EVALUATION OF POWER CONTROL METHODS FOR FREE-PISTON STIRLING ENGINE DRIVEN HEAT PUMPS*

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Abstract

Aside from the problems of starting, a free-piston Stirling engine (FPSE) driven heat pump will tend either to stall or to overstroke if the FPSE power output is not controlled or matched to the compressor power requirement. Ambient temperature variations can create large changes in the power required to compress the refrigerant. An analysis was performed to investigate power control options for FPSE heat pumps. Six different FPSE power control methods were analyzed to determine their effect on the thermal performance of a heat pump system. The six control options are: 1) a displacer gas spring average volume control, 2) a displacer gas spring damping valve control, 3) a displacer alternator control, 4) an engine average pressure control, 5) an engine compression space average volume control, and 6) a compressor clearance volume control. Predicted engine efficiencies were best for control options 4 and 6, and worst for options 2 and 3. The heat pump heating and cooling capacities were most nearly optimum for option 3, and worst for option 4.

Introduction

The U. S. Department of Energy (DOE) through Oak Ridge National Laboratory (ORNL) has sponsored research to develop Stirling engine driven heat pumps for residential applications[1] for several years. One of the concepts being pursued is a free-piston Stirling engine (FPSE) driving a Rankine cycle heat pump, because a free-piston machine can be hermetically sealed.

Some unique control problems, however, occur in FPSE heat pumps, because the power required to drive the refrigerant compressor depends on ambient conditions. A control system is needed to ensure that the engine power output matches the compressor power requirements at all ambient temperatures. If the FPSE and compressor are not properly matched, then the heat pump system will not work. The objectives of this study are: 1) to review briefly the power control systems on some existing Stirling engines, and 2) to present the results of an analysis of FPSE heat pump power control methods which may aid in the development of FPSE heat pumps.

Review of Power Control Methods

In a FPSE, the engine power output is transferred to the load through the motion of the power piston. Because the motion of the power piston is not constrained by any mechanical linkages, the FPSE/load system will operate at a power piston amplitude such that the average power output of the engine matches the average power absorbed by the load, as shown in Fig. 1.

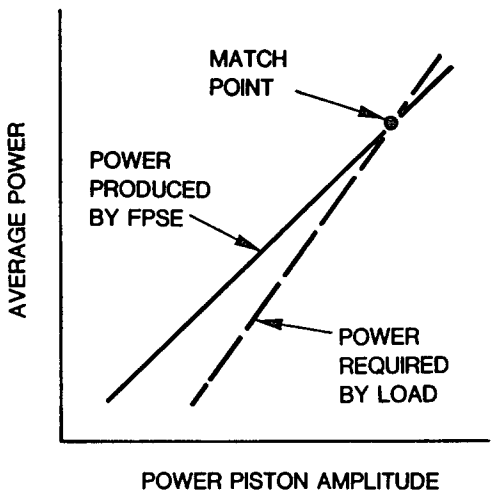

Figure 1. Illustration of FPSE/load matching.

The load shown in Fig. 1 results in a stable match point because the load line is steeper than the engine line. When operating at a (power) piston amplitude that is less than the match point, the FPSE produces more power than the load requires, which tends to drive the piston amplitude back toward the match point. If the piston amplitude is greater than the match point, then the FPSE produces less power than the load requires, and this also drives the system back toward the match point. A FPSE/load system with a load line that is flatter than the engine line will be unstable, and its power piston will either stall or overstroke.

FPSE power control systems simultaneously regulate the power output of an engine and the amplitude of its power piston by changing the position of the match point. FPSE power control methods may be divided into two general categories: 1) engine-side controls, and 2) load side controls. An engine-side control shifts the position of the engine line (see Fig. 2), while a load-side control shifts the position of the load line (see Fig. 3). The engine power output and the piston amplitude can both be reduced by a downward movement of the engine line or an upward movement of the load line.

