PERFORMANCE OF A FREE-PISTON STIRLING ENGINE FOR A HEAT PUMP APPLICATION

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ABSTRACT

The General Electric Company, in a cooperative effort with the Gas Research Institute and the Department of Energy, is currently developing a heat activated heat pump (HAHP). A free-piston Stirling engine coupled with a free-piston linear compressor has been selected as the most promising system for the eventual commercial product. The system presents a unique load matching problem over the heat pump's ambient temperature operating range. The system is a thermodynamically coupled, three-body machine consisting of a working fluid displacer, a power piston and a compressor piston. Unlike a conventional kinematic drive system, the motions of the dynamic components are constrained only by the working fluids. This paper presents the test results of the first prototype engine. The load matching characteristics of the engine and the approaches which have been employed to optimize the performance are discussed. It is shown that an engine thermal efficiency of over 30% can be achieved by proper load matching and by reducing heat losses and working fluid leakage.

INTRODUCTION

IN 1975, THE General Electric Company embarked with the American Gas Association on a program to develop a heat-activated heat pump. The Department of Energy joined the developmental effort in 1976. In 1977, the Gas Research Institute assumed program responsibility for the AGA, forming the current DoE/GRI/GE program team. The objective of the program is to develop a heat-activated heat pump product into a viable business venture. With the advantage of using the system waste heat to augment the heating capacity, such a product would not only conserve energy resources, but also would reduce the operating cost to the customer for the space conditioning function. A detailed system and tradeoff study conducted in the early phase of the program was presented in a previous paper (*). The primary product design goals include: a cooling mode design point capacity of three tons and a cooling mode COP which is competitive with that of an electric air conditioner.

A free-piston Stirling engine driving a free-piston linear compressor was selected for its high efficiency potential, mechanical simplicity, low noise and hermetically-sealed features. Analyses discussing the matching characteristics between the engine and the load, as well as a description of the hardware designs, were presented in previous papers (2 and 3). The testing of the first prototype gas heat pump system was completed in 1978; the development of a second prototype is currently underway. This paper presents the test results of the first prototype, free-piston, Stirling engine. The operating characteristics and the energy balance of the engine will be discussed.

System performance of the heat activated heat pump will be presented in a separate paper (4).

System and Component Description

Fig. 1 depicts a design layout of the engine/compressor system. The single cylinder engine is basically a thermally driven oscillator operating on a Stirling cycle. Its housing assembly is the pressure vessel member and contains the heater head, regenerators and cooler subassemblies. The engine/compressor system consists of three freely moving masses: a working fluid displacer, an engine power piston and a compressor piston. The compressor housing is an integral part of the engine power piston. Relative motions of the three free bodies are constrained only by the working fluid pressures, and are dependent on load and operating parameters.

The heat input to the engine is supplied from a gas-fired combustor consisting of a radiation-cooled transpiration burner. The oscillating motion of the engine displacer shuttles the working fluid back and

