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# AN UPDATE OF FREE-PISTON STIRLING ENGINE HEAT PUMP DEVELOPMENT

**R.A. Ackermann, P.E., Ph.D.** **J.M. Clinch, Ph.D.** **G.T. Privon**
*ASHRAE Member*

## ABSTRACT

A Free-Piston Stirling Engine Heat Pump (FPSE/HP) for residential applications has been under development for the past five years. The system consists of a natural gas combustor, free-piston Stirling engine, and a variable-stroke resonant piston refrigerant compressor. The compressor is linked to the engine via a unique hydraulic transmission that provides for both efficient power transfer and hermetic sealing between the engine working fluid (helium) and the compressor refrigerant.

This development effort has led to a breadboard heat pump power module, engine/transmission/compressor, that has undergone a comprehensive test program to evaluate the performance of an FPSE/HP and to judge its potential for further development. The results obtained from this testing are presented in this paper.

## 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% efficiency) furnace. In the past, the FPSE/HP has always been judged to have the potential for high efficiency; however, developers were unsuccessful in demonstrating either stable operation or adequate performance. The recent success in demonstrating both stable operation over a broad range of ambient operating conditions and good performance with the breadboard FPSE/HP provides a comprehensive data base for further evaluations.

Laboratory tests have indicated that a steady-state heating coefficient of performance (COP) of 1.6 and a cooling COP of 0.9 (including combustion losses but excluding electric parasitics) can be achieved by the gas heat pump. In addition, tests have shown that the laboratory heat pump is capable of producing 31.6 MJ/h (2.5 tons) of cooling at the design point of a 35°C (95°F) ambient temperature. The target capacity for the FPSE/HP is 37.4 MJ/h (3.0 tons).

The development of the heat pump has been under the joint sponsorship of the Gas Research Institute and the Department of Energy. The objective of this effort was to take an existing FPSE and couple it to a resonant refrigerant compressor to demonstrate performance. This paper chronicles the developmental history of the FPSE/HP and evaluates system performance and capacity data to determine the unit's present performance and future potential.

## FPSE/HP DESCRIPTION

In 1980, the first general-use FPSE was designed. This engine, named the Engineering Model (EM), provided the building block for the design of a breadboard heat pump. It was felt that several

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Dr. R.A. Ackermann, Manager, Free-Piston Engine Program, Mechanical Technology Incorporated, Latham, NY; Dr. J.M. Clinch, Heat Pump Program Manager, Gas Research Institute, Chicago, IL; and G.T. Privon, Program Manager, Oak Ridge National Laboratory, Oak Ridge, TN.

unique features of the EM engine over previous FPSEs would provide the potential for achieving high reliability and long life in an FPSE/HP. These included the following:

-- Resonant dynamics and cyclic operation at 60 Hz, which provides quiet operation and eliminates the need for all mechanical linkages on moving elements

-- Power modulation through a simple engine stroke control scheme that provides easy power modulation and, therefore, refrigerant flow control

-- Cast, monolithic heater head that eliminates the need for an elaborate, brazed, multitubed heater head

-- Gas-lubricated hydrostatic bearings and close-clearance seals that eliminate all rubbing and wear in the engine.

The development of the FPSE/HP was achieved by matching the EM engine to a resonant piston refrigerant compressor. This unit is shown schematically in Figure 1. It consists of an FPSE that is thermodynamically and dynamically coupled to the resonant piston refrigerant compressor. The unit is designed to oscillate at a prescribed frequency by matching the displacer and compressor piston masses to their respective gas spring stiffness. Power is developed in the FPSE by selecting the mass and spring components to give the proper dynamic phasing between the two moving elements (displacer and power piston).

Power is extracted from the engine by the compressor through a force link that dynamically connects the compressor piston to the engine's power piston. In the schematic, this link is represented by a rigid mechanical connection between the two elements.

In the heat pump application, the broad range of heating and cooling ambient conditions imposed on the refrigerant must be accommodated in the dynamic design and control of the FPSE/HP, if stable operation is to be achieved. Two unique features of the current design are the matching of the compressor to the FPSE and its control, and the development of a hydraulic transmission that provides an efficient force link and power coupling between the engine and compressor. The hydraulic transmission also provides a hermetic separation of the refrigerant and engine working fluid (helium). These features have been demonstrated in a breadboard FPSE/HP that tested successfully over a range of ambient temperature conditions from -17.8 to 35°C (0 to 95°F). Over this entire range, the FPSE/HP operated stably, without incident, for more than 150 hours.

The actual breadboard heat pump power module is shown in Figure 2. It consists of the FPSE, 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 turned at every operating point by varying the operating frequency and the heating and cooling capacity 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. This separation is shown in the compressor layout drawing given in Figure 2. 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 2, 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, that 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, thus 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 (Figure 2). 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 breadboard FPSE/HP are given in Table 1. The engine used 6.0 MPa (60 bar) helium as the working fluid, the hydraulic transmission used a Dow Corning silicone-based oil with a viscosity of 10 cs and the refrigerant used was Freon R-22. During testing, the engine was typically run with a heater head temperature of 725°C (1337°F), a cooler inlet water temperature of 0°C (32°F), and a laboratory ambient temperature of 25°C (77°F).

