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FURTHER DEVELOPMENTS IN THE DESIGN OF A FREE-PISTON STIRLING ENGINE HEAT PUMP FOR RESIDENTIAL APPLICATIONS

R.A. Ackerman, P.E., Ph.D. **J.M. Clinch, Ph.D.** **G. Privon**

ABSTRACT

During the past year (1986/1987), the development of an improved Stirling engine driver for the Free-Piston Stirling Engine Heat Pump (FPSE/HP) has led to a significant increase in heat pump performance. With the improved engine, the MARK I, the FPSE/HP has achieved its two critical milestones of producing 3.0 refrigeration tons (RT) at the 95 F (35°C) ambient temperature conditions and an engine efficiency of 25%, as measured from the fuel input energy, based on the higher heating value of the natural gas and mechanical power developed by the engine. The overall measured performance of the heat pump module (engine, transmission, and compressor) was:

Capacity
- Q (heating @ 47 F) = (60,000 Btu/h) 63.28 MJ/h
- Q (cooling @ 95 F) = 3.2 RT*

Performance
- COP (heating @ 47 F) = 1.61
- COP (cooling @ 95 F) = 0.86

*Refrigeration tons = 12,000 Btu/h (12.66 MJ/h)

This paper describes the latest configuration of the FPSE/HP module and presents measured performance data. Performance has been measured over a broad range of ambient temperature conditions and engine operating parameters. The results obtained from this testing are presented.

INTRODUCTION

The development of an efficient gas heat pump for a typical cold-climate residence could save up to one-half the natural gas now used by a conventional furnace or one-fourth the energy used by a high-efficiency (90% efficient) furnace. The design requirement for the FPSE/HP was to develop a gas-driven vapor compression heat pump module that would replace an electric compressor in a conventional heat pump. To be an effective alternative to the electric heat pump, the development specifications defined for the FPSE/HP were:

1. The refrigerant circuit would use refrigerant R-22
2. The FPSE/HP module would be hermetically sealed
3. The heat pump module would be capable of capacity load following
4. The heat pump module would meet the following proof-of-concept (POC) performance targets:

R. A. Ackerman is chief engineer for residential heat pump at Mechanical Technology Incorporated, Latham, NY. J. M. Clinch is program manager at Gas Research Institute, Chicago, IL. G. Privon is program manager at Oak Ridge National Laboratory, Oak Ridge, TN.

Capacity

- Q (heating @ 47 F) = 70,000 Btu/h (73.83 MJ/h)
- Q (cooling @ 95 F) = 3.0 RT

Performance

- COP (heating @ 47 F) = 1.5
- COP (cooling @ 95 F) = 1.0

The development of the MARK I heat pump module described in this paper, and presented in Figure 1, is intended to demonstrate the POC targets. A second, future objective is to integrate the heat pump module into a heat pump package, as shown in Figure 2, for field experimentation. Additional background to the development of the FPSE/HP is presented in the referenced literature.

MARK I FPSE/HP DESCRIPTION

The MARK I heat pump module is shown in Figure 1. It consists of the MARK I FPSE coupled to a resonant piston refrigerant compressor through a hydraulic transmission. Physically, the module consists of the MARK I engine, a diaphragm-actuated hydraulic transmission, and a linear-resonant Rankine refrigerant compressor. Located above the engine diaphragm, the FPSE assembly consists of a recuperated natural gas combustor, a monolithic-finned heater head, a motor-driven displacer, and heat exchanger components (i.e., cooler and regenerator). The hydraulic transmission and compressor are located below the engine diaphragm.

The motor-driven displacer enables the displacer stroke to be controlled electronically, providing the primary engine control to the heat pump for load matching at all ambient temperature conditions. Through this control device, the system may be dynamically tuned at every operating point by varying the operating frequency, and the heating and cooling capacity can be matched to the demand by modulating the displacer stroke. These two control parameters proved very effective in controlling and optimizing system performance during testing.

Hermetic separation between the engine working fluid and refrigerant (R-22) is achieved by employing a flexible metal diaphragm between the FPSE and hydraulic transmission. Engine power is transferred to the compressor through the volumetric displacements of the diaphragms and corresponding displacement of the oil in the hydraulic transmission. The motion of the diaphragms is produced by the pressure wave developed in the engine.

As shown in Figure 1, the compressor operates via an increase in engine pressure amplitude, produced from shuttling the engine working fluid between the hot and cold working spaces of the engine, which deflects the diaphragm into the upper oil volume below the engine diaphragm. Because the oil is incompressible and its quantity constant, the power piston is forced to the left, compressing the refrigerant in the left-hand compression volume and drawing refrigerant into the right-hand compression volume. The motion of the power piston to the left also displaces the oil in the lower oil volume above the gas spring diaphragm, forcing the gas spring diaphragm to deflect downward and compressing the gas in the lower gas spring. The reactive forces from the gas spring and compression volumes provide the restoring force for the compressor piston and produce the resonant characteristics of the compressor. The reciprocating motion of the compressor piston produces the suction and discharge strokes that pump the refrigerant from the suction to the discharge manifolds.

The operating specifications for the MARK I FPSE/HP are given in Table 1. The engine used 60 Bar (6.0 MPa) helium as the working fluid, and the hydraulic transmission used a silicone-based oil with a viscosity of 10 cs. The refrigerant was R-22. During testing, the engine was typically run with a heater head temperature of 1337 F (725°C), a cooler inlet water temperature of 77 to 104 F (25 to 40°C), and a laboratory ambient temperature of 77 F (25°C).

BACKGROUND

In 1985, the first generation FPSE/HP driven by an MTI FPSE, the Engineering Model (EM) engine breadboard heat pump module, was tested over ambient temperatures ranging from 0 F (-17.8°C) to 95 F (35°C). This testing demonstrated several important qualities of the FPSE/HP, namely:

1. By employing displacer control, the dynamics of an FPSE/HP, including start-up, could be effectively accommodated, controlled, and optimized over a broad range of ambient temperatures.

