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G. Chen and J. McEntee
Sunpower, Inc.
Athens, Ohio

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HARDWARE DEVELOPMENT AND INITIAL SUBASSEMBLY TEST OF A GAS-FIRED STIRLING/RANKINE RESIDENTIAL HEAT PUMP

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ABSTRACT

A gas-fired Stirling/Rankine heat pump is being developed at Sunpower, Inc. The free-piston Stirling engine/magnetic coupling/refrigerant compressor (FPSE/MC/C) assembly used as the power module for this type of heat pump is currently in the assembly and test phase. To achieve high efficiency, low cost, and a more durable system, modifications have been made to a previously introduced design. The modifications include changes in material selection, a different displacer drive, and the use of a low-cost and more efficient cooler design. A commercially available R-22 compressor is used in the prototype. Low cost iron-neodymium permanent magnets are used to provide an efficient magnetic coupling design. To match the engine power to the load, a double-acting variable gas spring is arranged in parallel with the engine and compressor. After the gas spring was designed and fabricated, it was tested with the compressor. Before system integration and tests, the engine/alternator and the compressor/heat pump have been set up and are to be tested separately.

1. INTRODUCTION

A gas-fired free-piston Stirling engine is used to drive a refrigerant compressor in a residential heat pump system. The engine is magnetically coupled to a compressor/gas spring assembly. Evaluation of a successful magnetic coupling in previous tests [1] and a study of the performance of a preliminary system design [2] showed the design concepts are feasible. The current program at Sunpower deals with the hardware development and system tests of the FPSE/MC/C assembly. The system tests to be performed involve component testing, proof-of-concept testing and performance mapping.

2. SYSTEM SPECIFICATIONS AND PRIMARY COMPONENTS

The FPSE/MC/C assembly being constructed is schematically illustrated in Figure 1. The prototype assembly consists of a natural gas burner, a heat pipe, a 3 kW free-piston Stirling engine, a permanent magnet axial coupling, a variable gas spring, and a R-22 compressor.

A direct-fire burner is an ideal candidate for a mass production design of the system. Such a burner would have to be developed if this heat pump system were to be commercially feasible. For testing purposes, a 40 kW, 10-finned gas-fired sodium heat pipe was selected as the thermal energy source for the free-piston Stirling engine. The temperature of the heat pipe evaporator is regulated by a programmable controller and electronic gas and air mass flow meters. The air/fuel ratio is fixed by these flow meters and is held around 50 percent excess air. These controls are necessary to keep the heat pipe operating within specified temperature limits.

The magnetic coupling is a means of hermetic power transmission and so links the engine piston and the compressor piston. Relatively cheap and easily available iron neodymium magnets are used in the design. A thin-walled titanium pressure vessel separates the inner and outer magnets and the engine bounce space from a volume of Freon held at mean compressor pressure. Titanium is used to reduce magnetic hysteresis losses in the pressure vessel wall. Hysteresis losses increase as wall thickness and electric conductivity increase [1]. Stainless steel with its relatively lower cost would be used in a production model.

The variable gas spring acts to match the engine and compressor load and is described in detail in a previous document [3]. It also maintains a constant engine stroke to achieve high volumetric efficiency in the compressor. Thus, as the cooling or heating load on the compressor varies, the load on the engine adjusts and leads to changes in engine operating frequency. The stiffness of the gas spring varies to keep the engine, the coupling, the gas spring, and the compressor piston a resonant mass-spring system over the entire operating frequency range.

The assembly design specifications and operating parameters are listed in Table 1. The engine operating