Prior to running the breadboard heat pump in the configuration shown in Figure 2, an earlier version of the hydraulic transmission and compressor was tested. This unit, shown in Figure 3, differed from the breadboard unit (or direct-drive version of the hydraulic transmission as it was termed) by the incorporation of an internal counterweight. The internal counterweight was designed into the hydraulic transmission to eliminate the transverse vibrations produced by the horizontally mounted compressor piston. The operation of the counterweight is similar to the direct-drive version previously shown, in that, as the engine diaphragm deflects downward, it displaces oil in volume A, which moves the counterweight to the left. The counterweight motion to the left, in turn, produces two effects. The first is the displacement of oil in volume B, which moves the gas spring diaphragm down into the gas spring cavity. The second effect is that the motion of the counterweight displaces oil in volume D, moving the compressor piston to the right, thereby maintaining constant oil volumes in cavities C and D. As described in the following sections, the internal counterweight was effective in producing a vibration-free compressor. However, the performance of the hydraulic transmission with the counterweight was too poor to pursue further, and the hydraulic transmission was converted to the direct-drive configuration shown in Figure 2.

## TEST PROCEDURE

Laboratory testing has been performed on a refrigerant desuperheater test loop that enables evaporator and condenser pressures to be set for a representative ambient temperature. The test points are given in Table 2, where the ambient temperature is the outdoor temperature and the pressures and temperatures represent the appropriate refrigerant conditions. The two test points refer to the standard heating and cooling rating points. Performance is calculated from the refrigerant state points described in Figure 4. Heating and cooling capacities are calculated as follows.

$$Q(\text{Cooling Capacity}) = \dot{m}(h_1 - h_6) \tag{1}$$

where

$\dot{m}$ = refrigerant mass flow rate
$h_1$ = enthalpy of the refrigerant in the suction manifold
$h_6$ = enthalpy of the liquid refrigerant prior to the expansion.

$$Q(\text{Heating Capacity}) = \dot{m}(h_2 - h_6) + 0.85\,(Q_{rej}) \tag{2}$$

where

$h_2$ = enthalpy of the refrigerant in the compressor discharge manifold
$Q_{rej}$ = the rejected engine heat measured from the engine coolant (0.85 represents the recovery heat exchange effectiveness).

The steady-state COP for the FPSE/HP system is calculated as follows.

$$\text{System COP} = \frac{\text{Heating/Cooling Capacity}}{\text{FR}} \tag{3}$$

where

FR = the combustor firing rate based on the higher heating value of the fuel.

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) differ from data scan to data scan by more than 3%. In all, testing at each ambient condition involved about six hours of running 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 the hydraulic transmission loss, the hydraulic transmission efficiency, the compressor COP, and the compressor isentropic efficiency. These terms are defined as follows.

-- Hydraulic Transmission Loss: the difference between the pressure-volume (PV) power measured in the engine and the PV power measured in the compressor cylinders.

-- Hydraulic Transmission Efficiency: the ratio of the PV power measured in the compressor divided by the PV power measured in the engine.

-- Compressor COP: the ratio of the heating and cooling effect produced by the compressor divided by the PV power measured in the engine.

-- Compressor Isentropic Efficiency: the ratio of the isentropic work of compressing the refrigerant divided by the PV power measured in the compressor cylinders. The isentropic work is defined as the product of the measured refrigerant flow rate times the isentropic enthalpy change across the compressor. This is shown in Figure 4 by the dashed isentropic compression line.

## TEST RESULTS

The breadboard FPSE/HP with the original internal counterweight transmission was run for the first time in late spring of 1983. The unit performed very poorly during this initial testing, producing less than 12.7 MJ/h (1.0 ton) of cooling capacity at the 35°C (95°F) ambient condition. Table 3 presents the results of this initial testing and provides a comparison of the test results with the original design parameters. As shown, due to the higher-than-predicted losses in the hydraulic transmission and compressor, the compressor piston was prevented from achieving its design stroke, and the refrigerant flow was less than 40% of the predicted value. The explanation for the reduction in capacity may be seen from Figure 5, which gives the load curve for the hydraulic transmission and compressor and the output power characteristics of the EM FPSE. The intersection of these two curves represents the operating point of the system, where the load generated by the hydraulic transmission and compressor matches the PV power produced by the engine. The result of the increased losses in the hydraulic transmission is seen to raise the load curve of the hydraulic transmission and compressor so that the intersection of the two curves occurs at a much lower piston stroke and, consequently, a lower power, i.e., the engine cannot develop enough starting power to overcome the load and achieve its rated capacity. This result is analogous to a rotating machine in which the load prevents the motor from reaching its rated rpm and, therefore, the motor cannot achieve its rated output.

A thorough evaluation of the original hydraulic transmission and compressor determined that the counterweight in the hydraulic transmission was the main contributor to the high losses. As shown in Figure 3, the four seals formed by the counterweight between volumes A, B, C, and D and the serpentine flow passages created by the counterweight were judged to be the factors contributing to the large loss. After several analytical and experimental evaluations, it was concluded in December 1984 that to improve the breadboard FPSE/HP's performance, the counterweight would have to be eliminated. An important element of the FPSE/HP design was that it was a fully dynamically based system with the counterweight canceling the out-of-balance inertia force produced by the compressor piston. This consideration led to the development of an external vibration absorber that could be conveniently mounted to the compressor housing and would produce no additional losses in the hydraulic transmission.