2. The hydraulic transmission would operate reliably and efficiently. Measured transmission efficiencies of 85% were consistently recorded at the 95 F (35°C) ambient conditions as compared to a design prediction of 78%.

However, testing with the EM engine fell short of achieving the design goals of 3.0 RT capacity at 95 F (35°C) ambient conditions and a cooling coefficient of performance (COP) of 1.0. The performance map for the EM FPSE/HP is given in Figures 3 and 4. As shown, the following maximum heating and cooling performance was achieved for the heat pump module:

Capacity

- Q (heating @ 47 F) = 60,000 Btu/h (63.28 MJ/h)
- Q (cooling @ 95 F) = 2.5 RT

Performance

- COP (heating @ 47 F) = 1.39
- COP (cooling @ 95 F) = 0.64

To achieve this performance with the EM engine, it was also necessary to operate the cooler at a reduced inlet coolant temperature of 32 F (0°C) as compared to a design criteria of 77 F (25°C). To overcome these shortfalls in performance, a second generation FPSE, the MARK I, was designed and fabricated. The design goals of the MARK I were to:

1. Develop an engine capable of achieving 3.0 RT of cooling at the 95 F (35°C) ambient conditions with a cooler temperature of 77 F (25°C)

2. Significantly reduce the electrical power required to drive the displacer

3. Achieve an engine efficiency of 25% as compared to the EM's efficiency of only 16.4%.

In the spring of 1986, the MARK I engine was assembled and run with the breadboard hydraulic transmission and refrigerant compressor for the first time. After several months of development, the module achieved its performance milestone goals of 3.0 RT of cooling and an engine efficiency of 25%.

TEST PROCEDURE

Laboratory testing was performed on a refrigerant desuperheated test loop that enables evaporator and condenser pressures to be set for any representative ambient temperature. The test points are given in Table 2; the ambient temperature is the outdoor temperature, and the pressures and temperatures represent the appropriate refrigerant conditions at the compressor.

Examples of the calculations used to determine the heating and cooling performance are given in Tables 3 through 7. As shown in Tables 3 and 4, the state-point enthalpies are calculated from the measured state-point properties; from the enthalpies and refrigerant mass flow, the heating and cooling capacity is determined. For the cooling capacity, this is:

$$Q \text{ (cooling capacity)} = \dot{m}\,(h_1 - h_3) \qquad (1)$$

where

$\dot{m}$ = Refrigerant mass flow rate
h_1 = Enthalpy of the refrigerant in the suction manifold prior to entering the compressor
h_3 = Enthalpy of the liquid refrigerant prior to the expansion

For the heating mode, the refrigerant heating capacity is:

$$Q \text{ (refrigerant heating capacity)} = \dot{m}\,(h_2 - h_3) \qquad (2)$$

where

$\dot{m}$ = Refrigerant mass flow rate
h_2 = Enthalpy of the refrigerant in the compressor discharge manifold prior to entering the condenser
h_3 = Enthalpy of the liquid refrigerant prior to the expansion

As shown in Table 5, the input energy to the heat pump power module is measured. The input energy consists of the fuel (natural gas) energy and the parasitic electrical power needed to drive the auxiliaries, displacer motor, and controls. As shown in Table 6, the engine-rejected heat is calculated from the measured temperature difference across the engine cooler and the coolant mass flow. For the heating mode, 85% of the engine-rejected heat is added to the refrigerant capacity to give the total heating capacity:

$$Q \text{ (heating capacity)} = \dot{m}\,(h_2 - h_3) + 0.85\,(Qrej) \tag{3}$$

where

0.85 = Recovery heat exchanger effectiveness

As shown in Figure 5, heat recovered from the engine does not include waste heat from the combustor and, therefore, for the cases considered, represents 60% of the overall rejected heat from the heat pump module. From the heating and cooling capacities, the heat pump performance is calculated as:

$$COP_{fuel} = \frac{\text{Heating/Cooling Capacity}}{FR} \tag{4}$$

where

FR = Combustor firing rate (input fuel energy) based on the higher heating value (HHV) of the fuel (HHV is assumed to be 1032 Btu/scf [38.44 MJ/m^3])

Also, a second parameter is defined to account for the electric parasitics:

$$COP_{El\,+\,fuel} = \frac{\text{Heating/Cooling Capacity}}{FR + El} \tag{5}$$

where

El = Electric parasitic power for the heat pump module in equivalent units

Table 7 presents the COPs as calculated for these two sample points.

Performance data for the breadboard unit were taken by establishing the evaporator and condenser conditions and the superheat in the suction stream. After the test loop and heat pump reached a steady-state operating point, data were recorded at several capacities by modulating the compressor piston stroke. Steady-state operation was defined as that point in time when none of the performance parameters of the total system (FPSE, compressor, and test loop) differs from data scan to data scan by more than 3%. In all, testing at each ambient condition involved about six hours to complete, and each ambient was run two to three times to verify repeatability.

Several other terms used to describe the performance of the FPSE/HP are hydraulic transmission loss, hydraulic transmission efficiency, compressor COP, and lower end efficiency. These terms are defined as follows:

1. Hydraulic Transmission Loss. The difference between the pressure-volume (PV) power measured in the engine and the PV power measured in the compressor cylinders.
2. Hydraulic Transmission Efficiency. The ratio of the PV power measured in the compressor divided by the PV power measured in the engine.
3. Compressor COP. The ratio of the heating and cooling effect produced by the compressor divided by the PV power measured in the engine.
4. Lower End Efficiency. The ratio of the compressor isentropic power divided by the PV power measured in the engine.

TEST RESULTS

Two series of tests were performed on the MARK I heat pump module. The first series, defined as verification testing, was performed to compare the performance of the MARK I unit to that of the EM unit. This involved testing the two free-piston Stirling engines with the same lower end (i.e., same hydraulic transmission and refrigerant compressor). The second series of tests was performed to measure the MARK I's performance at several heating and cooling ambient temperature conditions.

Verification testing consisted of the following:

1. Comparing the performance of the EM and MARK I at a similar operating point; however, the cooler temperature was raised for the MARK I testing.