* Numbers in parentheses designate References at end of paper

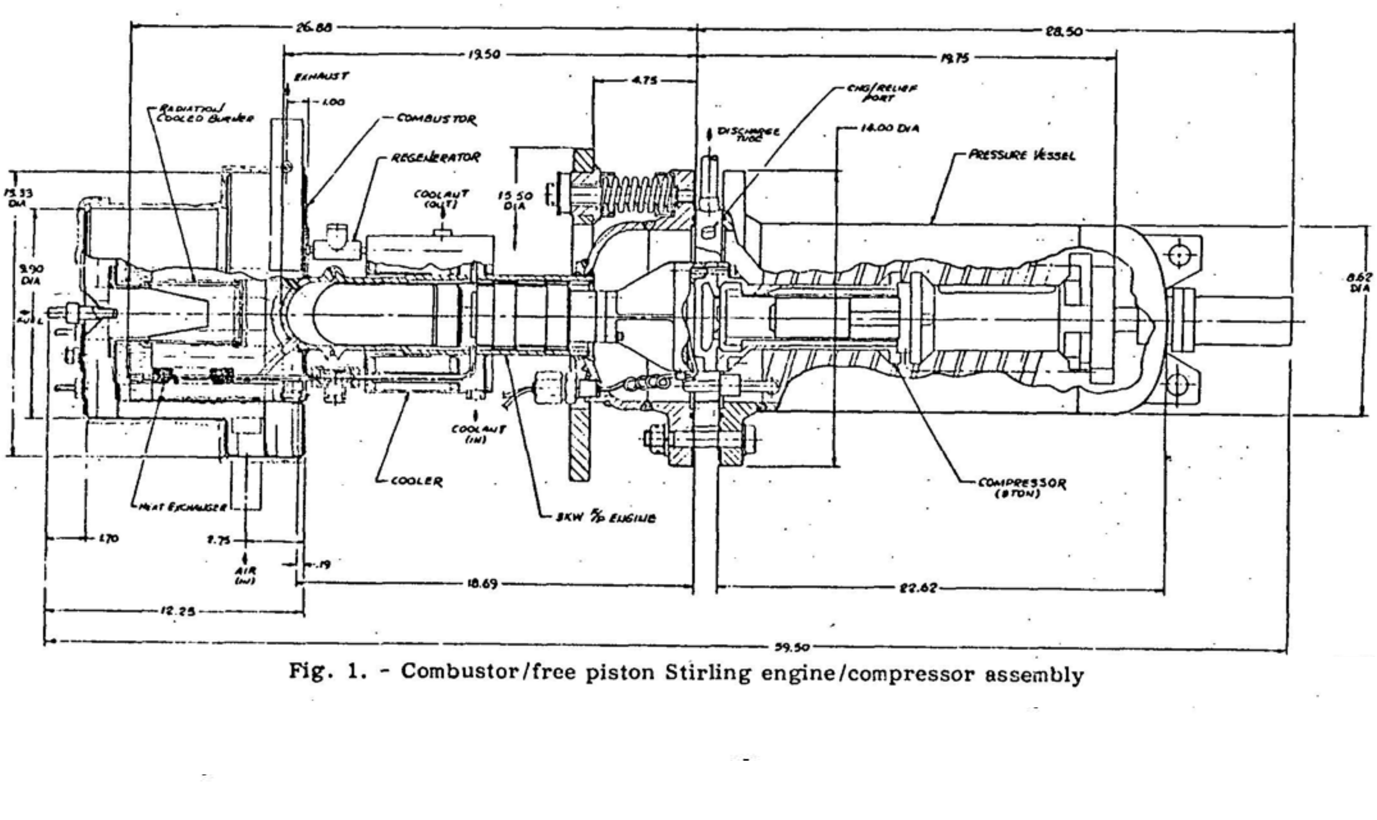


Fig. 1. - Combustor/free piston Stirling engine/compressor assembly

forth through the heat exchangers, thereby phasing the rates at which heat is absorbed and rejected from the engine working fluid. This motion is affected by a gas spring action between the power piston and the displacer rod. Fig. 2 schematically illustrates the dynamic component coupling and the main feedback paths through which these component functions are interrelated.

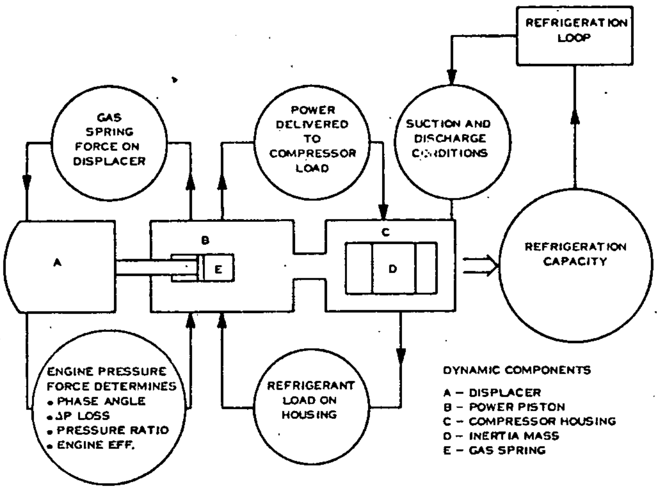


Fig. 2. - Compressor-engine coupling

The first prototype engine was designed with provisions for varying key engine parameters such as displacer gas-spring configuration and regenerator matrix design. This flexibility allows the engine to be optimized experimentally. Design parameters for the free piston Stirling engine are shown in Table 1.

Table 1 Free Piston Stirling Engine Design Parameters

Working Fluid	Helium
Charge Pressure (psia)	870/1160
Power Level (kW)	3.0
Design Efficiency (%)	30
Heater Head Temperature (°F)	1200
Coolant Inlet Temperature (°F)	140
Power Piston Stroke (in)	1.7
Frequency (Hz)	26
Bore Diameter (in.)	2.986
HX Void Volume (in^3)	11.156
Displacer Mass (lb)	2.2
Power Piston and Compressor Housing Mass (lb)	29.