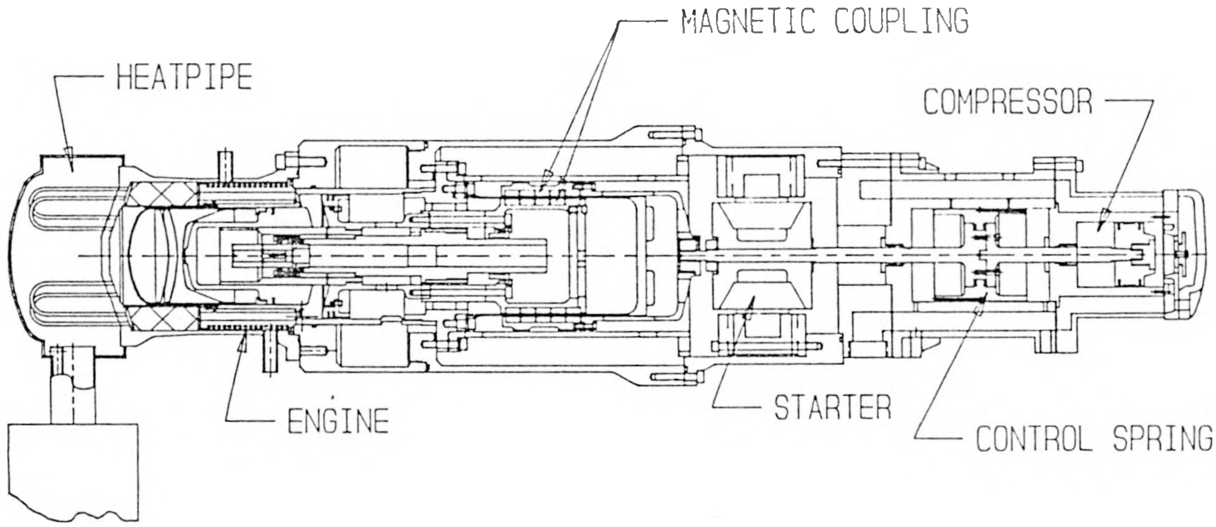

Figure 1 FPSE/MC/C Test Prototype

frequency is expected to vary between 60 hertz and 64 hertz, producing full power at 60 hertz and 25 percent of full power at 64 hertz. The full power operating frequency of 60 hertz was selected, in part, to comply with refrigerant compressor availability, and also to allow for electric power generation [2].

The mean engine pressure is 2.75 MPa. This pressure is relatively low compared with the 6 MPa chosen by most other FPSE-driven heat pump systems [4, 5]. This lower pressure allows the pressure vessel walls to be made thinner resulting in a more efficient magnet coupling.

The hot-end temperature of the system is limited by the heater tube and heater head material. The 316 stainless steel head and tubes in the testing assembly are designed for a 10,000 hour creep rupture life. The heater head is shown in Figure 2.

The cold-side temperature is selected so that the rejected heat from the engine in the cooling water is rejected to the environment in a closed cooling loop with a standard heat exchanger. The rejected heat could also be used for space heating and to heat water for domestic purposes, increasing the overall system efficiency.

There are a few differences between this prototype and the preliminary design described in [2], and these are described below.

Table 1. FPSE/MC/C Test Assembly Design Specifications

Working Gas	Helium
Maximum Output Power [kW]	3.3
Operating Frequency [Hz]	60 - 64
Design Pressure [MPa]	2.75
Design Phase Angle [degree]	48
Engine Design Efficiency (Full Power) [%]	35
Engine Heater	
Type	Tube
Tube-Wall Temperature [°C]	750
Engine Cooler	
Type	Fin
Wall Temperature [°C]	40
Regenerator	
Type	Random Packed Wires
Porosity	0.75
Magnetic Coupling	
Permanent Magnet Material	Iron Neodymium
Energy Product [kJ/m^3]	224
Loss [W]	150
Maximum Load Capacity [N]	2,600

A single-acting gas spring is used in the displacer due to its simpler mechanical design. A fraction of the rod area seen by the displacer gas spring is sprung to ground while