2. Comparing the maximum refrigeration capacity at the 95 F (35°C) ambient temperature conditions for the two modules.

3. Comparing the heating performance at the 47 F (8.3°C) ambient temperature conditions for the two modules.

Performance tests consisted of measuring the MARK I heat pump module's performance at the 95 F (35°C), 47 F (8.3°C), and 17 F (-8.3°C) ambient temperature conditions. In addition, tests were performed to determine the capacity modulation control and efficiency of the module at the 47 F (8.3°C) ambient conditions.

The results of the verification testing are given in Tables 8, 9, and 10. Table 8 compares the performance of the EM and MARK I. This comparison was made at a similar cooling capacity of 2.3 RT. In the MARK I tests, however, the cooler temperature was raised to 104 F (40°C). A comparison of these results shows the performance improvements achieved with the MARK I:

1. MARK I efficiency improved by 43%, from 16.4% to 23.5%. This improvement manifested itself in a much lower displacer stroke and motor input power, even with a much higher cooler temperature.

2. The heat pump module's COP increased from 0.60 to 0.80. This improvement is a result of the improved MARK I engine efficiency, most notably:

   - Reduced internal gas leakages because of better control of the seal clearances
   - Improved heat transfer characteristics of the regenerator and heater head
   - Reduced losses in the displacer gas springs due to reduced internal leakages and lower gas spring pressure amplitudes.

Table 9 compares the performance of the EM and MARK I at the maximum capacity achieved with each module. The most significant improvements in this testing were the increase in capacity from 2.3 to 3.2 RT and the increase in COP from 0.60 to 0.86. It should also be noted that, at this point, the capacity is limited by the safe working stress in the diaphragm. The engine, as measured by the reduced displacer stroke, is capable of producing more power.

Table 10 compares the performance of the two modules in the heating mode. At the 47 F (8.3°C) ambient temperature conditions, the MARK I module's COP increased from 1.39 to 1.60 under similar operating conditions, and the displacer motor power decreased from 769.5 to 146.8 W.

The results of the performance tests are presented in Figures 6 and 7. Figure 6 shows the maximum capacity and COPs measured at the 95 F (35°C), 47 F (8.3°C), and 17 F (-8.3°C) ambient temperature conditions. In addition, as shown in both figures, the capacity at 47 F (8.3°C) was modulated down to the load line to determine the modulation capability of the MARK I module. Figure 7 shows that the performance of the module remains constant at 1.6 to a capacity of 20,000 Btu/h (21.1 MJ/h) before it falls off. Reasons for the falloff have not been investigated; however, the flatness of the curve to 20,000 Btu/h (21.1 MJ/h) is a good indication of the modulating capability of the module. This evidence suggests that, with further development, a high COP can be achieved over a very wide range of modulation.

SUMMARY AND CONCLUSIONS

The MARK I heat pump module was tested at the 17 F (-8.3°C), 47 F (8.3°C), and 95 F (35°C) ambient temperature conditions to verify its performance. The module performed very well, accumulating more than 180 hours of uninterrupted running time while successfully demonstrating the POC cooling targets of 3.0 RT and a heating COP of 1.5. The achievements of the MARK I module over the past year are shown in Figure 8. Testing has also shown that, by further reducing internal leakages in the engine, the MARK I's current efficiency of 25% can be raised to increase the cooling COP to 1.0.

The next step in the MARK I test program is to ship the heat pump module to Lennox Industries' Carrollton, Texas, test facilities for further validation of its performance and a comprehensive parametric performance mapping. This work will be performed during calendar year 1987.

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TABLE 1

MARK I FPSE/HP Operating Specifications

Free-Piston Stirling Engine	
Working Fluid	Helium
Mean Cyclic Pressure, Bar (Mpa)	60 (6.0)
Operating Frequency, Hz	58
Heater Head Average Temperature, F (°C)	1337 (725)
Cooler Coolant Inlet Temperature, F (°C)	77-104 (25-40)
Combustor Input Fuel	Natural Gas
Combustor Pressure Drop, in. (mm)	3.0 H_2O (<76.2 H_2O)
Hydraulic Transmission/Compressor	
Hydraulic Oil	Dow Corning 200 Silicone Fluid, 10 cs
Transmission Mean Operating Temperature, F (°C)	95 (35)
Refrigerant	Freon, R-22

TABLE 2

Test Conditions for R-22 Refrigerant

	Cooling, F (°C)			Heating, F (°C)			
	95 (35.0)	87 (30.6)	80 (26.7)	47 (8.3)	32 (0)	17 (-8.3)	0 (-17.8)
Discharge Pressure, psia (kPa)	277.4 (1913)	225.5 (1555)	196.5 (1355)	210.6 (1452)	202.1 (1393)	196.5 (1355)	183.1 (1262)
Saturation Temp., F (°C) (Condenser)	121 (49.4)	105 (40.6)	95 (35.0)	100 (37.8)	97 (36.1)	95 (35.0)	90 (32.2)
Suction Pressure, psia (kPa)	90.7 (625)	94.7 (653)	98.7 (681)	76.2 (525)	57.7 (393)	42.9 (296)	29.8 (205)
Saturation Temp., F (°C) (Evaporator)	45 (7.2)	47.5 (8.6)	50 (10)	35 (1.7)	20 (-6.7)	5 (-15)	-12 (-24.4)
Suction Temp., F (°C)*	55 (12.8)	57.5 (14.2)	60 (15.6)	45 (7.2)	30 (-1.1)	15 (-9.4)	-2 (-18.9)
Pressure Ratio	3.06	2.38	1.99	2.76	3.50	4.58	6.14