Engine Test

The testing of the free-piston Stirling engine was performed in two stages: first coupling the engine with a helium pump and then with a free-piston linear compressor as the load absorbing devices. Operation with the helium pump as a load absorber yields a two free-body dynamic system. These initial tests reduced the additional controls necessary for the more complex three-mass system using the linear compressor. A force transducer was used for engine power measurement. After the engine operation was characterized through the above test, the engine was tested with the linear compressor as in a heat pump configuration. The engine power output was measured as the total enthalpy gain of the refrigerant across the compressor. Table 2 shows the range of test conditions of the first prototype engine.

Table 2 FPSE Test Conditions

Charge Pressure (psia)	600-1160
Firing Rate (KBTUH)	20-65
Frequency (Hz)	18-26
Power Piston Stroke (in.)	0.9-1.8
Displacer/Piston Phase Angle (Degree)	30-100

Engine Design Modifications

Initial testing of the free-piston Stirling engine revealed that several areas of design improvements were needed in order to obtain satisfactory engine operation. One important area was the reduction of working fluid leakages.

The piston sealing was initially intended to be accomplished by tight tolerance of the bearing sleeve. Initial test results indicated a lower engine power output than expected. Energy balance calculations showed that a significant quantity of energy was lost as the working fluid leaked around the engine power piston to the bounce space. Fig. 3(a) depicts the energy balance of a typical test run using a carbon-graphite bearing sleeve which resulted in significant leakage. A design change to incorporate a positive seal in the bearing sleeve substantially reduced the leakage and improved engine efficiency. The energy balance of the improved test is shown in Fig. 3(b).

The energy balance as shown in Fig. 3 still indicated a greater heat loss to the engine coolant than predicted. This was caused by excessive heat leakage from the hot to the cold working spaces, primarily by the flow leakage across the displacer sleeve. A positive seal was also incorporated on the displacer sleeve. Subsequent testing showed that the heat loss to the engine coolant was reduced to the level as predicted from the earlier analysis; these results are shown in Fig. 4.