An example of a load-side power control is the dashpot loading device in the Sunpower "RE-1000" FPSE that was tested recently by NASA Lewis Research Center[2,3] (LeRC). Helium working gas is pumped between the dashpot and the engine buffer space through a tapered orifice. The load line

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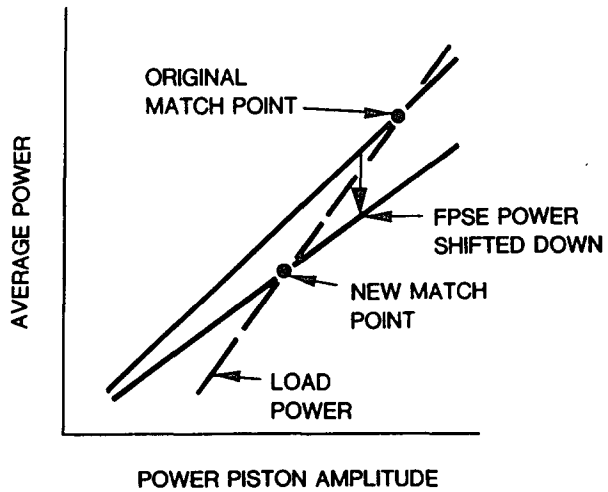

Figure 2. Illustration of engine-side power control strategy for a FPSE.

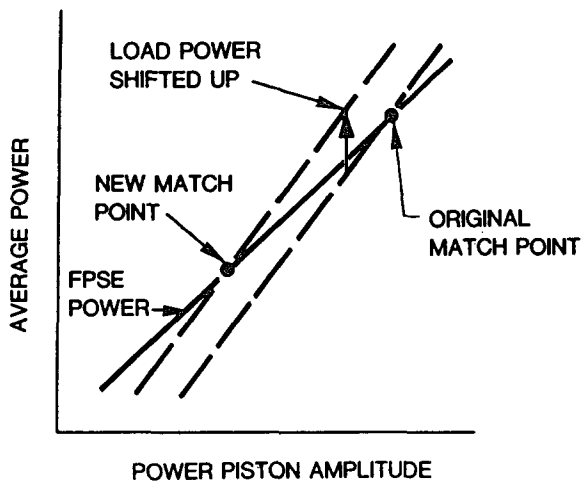

Figure 3. Illustration of load-side power control strategy for a FPSE.

can be shifted either upward or downward by increasing or decreasing the flow resistance through the orifice.

The "Breadboard" FPSE heat pump that was designed and tested by Mechanical Technology Inc.[4] (MTI) uses an engine-side power control that consists of a linear electric motor built into the displacer of the free-piston engine. The motor is used to adjust the amplitude of the displacer motion. When motor current is reduced, the displacer amplitude decreases and the engine line shifts downward.

Another engine-side power control is the adjustable displacer gas spring in the MTI "Technology Demonstrator Engine"[5,6] (TDE). A volume control rod changes the average volume, and therefore the stiffness of the displacer gas spring. The primary effect of the volume control rod is to change the displacer phase angle. There is also a damping valve between the displacer gas spring and the buffer space in the TDE. As the valve is opened, gas spring damping increases and displacer amplitude decreases.

Variable working gas pressure is the preferred power control method in most kinematic Stirling engines. Variable pressure has not been used as the primary power control in any of the free-piston machines that have been tested so far, though it is a feasible FPSE control option.

Analytical Procedure and Results

An R-22 refrigerant compressor was designed for the RE-1000 FPSE with the compressor piston coupled directly to the engine power piston as shown in Fig. 4. The effects that several different power control methods would have on the performance of the "modified RE-1000" FPSE/compressor system were then estimated.

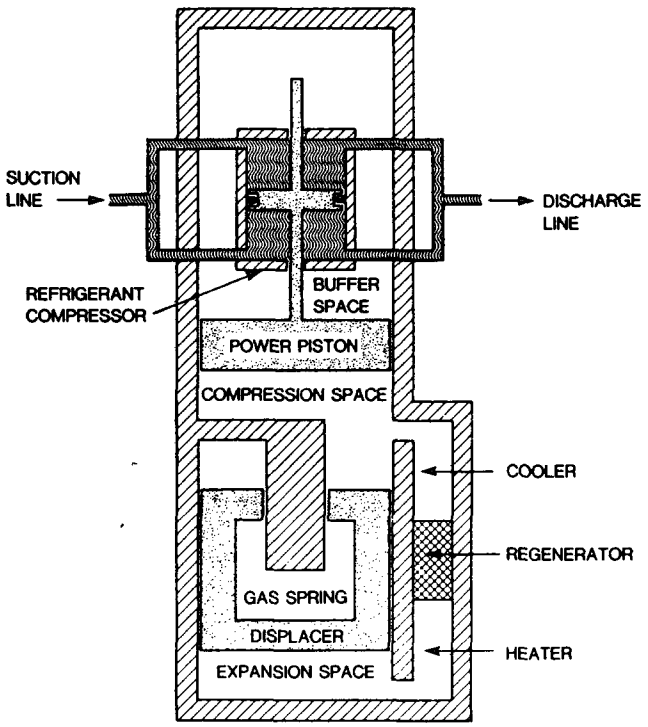

Figure 4. Simplified schematic of modified RE-1000 FPSE with compressor load.