In May 1985, the tuned vibration absorber was assembled and mounted on the compressor case of the direct-drive unit. Following checkout tests for natural frequency and slight retuning for 58-Hz operation, the absorbers were tested over a frequency range from 53 to 57 Hz at constant piston stroke equal to 14 mm (0.55 in.). Performance was excellent over the entire frequency range, with combustor lateral accelerations measured at less than 0.3 g (combustor acceleration remains low even when perfect bottom-end balance is not achieved because the motion that results from unbalance occurs as rotation about a point near the top of the combustor).

The results obtained for tests conducted at the 35°C (95°F) ambient conditions were:

-- Lower end losses were reduced by more than 70%. This result is shown in Figure 6, which gives the hydraulic transmission loss as a function of piston stroke. As shown, the total hydraulic transmission loss with the internal counterweight was 930 watts at a stroke of 16 mm (0.63 in.), and the equivalent loss for the direct-drive unit (without the internal counterweight) was 315 watts.

-- The hydraulic transmission efficiency measured at the 35°C (95°F) ambient condition was 87%. Figure 7 presents the measured hydraulic transmission efficiency as a function of compressor piston stroke. As shown, a transmission efficiency in the neighborhood of 85% was achieved over a broad operating range.

-- The reduction in the hydraulic transmission loss significantly reduced the rise in the hydraulic oil temperature, enabling better steady-state operation to be achieved. This is shown in Figures 8 and 9, which give the oil temperature rise at the 35°C (95°F) ambient operating point. With the internal counterweight, the temperature never reached a steady-state limit, nor did the rise show any indication of leveling off. For the direct-drive compressor, a steady-state oil temperature was achieved after 1.5 hours.

The overall effect of these improvements on the performance of the breadboard FPSE/HP is shown in Table 4. The performance improvement is seen by comparing the results to those obtained with the internal counterweight. The improvement is shown by the increases in refrigeration capacity to 29,457 Btu/h, hydraulic transmission efficiency to 87%, and compressor COP to 3.58.

The performance test results for the direct-drive unit are shown in Figures 10 and 11. Figure 10 presents the heating and cooling capacities measured relative to a typical four-bedroom house located in the mid-Atlantic region. As shown, the balance point for the breadboard FPSE/HP occurs at -11.1°C (12°F) and 33.3°C (92°F) for the heating and cooling loads, respectively. At the 35°C (95°F) ambient temperature, the cooling capacity of the unit is 31.6 MJ/h (2.5 tons) of refrigeration.

The measured compressor COP is given in Figure 11. The compressor COP ran from a high of 6.6 at 26.7°C (80°F) to 2.0 at -17.8°C (0°F). Comparing the breadboard compressor COP to an equivalently defined COP for an advanced electric heat pump compressor, we see that the breadboard hydraulic transmission and compressor are achieving comparable performance to an advanced electric heat pump compressor's performance. The advanced electric heat pump's COP is defined as the heating and cooling effect divided by the compressor shaft power, i.e., the electrical input power less the motor electrical losses and the crank drive mechanical losses.

Figure 12 gives the system COP measured for the breadboard FPSE/HP and represents baseline values for the heat pump using the EM/FPSE. This engine was not originally designed for use with the hydraulic transmission and compressor, and its adaptation to the breadboard system introduced significant losses. Work now in progress on the development of a second-generation FPSE is predicted to raise the cooling COP at 35°C (95°F) to 0.9 and the heating COP at 8.3°C (47°F) to 1.6. This second-generation FPSE/HP is scheduled to be operational during 1986.

## CONCLUSION

The breadboard FPSE/HP has performed well, attaining over 150 hours of uninterrupted laboratory operation. During this time, the compressor has demonstrated good performance, and valid baseline performance data have been obtained. With the experience gained from the breadboard unit and its performance data, the specifications of a second-generation FPSE and the criteria for mating an FPSE to a resonant refrigerant compressor have been established. Most significantly, this work has shown that by employing displacer control, the dynamics of an FPSE/HP may be optimized over a broad range of ambient temperatures, and the system made to operate stably during the steady-state and transient operating modes. In addition, by controlling the displacer stroke, it was demonstrated that a wide range of heating and cooling capacity modulation could be achieved.

This work has also demonstrated that the hydraulic transmission does provide an efficient power coupling between the FPSE and compressor and that the use of metallic diaphragms to provide a hermetic separation between fluids is reliable and effective. The testing has shown that a hydraulic efficiency greater than 85% is readily achievable, and that the performance of the hydraulic transmission does not degrade over a wide range of heating and cooling capacity modulation.

The next step in the program will be to mate this concept of a hydraulic transmission and resonant compressor to an FPSE that is designed specifically for this application. The goal will be to demonstrate a 37.4 MJ/h (3.0 tons) refrigeration capacity at 35°C (95°F) and a heating COP of 1.6 at 8.3°C (47°F) and 0.9 at 35°C (95°F).