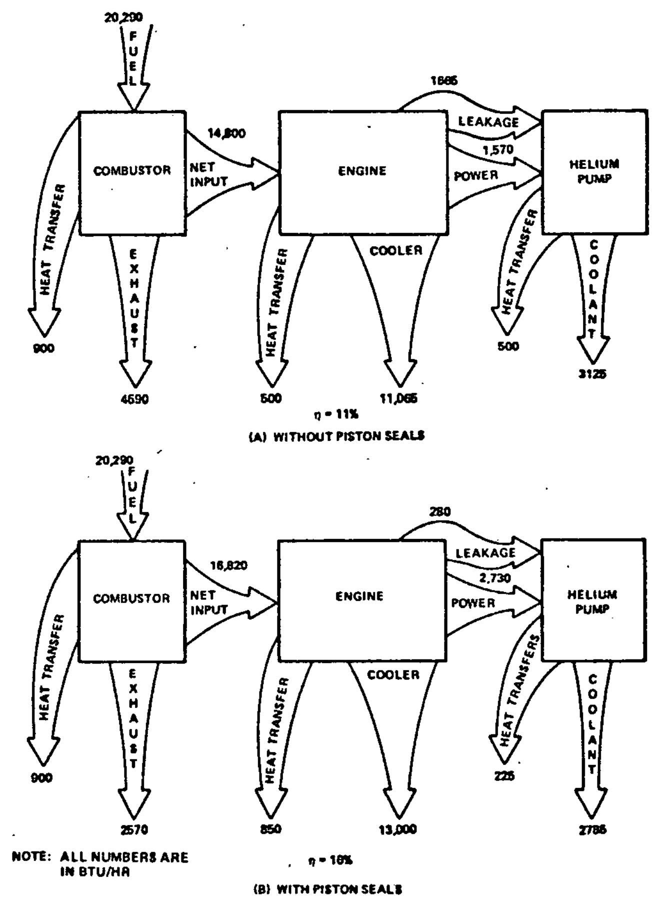


Fig. 3. - Engine test energy balance

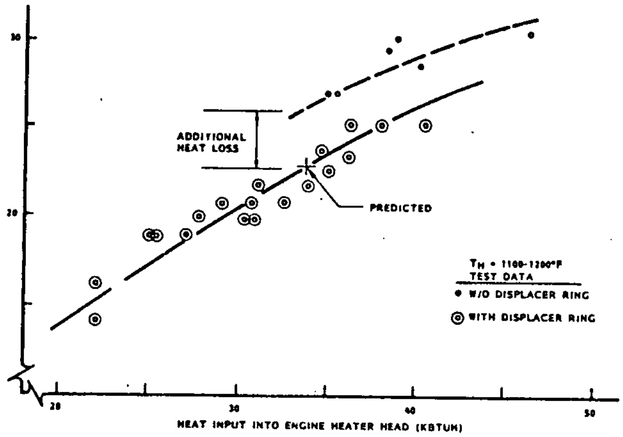


Fig. 4. - System test-engine heat input and output

Operating Characteristics

From test data, engine power piston stroke was identified as the fundamental engine parameter responding to a change in load conditions, i.e., either the compressor suction or discharge pressure level. The engine hot side temperature, efficiency and developed power responded to the power piston stroke variation at a constant combustor firing rate. This is in contrast to a kinematically driven engine, where engine speed is the fundamental parameter responding to a load change and the engine piston stroke remains constant. Thus, for the free-piston Stirling engine, performance maps were generated by characterizing the engine power output as a function of power piston stroke.

Fig. 5 illustrates two superimposed engine performance maps which were constructed from test data with the displacer rod configurations A and B as specified in Table 3.

Table 3 - Displacer Rod Configurations and Their Operating Characteristics

	Displacer A	Displacer B
Rod Area (in^2)	1.2	0.78
Gas Spring Volume (in^3)	7.6	6.8
Displacer/Piston Phase Angle (Degree)	70-100*	30-45
Swept Volume Ratio	1.25-1.67*	0.55-0.83

*Sensitive to load, charge pressure, etc.

Fig. 5 shows that, for a constant firing rate and a given engine configuration, the developed engine power decreases as the power piston stroke increases. This is due to the fact that, as the power piston stroke is increased, the engine swept volume is increased. For a constant heat input, the hot side temperature decreases as the swept volume (stroke) increases. Since the engine cold side temperature remains relatively insensitive to stroke, the Carnot cycle efficiency rapidly diminishes as the engine hot side temperature decreases.

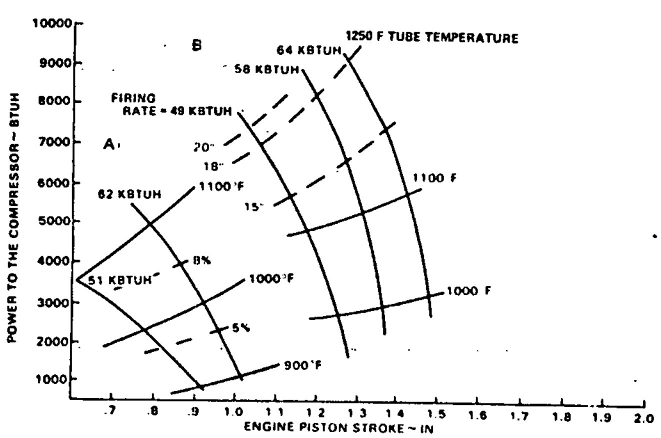


Fig. 5. - Prototype #1 engine/compressor performance maps

The importance of the proper timing of the displacer gas spring configuration is clearly illustrated in Fig. 5. For the displacer rod A, combustor firing rates greater than 62 KBTUH would be required in order for engine efficiencies in the 20 percent range to be realized. On the other hand, test data obtained with the displacer rod configuration B shows that engine efficiency of over 20 percent was obtained with a 49 KBTUH firing rate. Therefore, the rod configuration A would permit engine operation with a larger power output at the optimum efficiency point than does the configuration B. Analysis has also confirmed that the engine with the displacer rod configuration A would produce much more power than the design goal of 3 kW power output if it were operated at its maximum efficiency point. This engine hardware is therefore, somewhat oversized for operation with a three-ton refrigerant compressor and can be reduced in size for the product application.

Load Matching

As indicated in the above discussion, operation of a free-piston Stirling engine is extremely sensitive to a change in load conditions. In order to achieve a high engine efficiency, the refrigerant compressor natural frequency should be properly matched with the engine operating conditions. Greater flexibility in adjusting the compressor suction and discharge pressures in the laboratory test permitted the complete mapping of the engine performance shown in Fig. 5. However, for a given heat pump system, the compressor suction and discharge conditions are primarily determined by the ambient temperature. Therefore, matching between the engine and the compressor must be accomplished most effectively with design optimization of the engine and the compressor.