* Includes at least 10 F (+5.6°C) superheat

TABLE 3

Data and Computations for Heating/Cooling Rating Points: Refrigerant Conditions

Measured Performance Parameter	Ambient Temperature 95 F (35°C)	47 F (8.3°C)
Suction Gas		
Saturation Temperature, F (°C)	45.0 (7.2)	32.0 (0)
Pressure, psia (kPa)	90.7 (625)	72.96 (503)
Superheat Temperature, F (°C)	63.6 (17.5)	52.5 (11.4)
Discharge Gas		
Saturation Temperature, F (°C)	121.0 (49.4)	100.0 (37.8)
Pressure, psia (kPa)	275.7 (1901)	211.3 (1459)
Superheat Temperature, F (°C)	179.9 (82.2)	167.5 (75.3)
Liquid Refrigerant Temperature Entering Expansion Valve*, F (°C)	106.0 (41.1)	85.0 (29.4)

* Estimated

TABLE 4

Data and Computations for Heating/Cooling Rating Points: Refrigerant Energy Quantities

Calculated Performance Parameter	Ambient Temperature 95 F (35°C)	47 F (8.3°C)
Refrigerant Mass Flow*, lbm/h (kg/h)	384.4 (174.3)	417.2 (189.2)
Enthalpy, Btu/lbm (MJ/kg)		
• Evaporator: Inlet (h_3)	41.17 (0.096)	34.63 (0.081)
Outlet (h_1)	111.86 (0.260)	110.89 (0.258)
• Condenser: Inlet (h_2)	126.37 (0.294)	126.26 (0.294
Saturated Liquid	46.04 0.107)	39.27 (0.091)
Heating/Cooling Capacity, Btu/h (MJ/h)	27,173.2 (28.66)	38,228.0 (40.32)

* Measured

TABLE 5

Data and Computations for Heating/Cooling Rating Points: Input Energy

Input Energy Parameter	Ambient Temperature 95 F (35°C)	47 F (8.3°C)
Gas Fuel Flow*, scfm (m^3)	33.0 (0.93)	35.4 (1.00)
Btu/h @ 1032 Btu/scf** (HHV) (MJ/h @ 38.44 MJ/m^3)	34,019 (35.75)	36,489 (38.44)
Auxiliary Power, (W)		
- Combustion Blower*	100.0	100.0
- Coolant Pump**	150.0	150.0
- Displacer Motor*	248.5	209.0
- Controls*	50.0	50.0

* Measured
** Estimated

TABLE 6

Data and Computations for Heating/Cooling Rating Points:
Engine Rejected Heat

Calculated Performance Parameter	Ambient Temperature 95 F (35°C)	47 F (8.3°C)
Engine Cooling Water		
• Flow (M), gpm (l/min)	2.6 (9.84)	2.6 (9.84)
• Temperature (M), F (°C)		
- Inlet	97.5 (36.4)	90.7 (32.6)
- Outlet	111.4 (44.1)	106.5 (41.4)
Engine Rejected Heat, Btu/h (MJ/h)	18,227.8 (19.22)	20,879.4 (22.02)
Heating/Cooling Capacity, Btu/h (MJ/h)	27,173.2 (28.66)	55,975.5* (59.04)

*60% rejected heat recovery (85% refrigerant heat recovery)
M = Measured (no combustor heat recovery)

TABLE 7

Data and Computations for Heating/Cooling Rating Points:
Heat Pump Performance

Calculated Performance Parameter	Ambient Temperature 95 F (35°C)	47 F (8.3°C)
Heating/Cooling Capacity, Btu/h (MJ/h)	27,173.2 (28.66)	55,975.5 (59.04)
COP (fuel)	0.80	1.53
COP (electric and fuel)	0.76	1.46

TABLE 8

MARK I Heat Pump Performance: Raised Cooler Temperature

(Ambient Temperature 95 F (35°C); Suction Superheat 15 F (8.3°C); Liquid Subcooling 15 F (8.3°C))

Parameter	Breadboard EM#4 (1-7-86)	MARK I (9-23-86)
Displacer Stroke, mm	21.9	16.2
Piston Stroke, mm	16.2	16.6
Firing Rate, HHV, W	13,511.0	9,976.2
Engine PV Power, W	2,214.0	2,350.1
Displacer Motor Power, W	669.3	248.5
Average Cooler Temperature, °C	5.0	40.0
Capacity (Cooling), MJ/h	29.20	28.65
Engine Efficiency, %	16.4	23.5
Lower End Efficiency, %	66.0	63.3
Compressor COP	3.67	3.53
Heat Pump Module COP (Fuel), HHV	0.60	0.80
Heat Pump Module COP (El* & Fuel), HHV	0.56	0.76

* Includes displacer power and 300-W parasitics for blower, pump, and controls, as shown in Table 5.

TABLE 9

MARK I Heat Pump Performance: High Cooling Capacity Comparison

(Ambient Temperature 95 F (35°C); Suction Superheat 15 F (8.3°C); Liquid Subcooling 15 F (8.3°C))

Parameter	Breadboard EM#4 (1-7-86)	MARK I (9-18-86)
Displacer Stroke, mm	21.9	17.7
Piston Stroke, mm	16.2	18.2
Firing Rate, HHV, W	13,511.0	13,013.0
Engine PV Power, W	2,214.0	3,036.1
Displacer Motor Power, W	669.3	431.3
Capacity (Cooling), Btu/h	-27,681.0	-38,283.0
(MJ/h)	(29.20)	(40.38)
Engine Efficiency, %	16.4	23.3
Compressor COP	3.67	3.70
Heat Pump Module COP (Fuel), HHV	0.60	0.86
Heat Pump Module COP (El* & Fuel), HHV	0.56	0.82

* Includes displacer power and 300-W parasitics for blower, pump, and controls, as shown in Table 5.

TABLE 10

MARK I Heat Pump Performance: High Heating Capacity Comparison

(Ambient Temperature 95 F (35°C); Suction Superheat 15 F (8.3°C); Liquid Subcooling 15 F (8.3°C))

Parameter	Breadboard EM#4 (1-7-86)	MARK I (9-16-86)
Displacer Stroke, mm	22.0	15.0
Piston Stroke, mm	17.2	17.5
Firing Rate, HHV, W	13,118.0	10,840.0
Engine PV Power, W	2,203.0	2,148.0
Displacer Motor Power, W	769.5	146.8
Capacity (Heating), MJ/h	65.76	62.74
Compressor COP	4.92	4.89
Heat Pump Module COP (Fuel), HHV	1.39	1.61
Heat Pump Module COP (El* & Fuel), HHV	1.30	1.54

* Includes displacer power and 300-W parasitics for blower, pump, and controls.