TABLE 1
Breadboard FPSE/HP Operating Specifications

| | | |
|---|---|---|
| **Free-Piston Stirling Engine** | | |
| Working Fluid | | Helium |
| Mean Cyclic Pressure | MPa (bar) | 6.0 (60) |
| Operating Frequency | Hz | 58 |
| Heater Head Average Temperature | °C (°F) | 725 (1337) |
| Average Cooler Temperature | °C (°F) | 0 (32) |
| Combustor Input Fuel | | Natural Gas |
| Combustor Pressure Drop | mm (in.) | < 76.2 (3.0) $H_2O$ |
| **Hydraulic Transmission/Compressor** | | |
| Hydraulic Oil | | Dow Corning 200 Silicone Fluid, 10 cs |
| Transmission Mean Operating Temperature | °C (°F) | 35 (95) |
| Refrigerant | | Freon, R-22 |

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TABLE 2
Test Conditions for R-22 Refrigerant

| Ambient Temperature | | Cooling, °C (°F) | | | | Heating, °C (°F) | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | **35.0 (95)** | **30.6 (87)** | **26.7 (80)** | **Test Pt 2** | **8.3 (47)** | **0 (32)** | **−8.3 (17)** | **−17.8 (0)** | **Test Pt 1** |
| Discharge Pressure | kPa (psia) | 1913 (277.4) | 1555 (225.5) | 1355 (196.5) | 2148 (311.5) | 1452 (210.6) | 1393 (202.1) | 1355 (196.5) | 1262 (183.1) | 1662 (241.0) |
| Saturation Temperature (Condenser) | °C (°F) | 49.4 (121) | 40.6 (105) | 35 (95) | 54.4 (130) | 37.8 (100) | 36.1 (97) | 35 (95) | 32.2 (90) | 43.3 (110) |
| Suction Pressure | kPa (psia) | 625 (90.7) | 653 (94.7) | 681 (98.7) | 625 (90.7) | 525 (76.2) | 393 (57.7) | 296 (42.9) | 205 (29.8) | 479 (69.6) |
| Saturation Temperature (Evaporator) | °C (°F) | 7.2 (45) | 8.6 (47.5) | 10 (50) | 7.2 (45) | 1.7 (35) | −6.7 (20) | −15 (5) | −24.4 (−12) | −1.1 (30) |
| Suction Temperature | °C* (°F) | 12.8 (55) | 14.2 (57.5) | 15.6 (60) | 12.8 (55) | 7.2 (45) | −1.1 (30) | −17.2 (15) | −18.9 (−2) | 4.4 (40) |
| Pressure Ratio | | 3.06 | 2.38 | 1.99 | 3.43 | 2.76 | 3.50 | 4.58 | 6.14 | 3.46 |

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

TABLE 3
Breadboard FPSE/HP Performance
with Internal Counterweight

| | | Design | 9/1/83[1] |
|---|---|---|---|
| Refrigerant | | R-22 | R-22 |
| Transmission Fluid | | — | Suniso 3GS |
| Displacer Stroke | mm (in.) | 20.84 (0.82) | 19.51 (0.77) |
| Piston Stroke | mm (in.) | 19.05 (0.75) | 10.68 (0.42) |
| Displacer Phase Angle | ° | 69.3 | 56.71 |
| Mean Pressure | MPa (bar) | 6 (60) | 5.6 (56.3) |
| Frequency | Hz | 60 | 57.1 |
| Suction Pressure | KPa (psia) | 625 (90.7) | 632 (91.7) |
| Suction Temperature | °C (°F) | 7.2 (45.0) | 40.2 (104.4) |
| Discharge Pressure | KPa (psia) | 1913 (277.4) | 1929 (279.9) |
| Discharge Temperature | °C (°F) | — | 95.3 (203.5) |
| Refrigerant Flow | g/s (lbm/h) | 72 (570) | 27.3 (216.8) |
| Capacity (Cooling at 35°C (95°F)) | MJ/h (Btu/h) | -38.9 (-36,000) | -15.4 (-14,608) |
| Displacer Motor Power | watts | 313.0 | 527.1 |
| Engine PV Power | watts | 2,907 | 1,191 |
| Total Hydraulic Transmission Loss | watts | 658.0 | 598.1 |

[1]Base run used to establish performance at the end of the Phase I program, September 30, 1984

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TABLE 4
Comparison of Breadboard FPSE/HP
with and without Internal Counterweight

| Parameter | | Measured Data (R-22) | |
|---|---|---|---|
| | | With Counterweight | Without Counterweight |
| Mean Pressure | MPa (bar) | 6.2 (62.1) | 6.0 (59.6) |
| Frequency | Hz | 58.0 | 56.9 |
| Head Temperature | °C (°F) | 710 (1310) | 715 (1319) |
| Counterweight Stroke | mm (in.) | 15.25 (0.60) | N/A |
| Piston Stroke | mm (in.) | 15.25 (0.60) | 16.43 (0.65) |
| Displacer Stroke | mm (in.) | 21.73 (0.86) | 21.71 (0.85) |
| Suction Pressure | kPa (psia) | 618 (89.6) | 602 (87.3) |
| Discharge Pressure | kPa (psia) | 1933 (280.3) | 1928 (279.6) |
| Suction Temperature | °C (°F) | 8.1 (46.6) | 6.7 (44.1) |
| Freon Flow | g/s (lbm/h) | 43 (342) | 55.4 (439) |
| Refrigeration Capacity (Cooling at 35°C (95°F)) | MJ/h (Btu/h) | -24.3 (-23,000) | -31.1 (-29,457) |
| Hydraulic Transmission Loss | watts | 911.5 | 278 |
| Engine PV Power | watts | 2512 | 2413 |
| Displacer Motor Power | watts | 488 | 504 |
| Compressor Isentropic Efficiency | % | 71 | 78 |
| Hydraulic Transmission Efficiency | % | 63 | 87 |
| Compressor COP | | 2.69 | 3.58 |