Three different compressor piston weights were tested in order to investigate matching characteristics. The test results from the heat pump system are shown in Fig. 6 and have been superimposed on the performance map of Fig. 5. Better performance was obtained with a heavier compressor piston due to a closer match of the resonant conditions between the compressor natural frequency and the engine operating frequency.

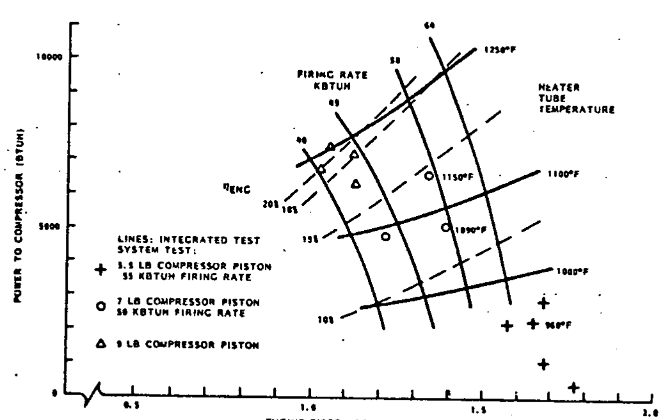


Fig. 6. - System test

Comparison with the Analysis

Comparisons between the test data and the analytical predictions from a thermodynamic simulation model have been found to be in excellent agreement.

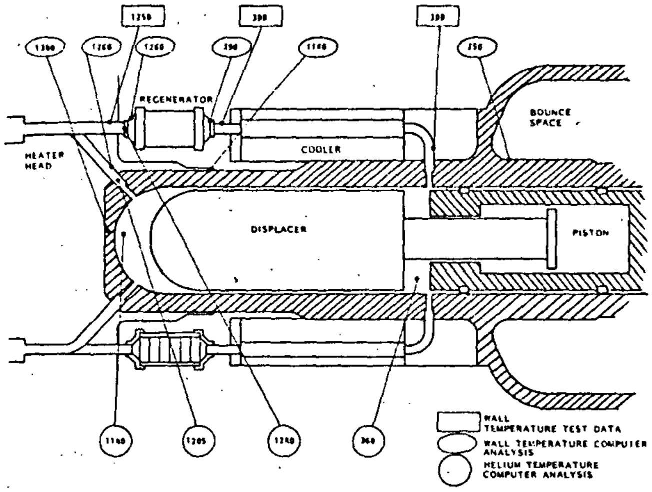


Fig. 7. - Engine temperature distribution (°F)

Fig. 7 shows typical metal temperatures at various locations in the engine as measured and predicted. Even though no direct measurement of heat exchanger and regenerator effectivenesses were available, good agreement between the measured and the predicted temperature distributions were indicative of satisfactory performance of these components. The dynamics of the moving masses also agreed well with the analysis. Fig. 8 shows a typical comparison.