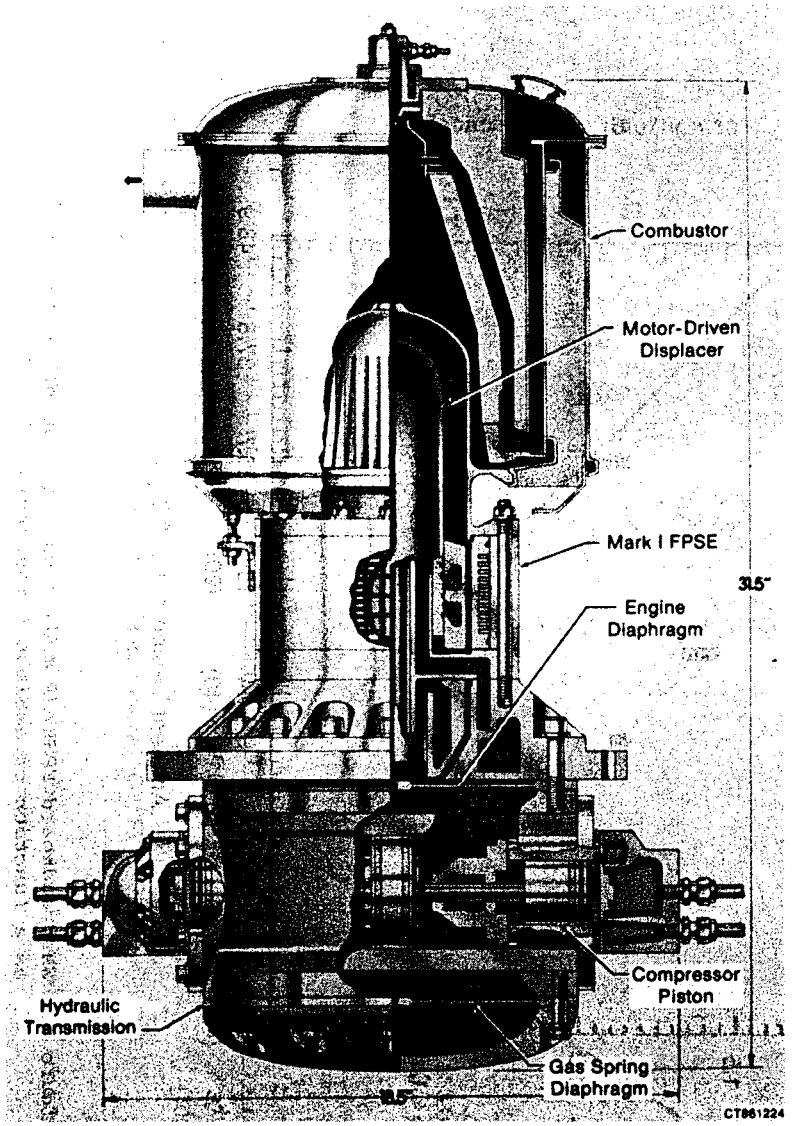

Figure 1. MARK I free-piston Stirling engine/heat pump module

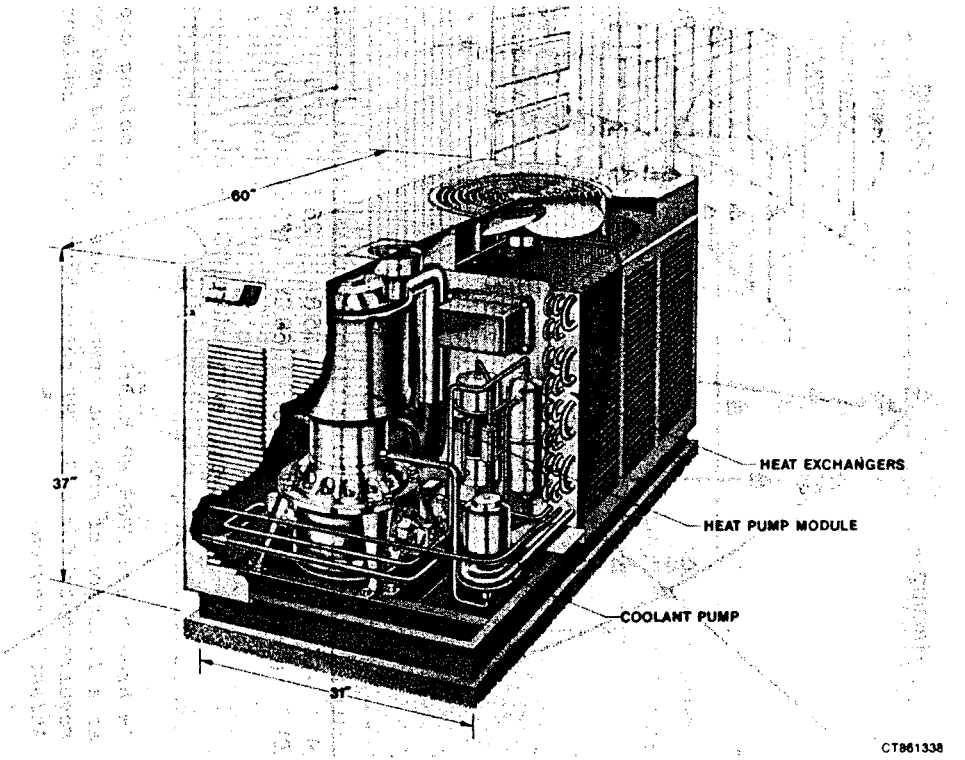

Figure 2. Free-piston Stirling engine heat pump

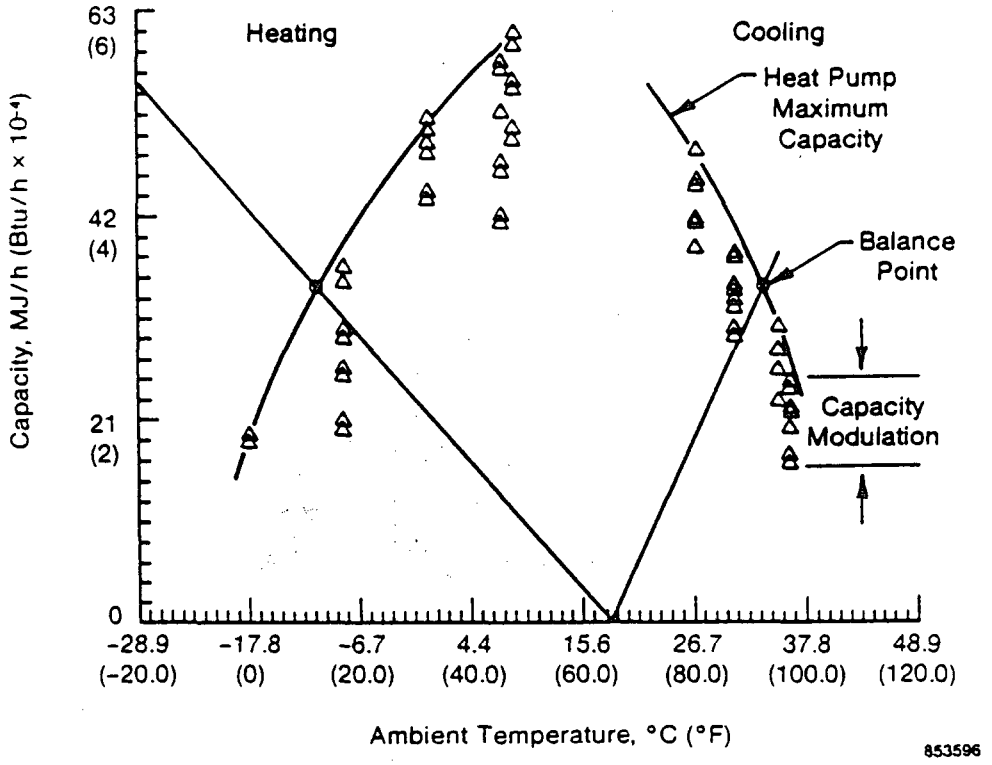

Figure 3. EM breadboard FPSE/HP module heating and cooling