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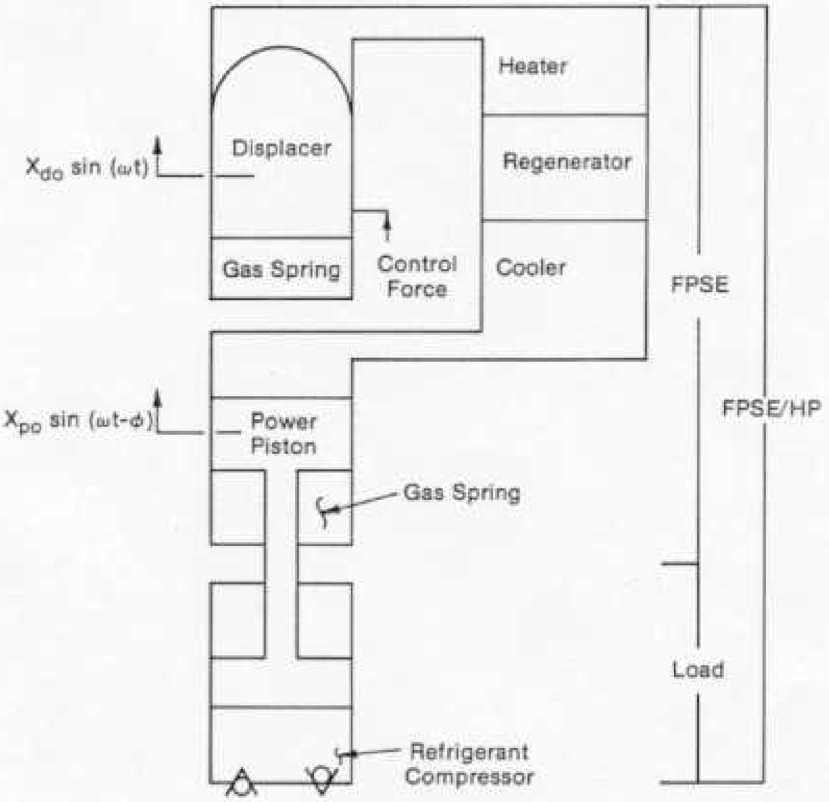


*Figure 1. Schematic FPSE and compressor load*

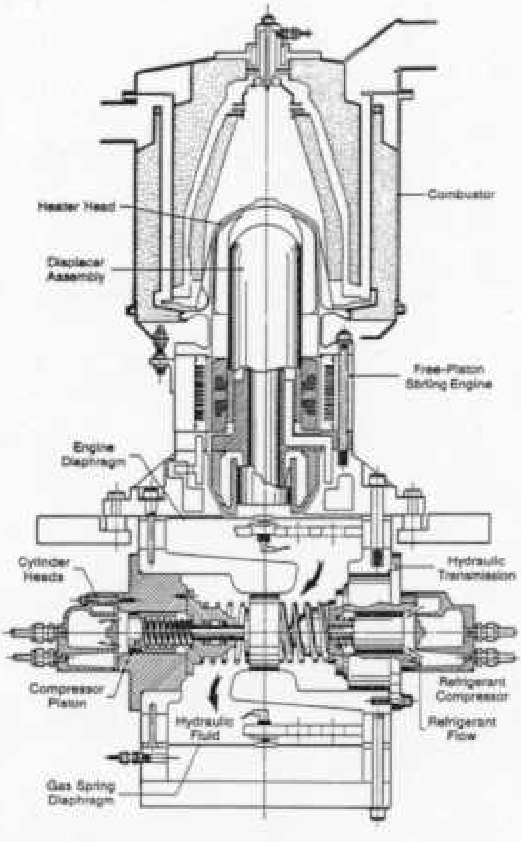


*Figure 2. FPSE heat pump power module with new direct-drive hydraulic transmission and compressor*