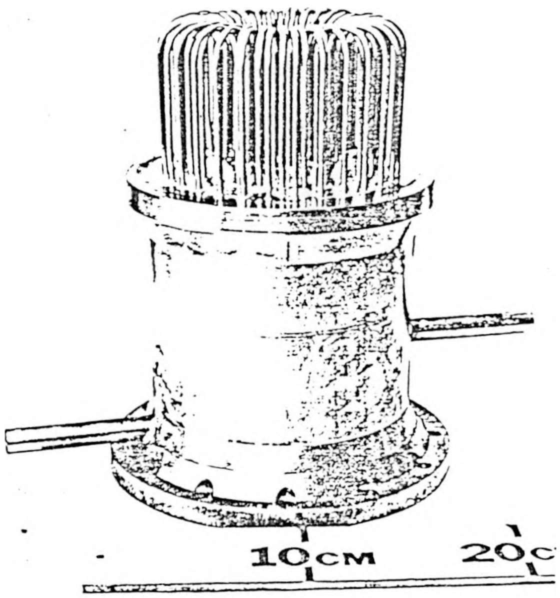

Figure 2. Engine Heater Head

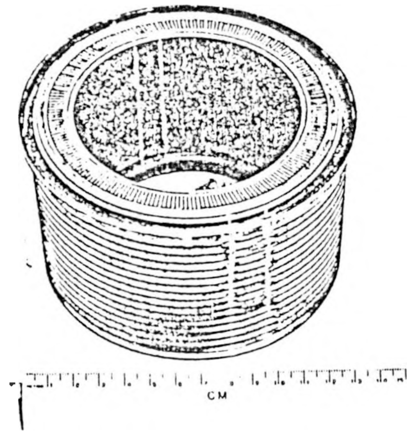

Figure 4. Engine Cooler

the remainder is sprung to the power piston. This displacer drive minimizes the tolerance stack up because the displacer no longer needs to be aligned with the rod. Displacer components are shown in Figure 3. Using a single-acting gas spring also reduces by one the number of seals required. A spider is included to support the displacer rod.

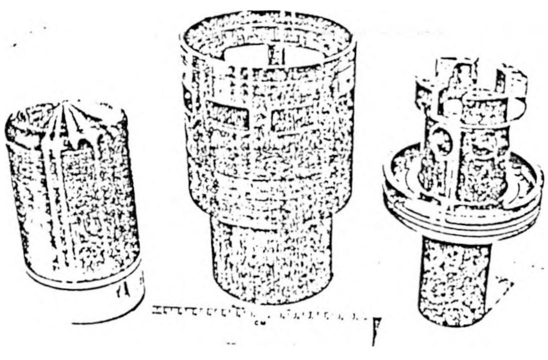

Figure 3. Displacer/Piston/Cylinder Components

Figure 4 shows a fin type cooler made from copper. The gas passes through fins that run axially through the cooler while the cooling water runs along radial fins on the outside of the cooler. This type of construction is expected to be less expensive than the tube bundle cooler presented in the previous design. In additon, the performance of this cooler is expected to improve.

The main cylinder is made from cast iron which costs less than the previous design and which also has higher conductivity. This ensures that the bearings and seals will be better cooled than in the previous design. The piston/cylinder components are also shown in Figure 3.

The magnetic coupling (shown in Figure 5) in this assembly has been built and tested for its static behavior. The measured force-deflection characteristic is plotted in Figure 6. The maximum load capacity is about 2,600 Newtons. The force required to transmit full power is around 1,100 Newtons for a double-acting reciprocating compressor with the designed operating parameters.

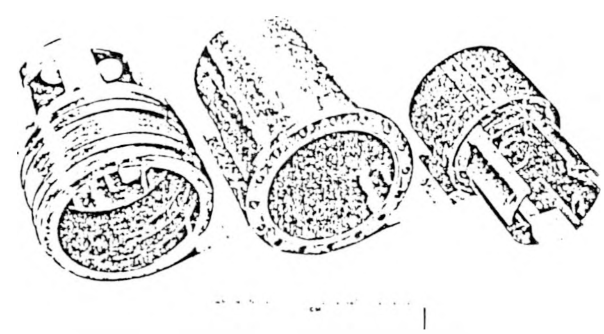

Figure 5. Magnetic Coupling Components

A clearance seal is used for the seal between the displacer and piston and also for the seal separating the gas spaces in the variable gas spring. For easy

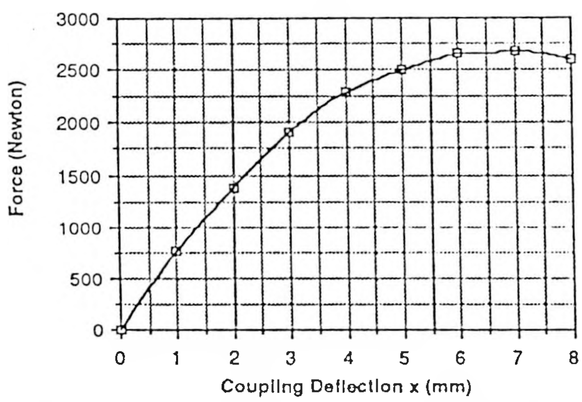

Figure 6. Coupling Force-Deflection Characteristic

implementation of a test program, ring seals are used throughout the rest of the engine.

A commercially available single-acting reciprocating R-22 compressor modified at its interface to the engine assembly is used in this testing system. The variable gas spring and the compressor components are shown in Figure 7 and Figure 8, respectively.