FPSE Design Point Predictions

A slightly modified version of the ORNL Linear Harmonic Analysis Version 1 (LHA1) computer program[7] was used to predict the thermodynamic and dynamic performance of the RE-1000 FPSE. The LHA1 modifications consisted of pressure drop model improvements that are described in ref. 8.

The analytical procedure was to define a design point for the RE-1000 FPSE predictions that corresponded to NASA LeRC test #602 reported in ref. 3. The RE-1000 engine dimensions and operating conditions that were used in the LHA1 program for the design point calculations are given in ref. 8.

The LHA1 program does not contain correlations to calculate the magnitude of the heat transfer, pressure drop, and seal leakage losses. Instead, loss coefficients were selected and specified in the LHA1 input data to calibrate the predicted values of displacer amplitude, indicated power output, and etc. to the measured values from test #602. Table 1 shows that the predicted values are within ±4% of the measured values for all of the performance parameters except displacer phase angle.

Compressor Load Characteristics

A computer program was written to predict the performance of the R-22 refrigerant compressor in the modified RE-1000 FPSE. Loss calculations for valve pressure drop, suction gas superheating, and seal leakage were included. Four ambient temperatures were examined: two cooling points [35.0 and 26.7°C (95 and 80°F)] and two heating points [8.3 and -8.3°C (47 and 17°F)]. The R-22 pressures and temperatures in the suction and discharge lines that were assumed for each ambient temperature are provided in ref. 8 along with more details about the compressor performance predictions.

The dimensions of the refrigerant compressor were selected so that the predicted compressor power requirement at the 35.0°C (95°F) ambient conditions matched the LHA1 design point prediction for engine power output. The ratio of clearance volume to maximum volume displacement by the compressor piston was assumed to be 10%. The compressor piston diameter was determined to be 4.095 cm, which is about 70% of the RE-1000 power piston diameter.

Figure 5 shows the work per cycle required to drive the RE-1000 compressor at the four ambient conditions and the work produced by the RE-1000 FPSE as a function of power piston amplitude. Work per cycle, which is defined as average power divided by frequency, is plotted along the ordinate in Fig. 5 so that the compressor load lines are independent of frequency.

The compressor load lines in Fig. 5 define the range in which a control mechanism is needed so that the work produced by the FPSE matches that required by the compressor at the different ambient conditions. As shown, the FPSE/compressor system will operate at its design piston amplitude

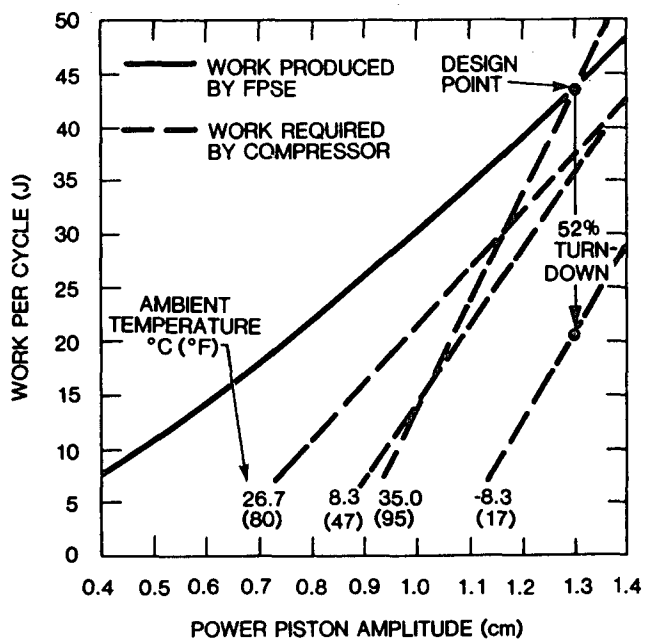

Figure 5. Predicted performance of modified RE-1000 FPSE with compressor load (compressor piston diameter = 4.095 cm, compressor clearance volume = 10%).

of 1.3 cm when the ambient temperature is 35.0°C (95°F). At the other ambient conditions, the engine produces more work than the compressor requires, and the system cannot be operated unless the engine line is shifted downward by an engine-side control or the compressor lines are shifted upward by a load-side control. Figure 5 shows that the engine work output will have to be reduced 52% (from 43.52 J to 20.95 J) to match the -8.3°C (17°F) compressor line at a piston amplitude of 1.3 cm.