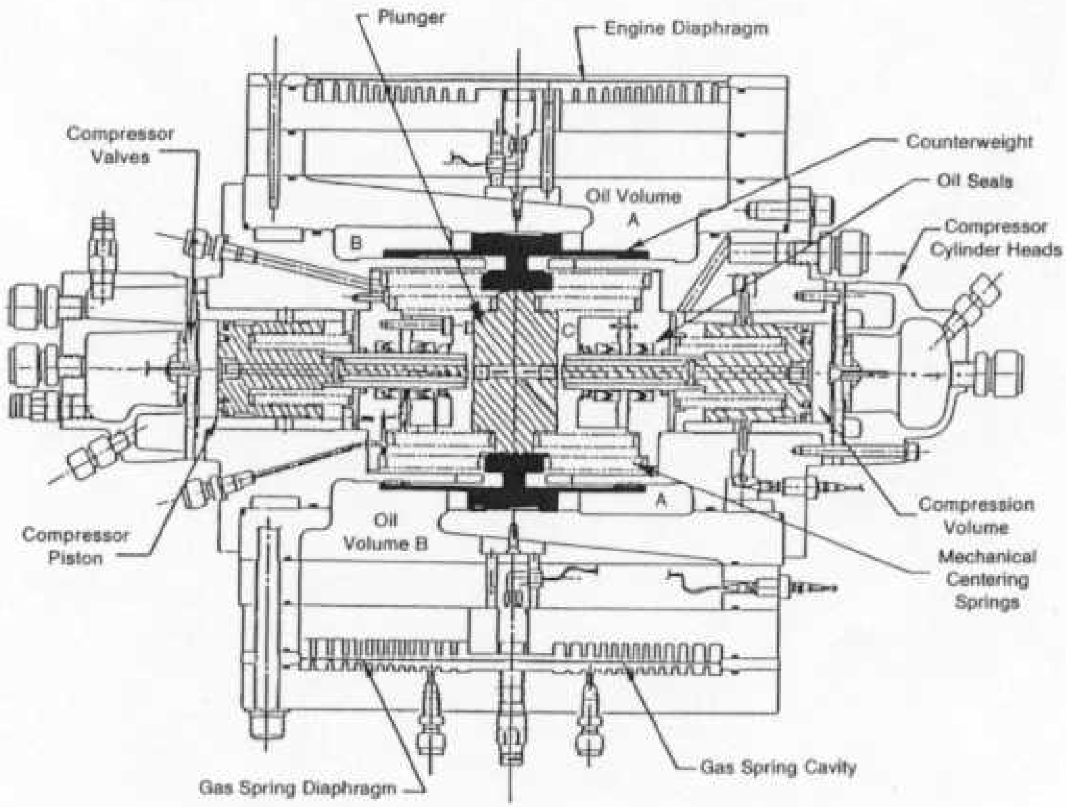


*Figure 3. Original internal counterweight hydraulic transmission and compressor*

