.9

THERMODYNAMIC ANALYSIS

Although the frequency response analysis served as a convenient method for determining the system dynamics, further analytical efforts were required in order to specify the operational power output and the engine efficiency at compressor loadings other than the 95°F day cooling mode design point. To this end, a transient heat transfer thermodynamic analysis similar to Finkelstein's (Reference 4) was used to simulate the system performance for various hot and cold side engine temp-

eratures as well as for a range of ambient temperature dependent refrigerant loads. A good agreement was obtained between the system dynamics, as prescribed by the complex thermodynamic analysis, and the simple spring/mass frequency response model. These results are illustrated in Figure 8. The plots show the thermodynamic model simulation of the high frequency engine operation in response to the heating mode low compressor natural frequency loads. This phenomenon analytically verifies the dominant second harmonic operation that was predicted by the three-free-body dynamic model.

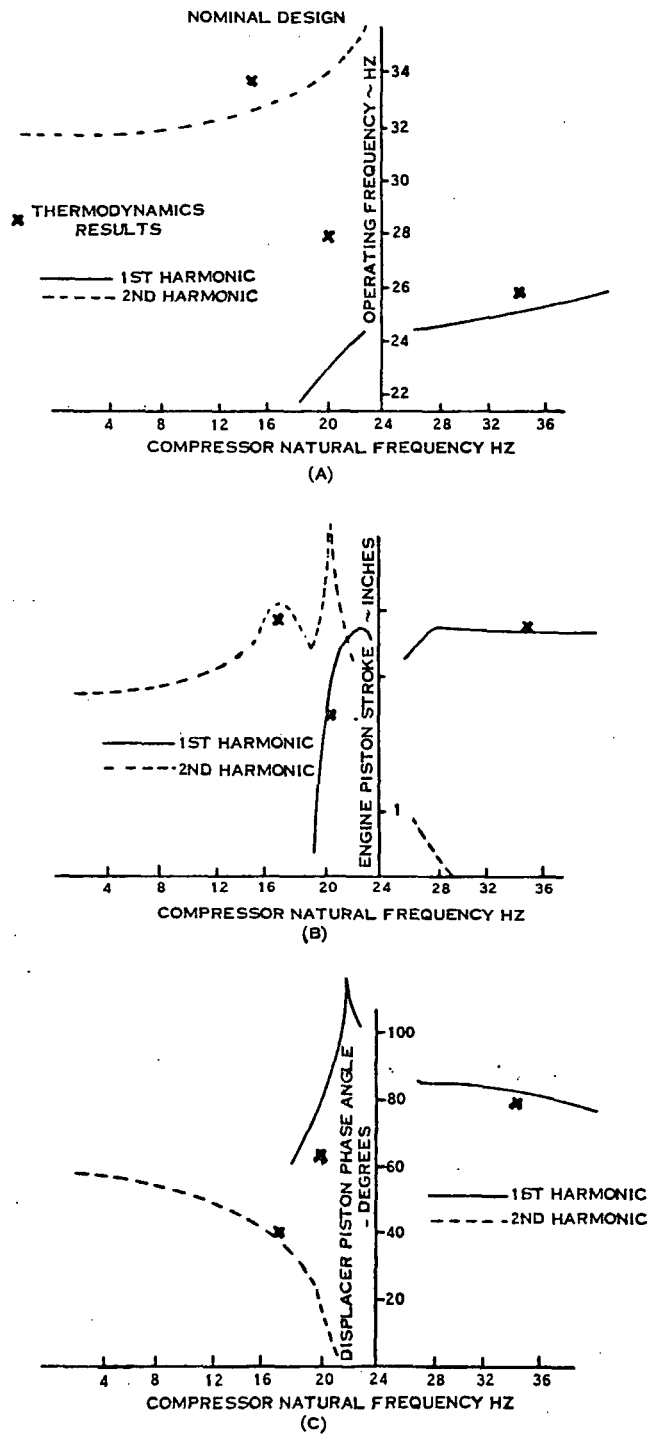

Fig 8 - Comparison of dynamic versus thermodynamic prediction model results

OFF-DESIGN ANALYSIS

The thermodynamic engine performance analysis included a simulation of the system operation with compressor suction and discharge refrigerant pressure which are typical of 60°F ambient temperature day. This load condition established a compressor natural frequency at which neither the first nor the second engine harmonic resonance is dominant. At this loading, a steady engine performance simulation was not established. Rather, an operational "hunting" between the first harmonic and the second harmonic characteristics resulted.