Comparison of Power Control Methods

Five engine-side controls were examined. These included: 1) a displacer gas spring average volume control, 2) a displacer gas spring damping valve control, 3) a displacer alternator control, 4) an engine average pressure control, and 5) an engine compression space average volume control. One load-side control, an adjustable compressor clearance volume, was examined.

Table 1. Comparison of LHA1 design point predictions and RE-1000 test #602

Parameter	LHA1 predictions	Test #602
Power piston amplitude, cm	1.30	1.29
Frequency, Hz	29.3	29.9
Displacer amplitude, cm	1.37	1.36
Displacer phase angle, deg	49.7	57.1
Compression space pressure phase angle, deg	-21.3	-21.6
Indicated power output, W	1276	1262
Cooler heat output rate, W	2894	2789
Heater head heat input rate, W	4170	4051
Indicated thermal efficiency, %	30.6	31.2

For the analysis of the engine-side controls, the LHA1 computer program for the RE-1000 FPSE was used to predict how much a particular engine dimension or parameter had to be changed in order to move the engine line from the 35.0°C (95°F) ambient design point to the 52% turndown point for the -8.3°C (17°F) ambient conditions. This temperature range was chosen because it represents the two extremes in compressor work requirements at a power piston amplitude of 1.3 cm.

For the analysis of the load-side control, the compressor computer program was used to predict how much the compressor dimensions had to be changed so that its work requirements at both the 35.0°C (95°F) and -8.3°C (17°F) ambient conditions matched the RE-1000 engine work output at the design point.

The results for the six control methods analyzed as well as the design point predictions are presented in Table 2 and discussed below individually.

Displacer Gas Spring Average Volume Control. Increasing the average volume of the gas spring results in a reduction in the displacer gas spring stiffness, which affects the engine dynamics. To achieve a 52% turndown in engine work, a 445% increase in displacer gas spring volume was required. The primary effect was a displacer phase angle reduction to 25.1°, which is almost half of the design point value. Engine efficiency and frequency also dropped.

Displacer Gas Spring Damping Valve Control. For this control, a 52% turndown in engine work was achieved by increasing the leakage of the working gas between the displacer gas spring and the buffer space. This resulted in reductions in both displacer amplitude and displacer phase angle. Engine efficiency also fell, but not as much as might be expected, considering that gas spring losses were increased.

Displacer Alternator Control. This control regulates the displacer motion through the magnetic forces it applies on the displacer. The alternator was represented analytically as a linear dashpot; i.e., the magnetic forces on the displacer were assumed to be proportional to the negative of displacer velocity, and the power absorbed by the linear dashpot was assumed to be useful electric power. As shown in Table 2, the largest turndown effect that the displacer alternator had on the engine predictions was a 36% decrease in displacer amplitude.

Engine Average Pressure Control. The operating frequency of a FPSE depends on the average working gas pressure. To achieve a 52% turndown in engine work, a 59% reduction in average pressure was required, which resulted in a 38% decrease in engine frequency. However, engine efficiency improved with a drop in average gas pressure. For this control method, the heat pump heating capacity was very low because the heating capacity is nearly proportional to frequency.

Engine Compression Space Average Volume Control. To achieve a 52% turndown in engine work, the compression space average volume was increased 282%. The average heat transfer surface area in the compression space was also increased accordingly. Significant changes occurred in engine frequency, displacer amplitude, and displacer phase angle.

Compressor Clearance Volume Control. A modified design procedure was required for the adjustable clearance volume compressor. The compressor was not sized for the 35.0°C (95°F) cooling load. A clearance volume ratio of 10% was assumed and it was sized instead for the -8.3°C (17°F) heating load. A compressor piston diameter of 5.902 cm was selected so that the -8.3°C (17°F) compressor line passes through the engine design point.