capacity as a function of ambient temperature (heating and cooling requirements for a 3.0 RT rated home at 95 F ambient in mid-Atlantic region)

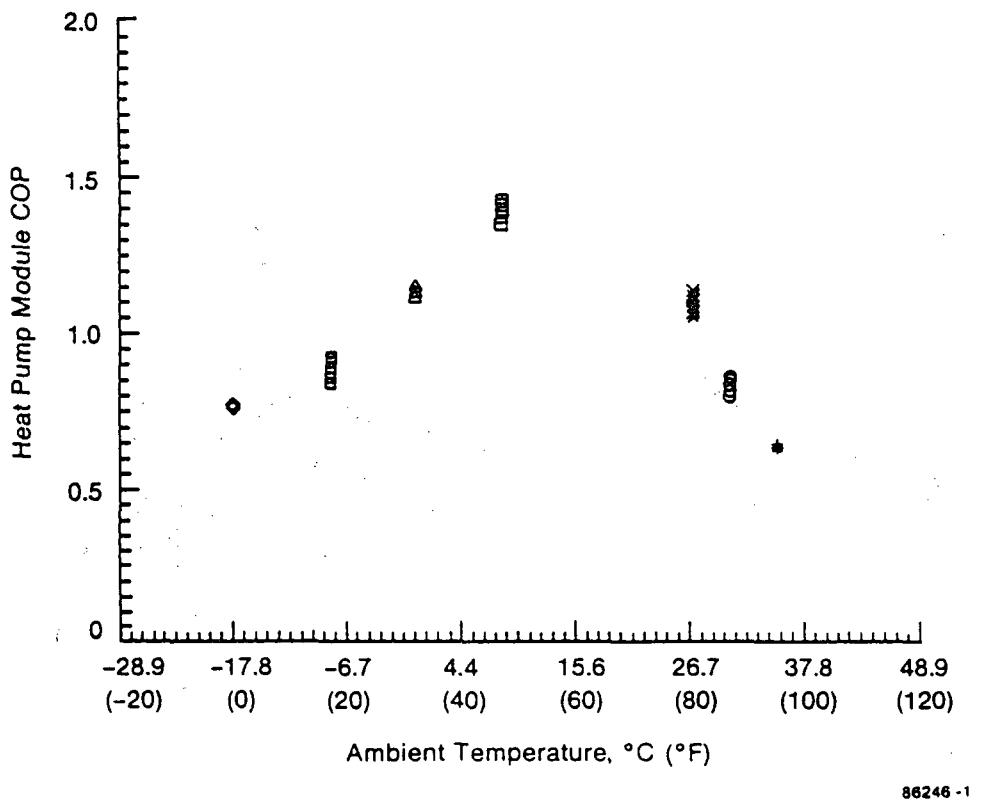

Figure 4. EM breadboard FPSE/HP module coefficient of performance as a function of ambient temperature