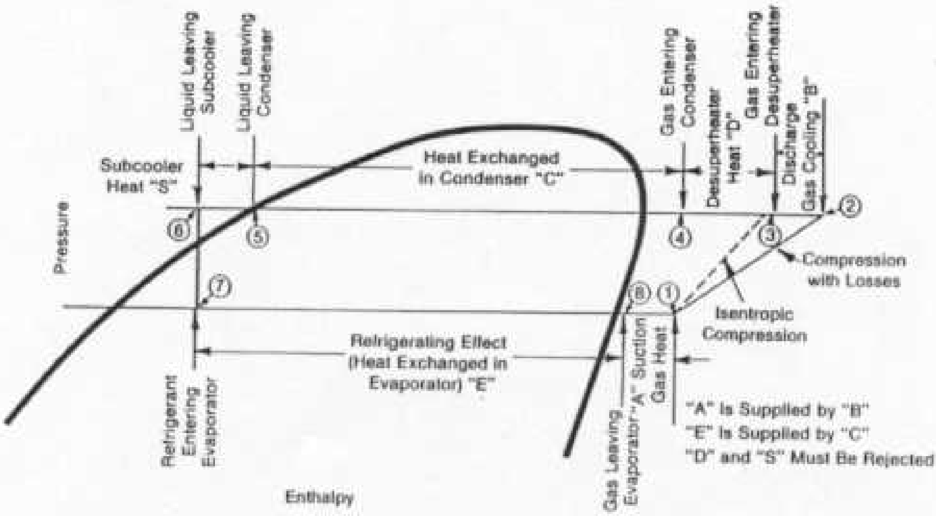


*Figure 4. Refrigerant test loop state points*

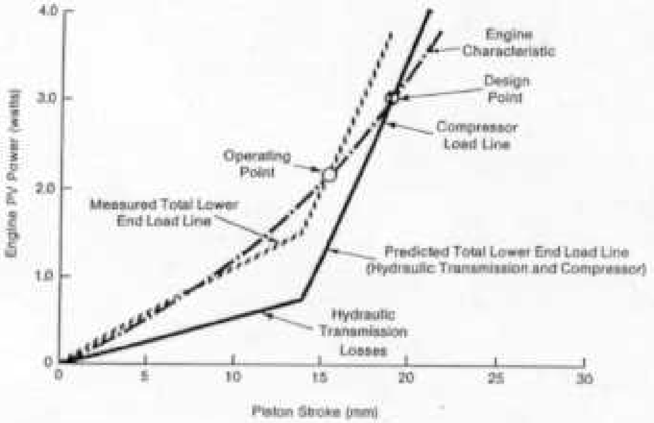


Figure 5. Representative lower end and EM FPSE power characteristics

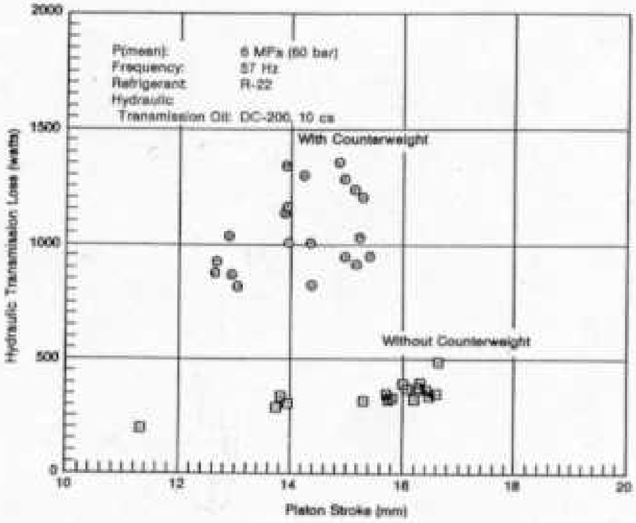


Figure 6. Hydraulic transmission loss

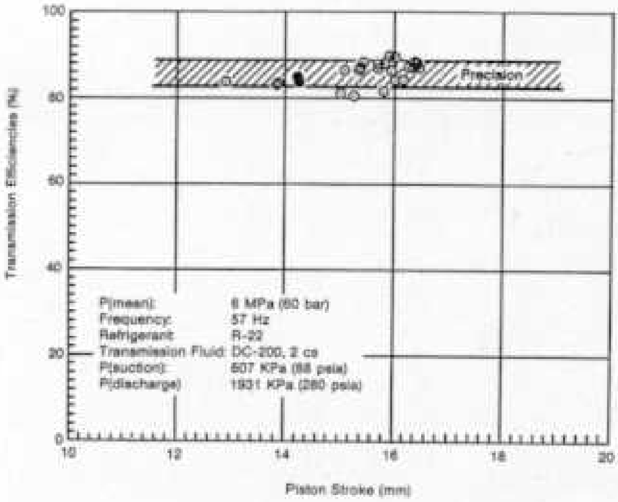


Figure 7. Hydraulic transmission efficiency for direct-drive compressor

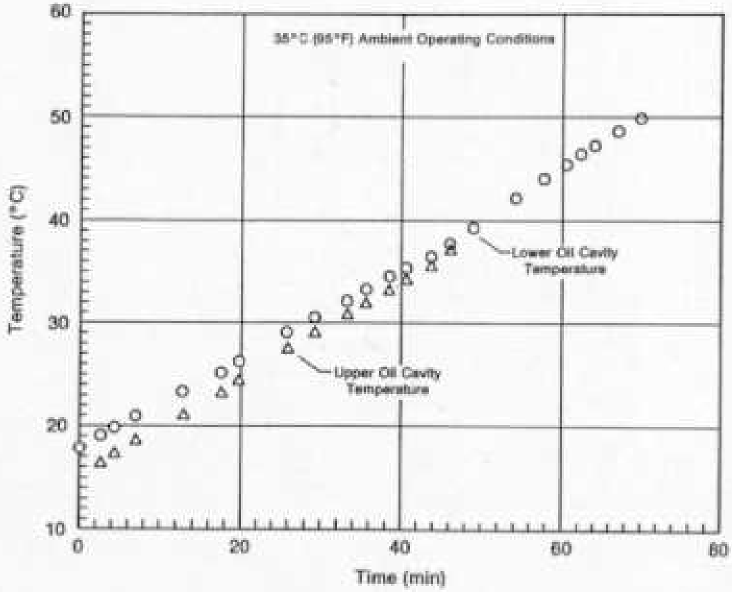


Figure 8. Oil cavity temperature rise with internal counterweight

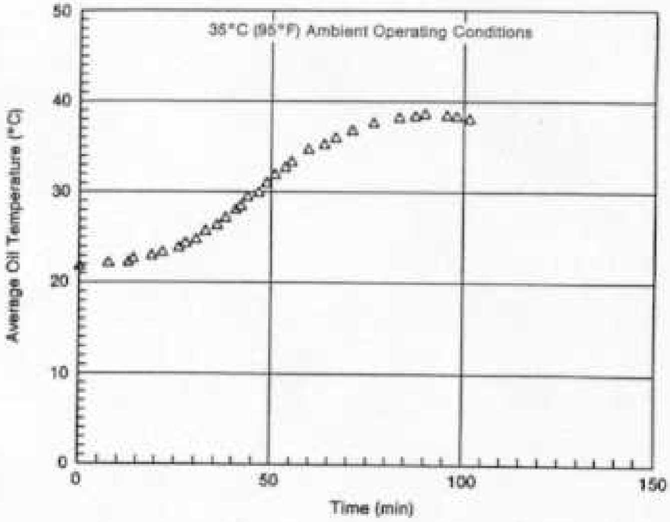


Figure 9. Oil cavity temperature rise for direct-drive compressor

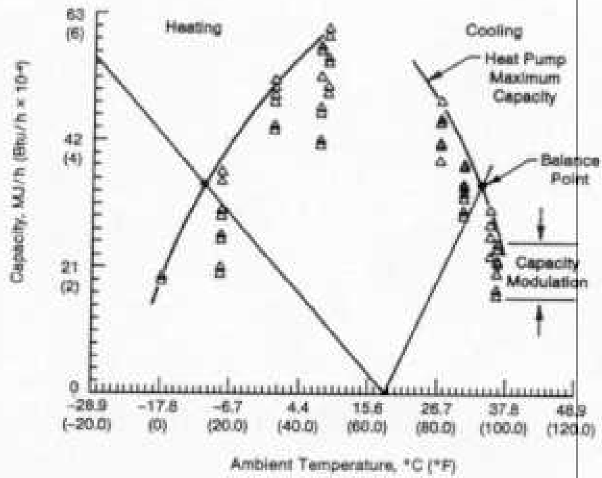


Figure 10. FPSE heat pump power module baseline map (Mid-Atlantic region)