At the other ambient temperatures, the adjustable clearance volume is used to control the work

Table 2. Summary of performance predictions for design point and six different power controls

	Design point	Displacer gas spring average volume increased 445%	Displacer gas spring damping valve opened	Displacer alternator current increased	Engine average pressure reduced 59%	Engine compression space average volume increased 282%	Compressor clearance volume reduced from 26% to 10%
Control category	-	engine-side	engine-side	engine-side	engine-side	engine-side	load-side
Operating conditions							
Ambient temperature, °C (°F)	35.0 (95)	-8.3 (17)	-8.3 (17)	-8.3 (17)	-8.3 (17)	-8.3 (17)	-8.3 (17)
Piston amplitude, cm	1.300	1.300	1.300	1.300	1.300	1.300	1.300
Work per cycle, J	43.52	20.95	20.95	20.95	20.95	20.95	43.52
Engine dynamics							
Frequency, Hz	29.32	25.12	28.74	31.60	18.26	23.62	29.32
Displacer amplitude, cm	1.367	1.392	1.081	0.876	1.495	1.001	1.367
Displacer phase angle, deg	49.7	25.1	37.2	54.8	49.7	81.2	49.7
Engine thermodynamics							
Indicated power output, W	1276	526	602	662	382	495	1276
Electric power output, W	0	0	0	117	0	0	0
Cooler heat output rate, W	2894	1413	1814	2325	801	1416	2894
Heater head heat input rate, W	4170	1939	2416	3104	1183	1911	4170
Indicated thermal efficiency, %	30.6	27.1	24.9	25.1[a]	32.3	25.9	30.6
Heat pump capacity							
Cooling, W	4581	-	-	-	-	-	-
Heating[b], W	-	3243	3908	4628	2132	3137	7333
Ratio of heating to cooling	-	0.71	0.85	1.01	0.47	0.68	1.60

[a]Efficiency is based on sum of indicated and electric power outputs.
[b]Heating capacity includes refrigerant condenser heat output rate and engine cooler heat output rate.

requirement of the compressor so that it matches the engine work output. The appropriate clearance volume ratio for the 35.0°C (95°F) conditions was determined to be 25.9%. Figure 6 illustrates the operation of the compressor clearance volume control. It should be noted that the compressor isentropic efficiency predictions were not affected by clearance volume, piston amplitude, or piston area.

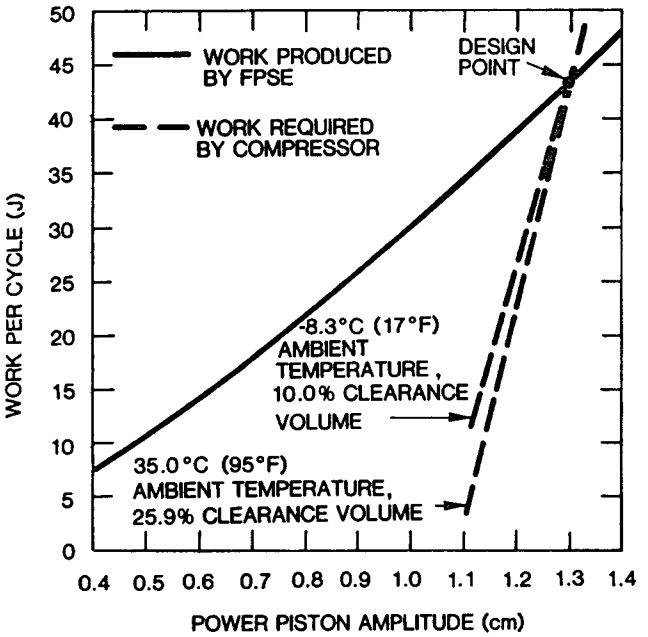

Figure 6. Predicted performance of adjustable clearance volume compressor (compressor piston diameter = 5.902 cm).

Conclusions

The results in Table 2 from the analysis of six different power control methods for a FPSE heat pump showed some interesting trends in thermal performance. Predicted engine efficiencies were highest for the engine average pressure and compressor clearance volume controls, and lowest for the displacer gas spring damping valve and displacer alternator controls. The heating/cooling capacities of a heat pump will match the requirements of a typical home best if the -8.3°C (17°F) heating capacity is about equal to or slightly greater than the 35.0°C (95°F) cooling capacity. The predicted FPSE heat pump heating/cooling capacities were best for the displacer alternator control, and worst for the engine average pressure control. The selection of a power control system for a FPSE heat pump will depend not only on thermal performance and balanced capacity, but also on other factors such as mechanical complexity, equipment costs, control response time, and problem of starting.

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