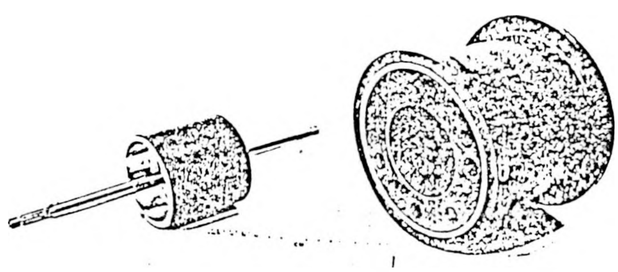

Figure 7. Control Spring Components

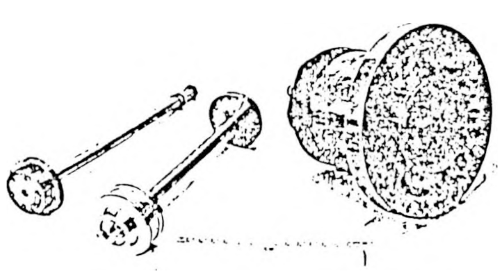

Figure 8. Compressor Components

3. TEST SETUP

The testing program principally involves the following three distinct areas: component testing, proof-ofconcept testing and performance mapping. Before a system integration (schematic of arrangement shown in Figure 9) and test, the engine and the compressor will be tested separately. In the near-future engine test, a 3 kW linear alternator previously built at Sunpower will be used. The alternator will be a motor driving the engine and a load to the engine. Initially, the compressor will also be driven by the alternator to test its performance with a refrigerant loop.

Because of the heat pipe's sensitivity to vibration, it is very important to reduce the amplitude of the engine casing vibration as much as possible. Since the engine is expected to run over a range of frequencies, it was decided not to use a dynamic absorber in the early stages of testing. The assembly to be tested is hung from the ceiling using steel cables and a large mass is attached to the assembly casing. As the spring stiffness of the steel cables is comparatively small, the casing motion amplitude is approximated by $X_c=F_i/M\omega^2$ where X_c is the casing amplitude, F_i is the excited force, M is the mass of the assembly including the attached mass, and ω is the angular velocity of the excitation. A finite element model was created and analyzed to determine the system resonant frequencies and the stresses on the heat pipe.

A data acquisition system created at Sunpower for free-piston machines was modified for use in this program. The modified acquisition system has the ability to measure and record the data relevant to compressor performance, as well as the engine and linear alternator parameters.

The variable gas spring was designed to provide a spring force in parallel with that of the compressor so that power-load matching and system tuning are preserved over a wide range of heat pump operating conditions. To ensure stable operation of the compressor with its required capacity and of the variable gas spring with its specifically designed features, the variable gas spring/refrigerant compressor subassembly to be used for the FPSE/MC/C assembly has been fabricated and tested in parallel with the engine-subassembly fabrication and prior to the FPSE/MC/C system integration. A commercially available, single-acting R-22 compressor modified to interface to the rest of the FPSE/MC/C assembly is used.

The variable gas spring/compressor subassembly was installed on a motor/crank/magnetic coupling test rig

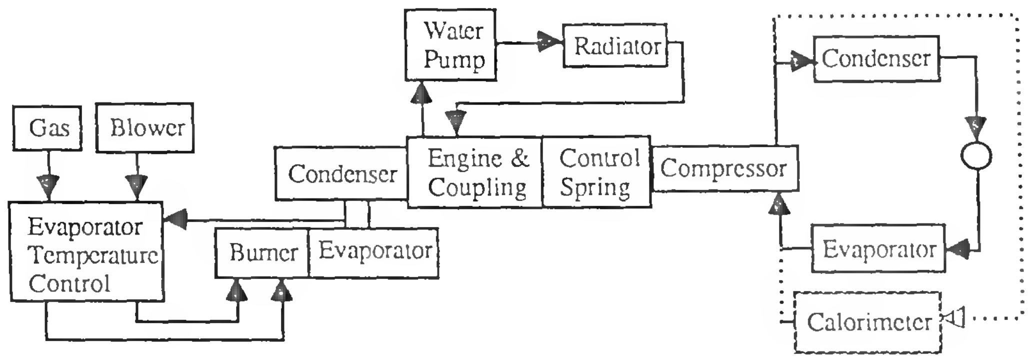

Figure 9. Schematic of Test System

described in reference [1] and operated at 35 hertz and 58 hertz during the initial tests. The compressor running on air provided a load for the test rig operation. The following changes were made to show how subassembly performance is affected by design parameters: (a) sizes of the orifices connecting the high- and low-pressure air flow to the spring working space, (b) sizes of the orifices connecting the high- and low-pressure air flow to the compressor back-pressure space, (c) tension on the check valve springs back-up for the bleed orifices on the gas spring, (d) tension on the check valve springs for the compressor back-pressure space, and (e) compressor discharge pressure.