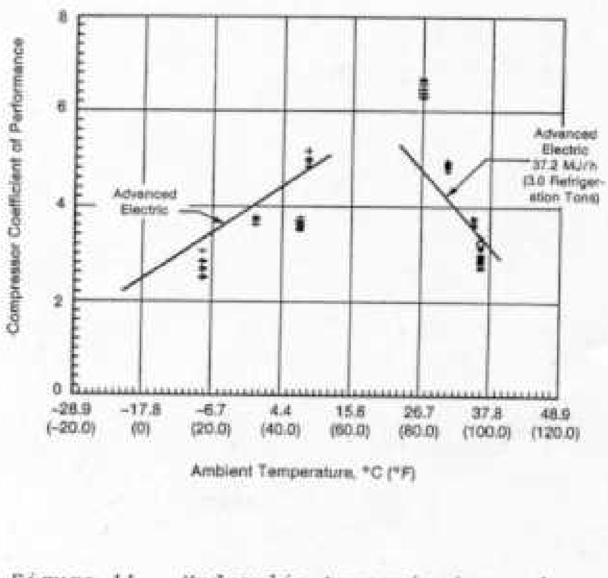


Figure 11. Hydraulic transmission and compressor performance map

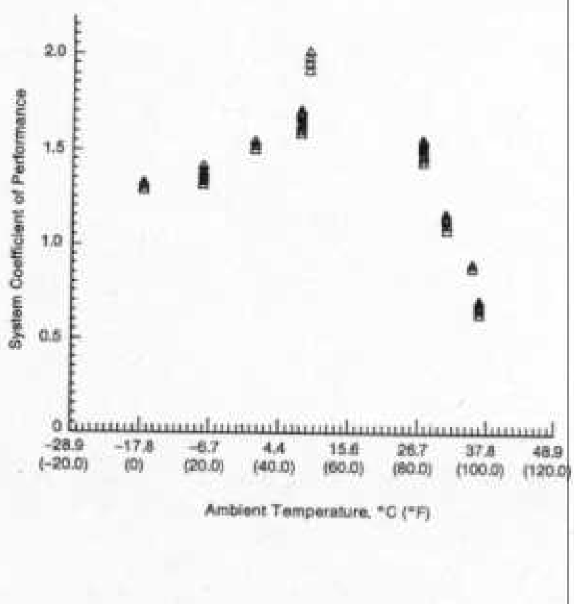


Figure 12. System coefficient of performance

# Discussion

UNKNOWN, Honeywell, Minneapolis, MN: Can machine be run automatically during operation? Is frequency timing automatic?

R.A. ACKERMANN: The heat pump is semi-automatic. The controls for the combustor, hydraulic transmission, and compressor are all automatic; however, the dynamic tuning control, or frequency control, is performed manually by adjusting the displacer frequency to optimize the refrigerant flow rate. This adjustment is small, varying less than 5% from a 95 F to a 0 F day.

W.H. SEYBRING, Columbia Gas, Norwalk, OH: How do you cool the engine and with what?

ACKERMANN: Water is used as the engine coolant. For testing in our laboratory, the coolant temperature is controlled with a commercial chiller. Auxiliary power for the heat pump consists of:

| | Ambient Temperature 95°F | 0°F |
|---|---|---|
| Displacer Motor Control | -250W* | +100W* |
| Combustor Blower | -100W | -100W |
| Coolant Pump | -150W | -150W |
| System Control | - 50W | - 50W |
| | -550W | -200W |

* Negative number signifies input; positive signifies output

S.V. SHELTON, Georgia Institute of Technology, Atlanta: How does the compressor piston stroke vary with decreasing evaporator pressure and constant condenser pressure?

ACKERMANN: Decreasing the evaporator pressure has two effects on the piston stroke. First, lowering the evaporator pressure raises the pressure ratio of the refrigerant. This loads up the compressor and reduces the piston stroke. Second, lowering the evaporator pressure changes the dynamics of the compressor, which leads to a change in dynamic tuning and a lowering of the piston stroke. Both of these effects may be controlled through the displacer stroke, and the piston stroke may be maintained constant within the power envelope of the engine.

K.J. KOUNTZ, Inst. of Gas Technology, Chicago: What would the effect of 100 F or greater engine coolant water temperature (versus the 50 F or less used in the experiments to date) on engine efficiency?

ACKERMANN: We have estimated that a $20^{o}C$ rise in coolant temperature will reduce engine efficiency by 1.4 points.