These simulation results cast some doubt on the desirability of operating the free piston engine in a harmonic "hunting zone." Therefore, a free piston compressor design modification was implemented to maintain the compressor natural frequency at a sufficiently high value across the entire load range. This modification should ensure that the engine system will operate outside the "hunting zone." Component testing of the modified design compressor is currently in progress.

TEST DATA

Prior to the actual integration of the free piston engine with the free piston compressor, a series of engine component tests were completed. A helium pump load mechanism was used to absorb the developed engine power. Unlike the free piston compressor inertia mass, the helium pump plunger was hard mounted to the engine power piston. With this configuration, the load compression and expansion processes were exactly in phase with the power piston motion. This type of a load mechanism is analagous to a free piston compressor with an infinite inertia mass and a compressor natural frequency equal to zero.

From the engine test performance results it was concluded that by decreasing the displacer rod diameter a better piston displacer phasing and higher engine efficiency could be established with the free piston compressor load. The displacer rod diameter was reduced from 1.25 inches to 1.0 inches prior to the integrated engine/compressor testing. The standard test instrumentation includes linear voltage differential transducers (LVDT's) which are used to measure the engine power piston and displacer motions as a function of time. From these measurements the displacer lead can be determined. Figure 9(b) compares the phase angles that were recorded for both the engine component and integrated engine/compressor tests with the predicted phase angles that were established from the parametric study. The correlation between the test data and the theoretical predications is quite accurate. A similar result is shown in terms of the engine operating frequency in Figure 9a. Several more engine-displacer configurations are scheduled to be tested in order to verify the analytical parameter optimization techniques.

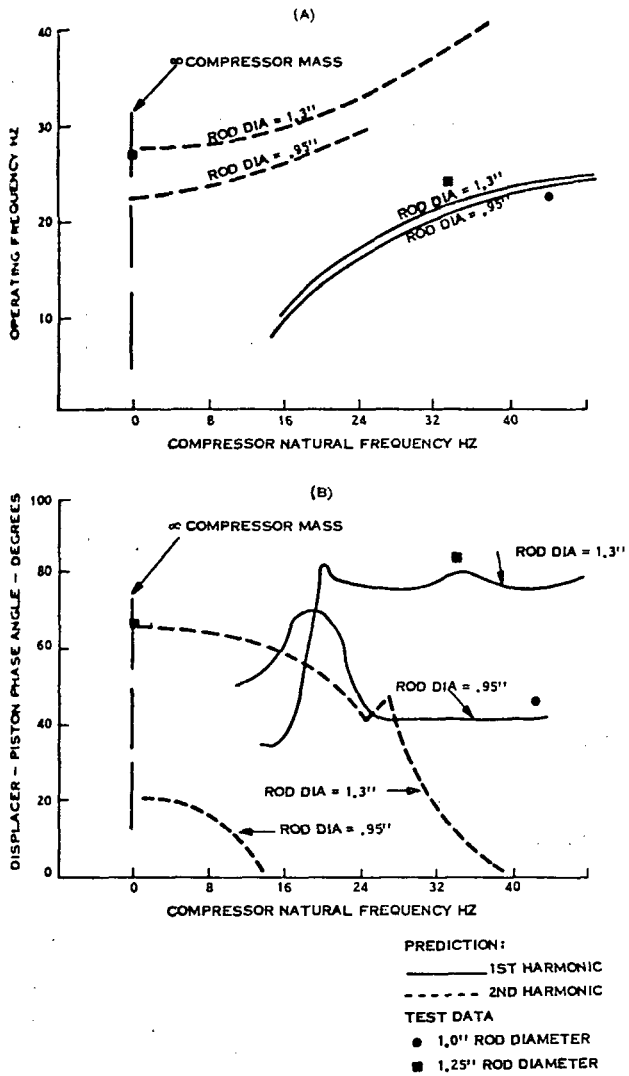

Fig 9 - Test data versus analytical predictions

CONCLUSION AND SUMMARY

The performance and the dynamic characteristics of the free piston Stirling engine/free piston compressor are sensitive to many design parameters. This sensitivity is a critical design concern since the free piston compressor load spring rate varies appreciably as a function of the ambient temperature. A linearized spring/mass dynamic model had been used as a tool to assess the parametric trade-offs during the early engine design phase. Later, a thermodynamic analysis verified the results of this first order approach for describing the component dynamics. A good correlation has been obtained between the preliminary test data and both the dynamic and thermodynamic predictions. Currently, these analytical tools are being extensively employed in an effort to further develop the hardware performance.

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