Summary of results made from observation of the unit's initial operation are summarized as follows: (a) the gas spring is effective; (b) the orifice sizes effectively control the sensitive and stable response of the system; (c) the gas spring should be carefully sized to reduce the gas-spring hysteresis loss, even though the hysteresis loss associated with R-22 is small because of its low thermal conductivity; (d) the back pressure of the compressor is effective in balancing the compressor-piston motion.

4. SYSTEM PERFORMANCE PREDICTION

One of the primary motives to develop a gas-fired Stirling/Rankine heat pump is an interest in exploring the benefits of the relatively high frequency of the free-piston Stirling engine. Prior to ~~a~~ system integration and tests, the system performance of the constructed assembly is analytically evaluated.

The Stirling engine efficiency was evaluated from SAUCE, a Sunpower computer simulation code of a FPSE design. An efficiency of 35 percent was predicted at full power and 28 percent efficiency at 40 percent of the full power.

The heating system of the engine in the FPSE/MC/C testing assembly is a burner/heat pipe unit without an air preheater. Thus, the overall system COP to be measured in tests will be lower than a system employing a burner with an air preheater. Adjustment of the measured information based on available preheater data for the presence of a preheater will yield the expected COP of the final design. Therefore, two system efficiencies corresponding to two burner efficiencies will be evaluated: the measured system COP without an air preheater, and the adjusted system COP with an air preheater. A burner efficiency of 80 percent to 85 percent based on the gas high heating value (HHV) is used at this point to analytically evaluate the system performance.

Power loss in the titanium pressure vessel in between the inner and outer magnetic coupling rings at full-power engine operation was calculated to be about 150 watts by a method introduced in the previous magnetic coupling evaluation [1]. This is about 5 percent of the engine output power. The compressor total thermal efficiency of 75 percent is used in the analytical prediction. A definition of overall system thermal COP is:

$$\text{COP} = \frac{\text{Thermal Load in Heating or Cooling}}{\text{Fuel Thermal-Power Input}}$$

The results at typical operating heating and cooling conditions are also provided in Table 2. It is noted that the engine-rejected heat will be partially recovered for heating as the system works as a heat pump. The recovered heat will be approximately the heat rejected from the engine to the cooler. Based on the engine design and engineering experience, this heat is about 85 percent of the total engine-rejected heat.

The output capacity of a heat pump system using the constructed FPSE/MC/C assembly can also be predicted analytically. The results with system performance at various ambient conditions are also tabulated in Table 2.

Table 2. Predicted FPSE/MC/C Heat PumpPerformance (Test assembly, R-22, assuming a compressor-output modulation)

Ambient Temp [°C]	35.0	26.7	8.3	-8.3
Heating/Cooling Load [kW]	-10.5	-5.4	17.5	-
Suction Pressure [MPa]	0.63	0.68	0.53	0.30
Evaporator Temp [°C]	7.2	10.0	1.7	-15.0
Discharge Pressure [Mpa]	1.91	1.36	1.45	1.26
Condenser Temp [°C]	49.4	35.0	37.8	35.0
Degree of Superheat [°C]	10.0	10.0	10.0	10.0
R-22 Flow Rate [kg/h]	247.4	112.2	249.7	-
Operating Frequency [Hz]	60	60	60	60
Compressor Power [kW]	2.69	0.73	2.39	-
Engine Power [kW]	2.84	0.88	2.54	-
Engine Efficiency [%]	35.0	28.0	34.0	35.0
Recovered Heat [kW]		-	3.9	-
Fuel Thermal Input [kW]	10.1	3.93	9.3	-
System COP	1.04	1.37	1.88	1.48

5. CONCLUSIONS

The FPSE/MC/C assembly for a gas-fired residential heat pump is under development in the fabrication and test phase. In summary, the following conclusions can be made from the work described in this paper:

- The assembly for system test and demonstration has been constructed. This assembly is based on modification of the previously introduced preliminary design.
- The system test and demonstration consist of subassembly tests such as the engine test and variable gas spring/compressor test, and FPSE/MC/C system integration and test. The testing program has started with the variable gas spring/compressor subassembly test. The initial tests of the subassembly loaded by air indicated that the gas spring is effective and the design parameters such as the orifice sizes effectively control the spring's sensitivity and stability.
- The performance prediction showed a relatively high engine efficiency and low coupling loss. Both will be tested in the near future.

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