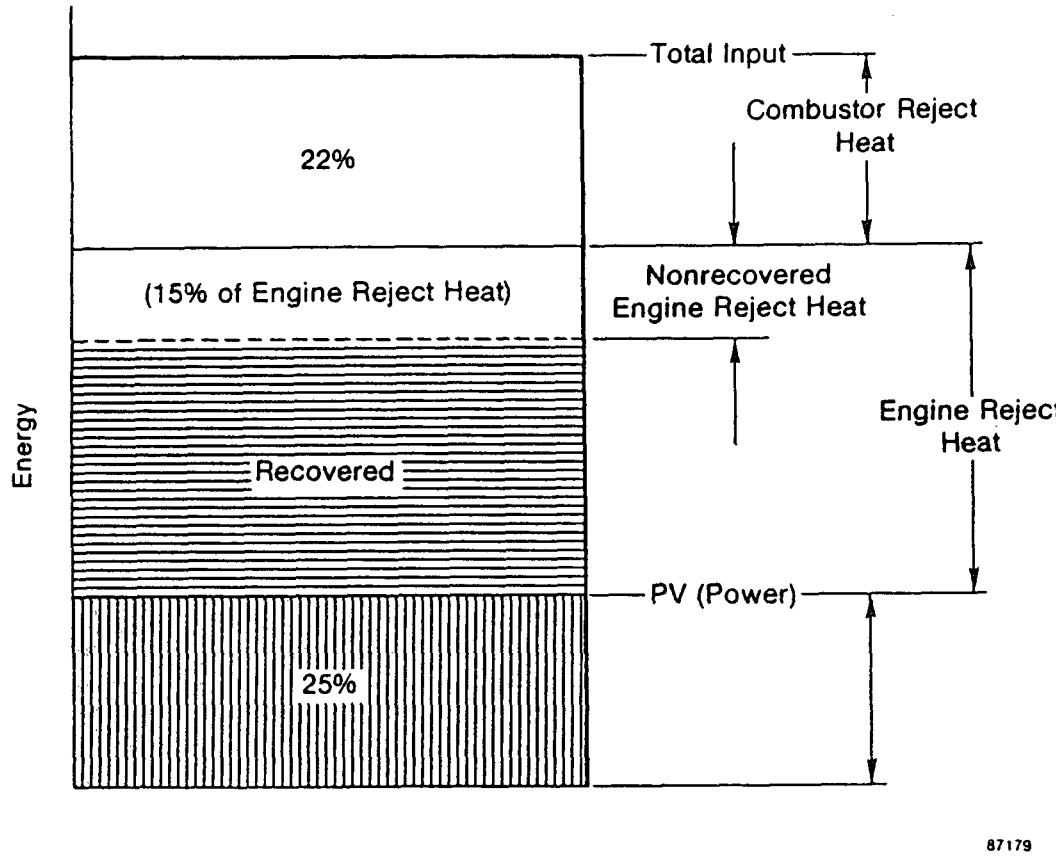

Figure 5. Engine heat recovery

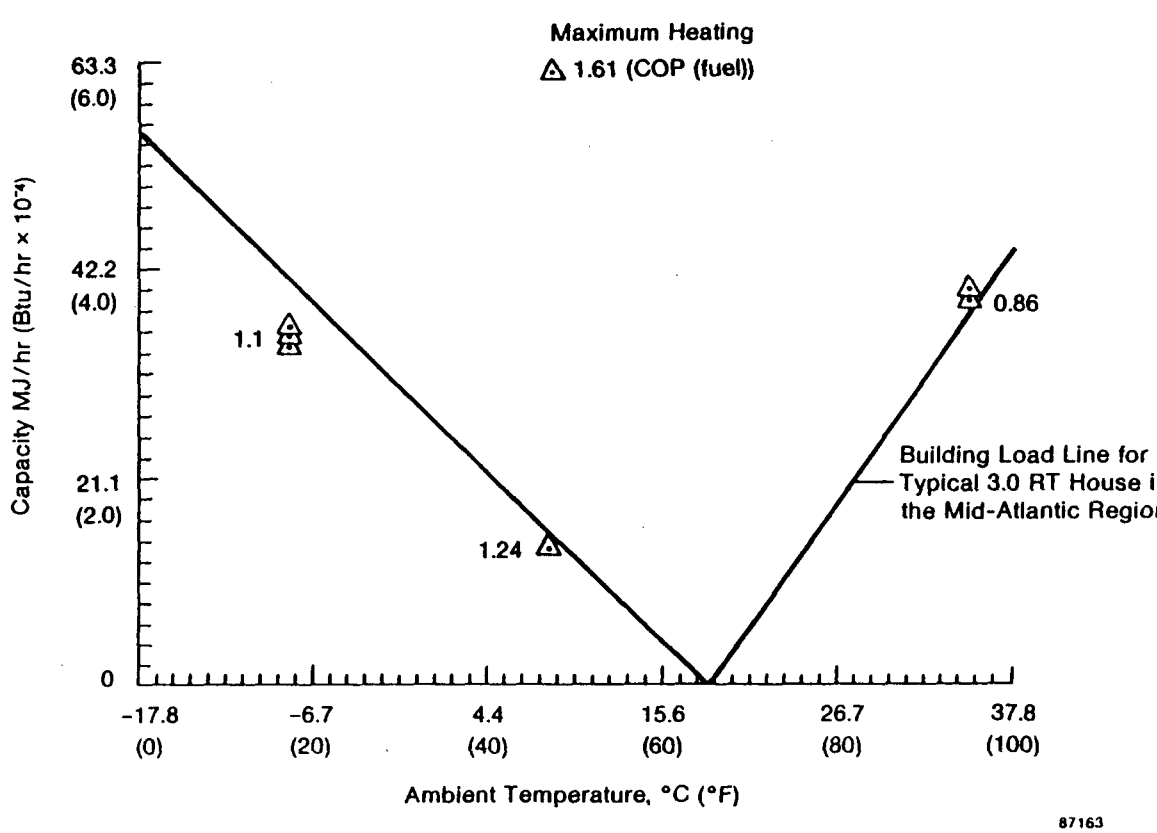

Figure 6. MARK I FPSE heat-actuated heat pump module heating/cooling capacities and COPs (September 1986)