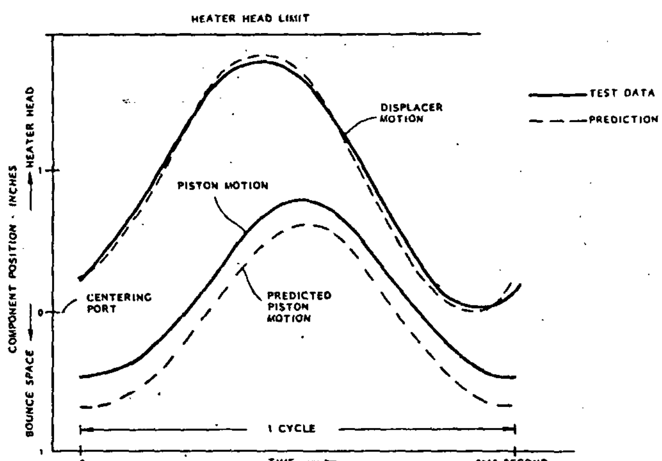


Fig. 8. - Typical engine component motion

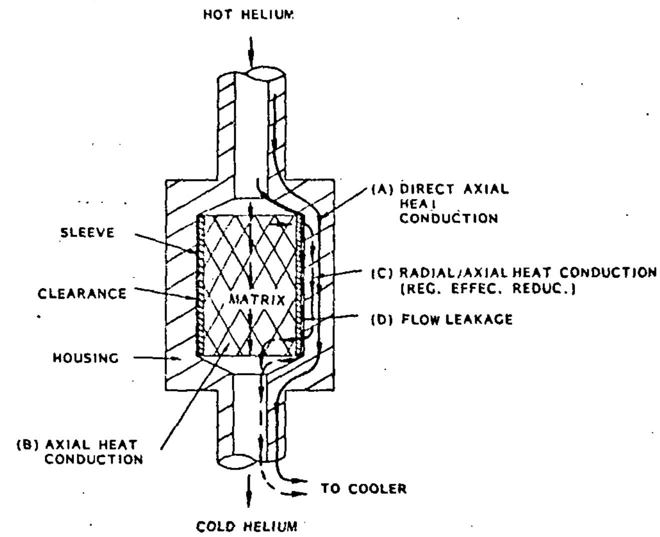


Fig. 9. - Regenerator - thermal losses

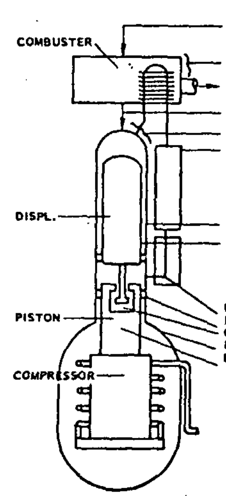
Discussion

The first prototype free-piston Stirling engine has served to demonstrate its operation and load matching characteristics for the heat activated heat pump application. The engine efficiency initially demonstrated was in the low 20% range, lower than the eventual goal of 30%. This was in part due to compromises made on the test hardware to insure design flexibility during testing.

One of the main heat losses in the engine occurs at the regenerator. Fig. 9 shows the first prototype regenerator design. The regenerator matrices were contained in cylindrical shells which were placed inside thick-walled, two-piece housings. The design was adopted to provide flexibility in replacing regenerator matrices to that the best regenerator matrix could be identified. However, additional heat losses were introduced. As shown in Fig. 9, the principal losses are: the axial heat conduction across the thick housing, axial heat conduction across the matrix, the radial heat conduction from the hot side matrix to the housing and then back to cold side matrix, and the flow leakage through the gaps between the matrix cylinder and the housing inside diameter.

Another source of power loss was due to the working fluid leakage through the power piston centering vent ports into the bounce space. This leakage also resulted in an additional loss of high pressure working fluid which could otherwise produce useful work.

Table 4 - Stirling Engine - Energy Balance



	PREDICTED UNITS IN WATTS		TEST	PROJECTED	
TOTAL HEAT INPUT		12110	12922		12110
INSULATION LOSS		328	391		392
EXHAUST		1884	1935		1873
INPUT TO ENGINE		9898	10596		9845
INSULATION LOSS		55	363		243
REGENERATOR LOSSES		1513			517
A) AXIAL WALL COND	276			180	
B) AXIAL MATRIX COND	27			27	
C) RADIAL COND	910			310	
D) FLOW LEAK	300			0	
CYLINDER WALL COND		69			69
DISPLACER LOSSES		203			203
CONDUCTION	11			11	
RADIATION	41			41	
SHUTTLING	151			151	
HEAT TO COOLER		6680	6825		5662
FLOW LEAKAGE			931		293
GAS SPRING LOSS		163	163		163
HEAT CONDUCTION			280		705
ENGINE POWER		3000	2144		3287
ENGINE CYCLE EFFICIENCY		30%	20%		32%

Table 4 summarizes the energy balance of a typical test as compared to the prediction from an earlier analysis. Good agreement was obtained between the test and the predictions on the heat input to the engine and the heat loss to the cooler. However, the engine power output as measured was significantly lower than that predicted. The discrepancy was due mainly to the flow leakage and heat conduction across the power piston section. With the improved designs of a thin-wall regenerator housing and reduction of flow leakage and other heat losses, an engine efficiency of over 30% can be realistically projected.

Conclusion

Significant progress has been accomplished in analyzing, designing and developing a free-piston Stirling engine for a heat-activated heat pump system. Critical parameters influencing the engine operation have been identified and the performance characterized. The first prototype development effort has laid important ground work for the second prototype development. While there is much more work to be accomplished, confidence in the commercial development of a heat activated heat pump driven by a free-piston Stirling engine has substantially increased.

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