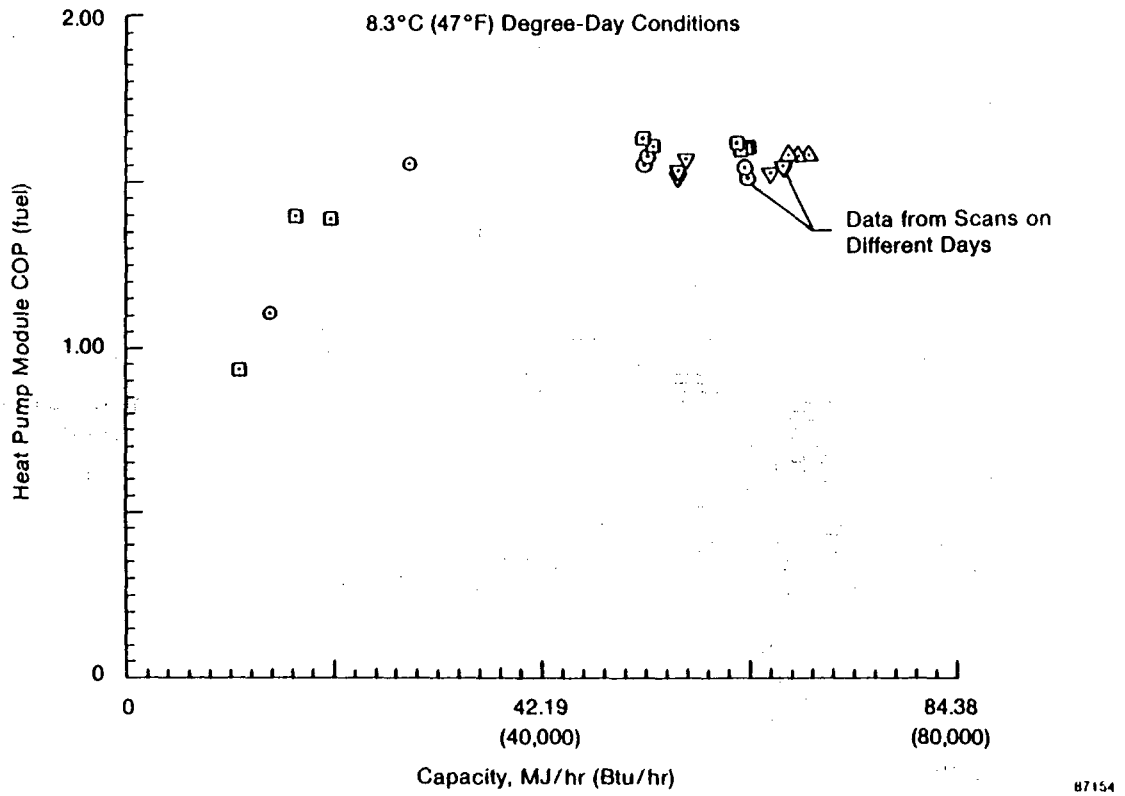

Figure 7. MARK I heat pump module capacity modulation performance results (steady-state tests at different compressor piston strokes)

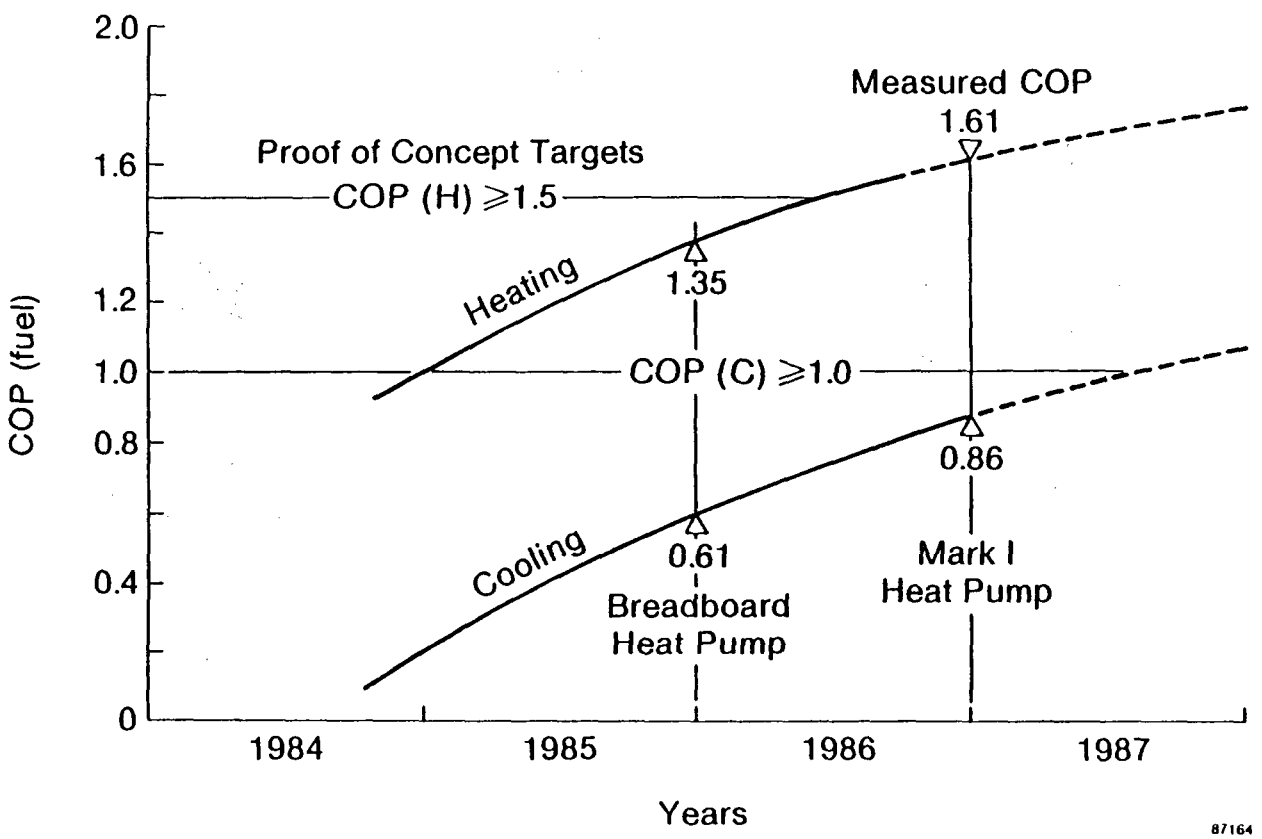

Figure 8. Free-piston heat pump performance targets and measured performance