PERFORMANCE OF A STIRLING ENGINE POWERED HEAT ACTIVATED HEAT PUMP

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ABSTRACT

A heat activated heat pump (HAHP) which employs a natural gas fired Stirling engine to drive a Rankine cycle vapor compressor is presently being developed by the American Gas Association, the Gas Research Institute, The Department of Energy and the General Electric Company. The concept of a HAHP has the potential for reducing the amount of gas required for space heating since it directly utilizes the engine waste heat in addition to the energy obtained by the heat pump effect. The Stirling engine/Rankine cycle refrigeration loop heat pump being developed would conserve about one-half the gas required by conventional space heating equipment. As a result, this gas could be made available for new residential customers or for alternate applications. Additionally, the Stirling/Rankine concept offers an efficient and competitive means of providing summer cooling.

This paper updates the status of the HAHP program as it was presented at the 12th IECEC. The present paper concentrates on a description of the hardware comprising the HAHP and presents the results of subsystem level testing, including the combustor, Stirling engine, compressor and air handler packages. The impact of measured component performances on the system level performance predictions is discussed.

INTRODUCTION

The joint development of a prototype gas heat pump began in January 1975. When commercialized, such a product will not only conserve our nation's energy resources, but it will also reduce the cost to the consumer for the space conditioning function. The operating savings become more significant as energy costs continue to escalate.

The heat activated heat pump is envisioned as a unitary heating and cooling product competing with the same market as the gas year-round and the gas furnace/electric air conditioning systems. As with its competitors, the HAHP could serve the residential and commercial markets including both new and replacement segments of each.

The Preliminary Design was completed in 1975. During this phase, several technical approaches to the implementation of a HAHP were evaluated. In addition, market cost and business evaluations were carried out to aid in evaluating the feasibility and viability of the concept as a product. As a result of this phase, a Stirling/Rankine, prime mover/refrigeration cycle approach was selected as the most attractive of several approaches.

Currently, the program is in the Prototype Development Phase. The objectives of this phase are to demonstrate that the projected performance can be attained. To accomplish these goals, a prototype of the gas heat pump has been designed and fabricated and is presently being tested. This first prototype will serve to demonstrate component and system interactions. Following evaluation of test data, a second prototype will be designed and fabricated. The performance characteristics of the prototype and the results of concurrent marketing and business evaluations will be used to assess the attractiveness of the present concept.

SYSTEM DESCRIPTION

The heat activated heat pump (HAHP) employs a prime mover in place of an electric motor to drive the Rankine cycle refrigeration compressor. The primary advantage of the HAHP is that the engine waste heat can be employed to augment the heat pump effect in the heating mode of operation. Consequently, the HAHP has both a higher capacity output and a higher coefficient of performance (based upon primary fuel utilization) than do conventional heating systems. Performance estimates (1,2) have shown that Stirling driven heat pumps can achieve Seasonal Performance Factors (SPF's) up to twice that of conventional systems in the heating mode and 50% better when both heating and cooling are considered.

The objective of the prototype HAHP is to demonstrate that these performance levels can be obtained. This Stirling driven heat pump which has been designed to provide three tons of cooling at the 95°F ARI rating temperature will provide approximately 62,000 Btu/hr at the 47°F ARI ambient rating temperature in the heating mode of operation; the COP's at these operating temperatures are 1.06 and 1.75, respectively, excluding the electrical power requirements to operate the fans, blowers and pump.

There are several possible modes for operating the HAHP; a constant firing rate option was discussed in Reference 1. An operating mode alternative which is being considered for the HAHP is a dual stepped power level; the engine's power level will be stepped

at the extremes of the ambient operating temperature range as is shown in Figure 1. The stepped mode of operation has an advantage in that the system capacity characteristics would better match that of the building thermal load than do constant speed and constant load machines. Consequently, cycling losses will be reduced and overall system performance will be improved. This stepped mode of operation resulted in a HAHP with SPF's higher than those obtained with a fully modulated operating mode. This is attributable to the reduced engine performance for the fully modulated system at the moderate ambient operating temperatures where the engine is operated at an off design operating point and the power output and performance is low. Performance predictions - capacity and COP - for the stepped and the constant firing rate operating modes are presented in Figure 2 as a function of ambient temperature.

A schematic of the HAHP and a description of the components was presented in Reference 1. Briefly, the HAHP employs a free piston Stirling engine (FPSE) to directly drive a free piston linear compressor. Energy is supplied to the system through the combustion of natural gas. The thermal energy which has been liberated is transferred to the FPSE working fluid via a nest of finned tubes which surround a radiation cooled transpiration burner. The engine converts approximately 1/3 of this thermal energy to mechanical power. The remaining energy is engine "waste" thermal energy which can be used to augment the heat pump effect in the heating mode of operation. A hydronic loop transfers this energy to the indoor air handler during the heating mode of operation and to the outdoor heat exchanger package during the cooling operation mode. The refrigeration loop is comprised of conventional HVAC components, and Refrigerant-22 is the working fluid.

The requirements and goals for the HAHP subsystems described above are summarized in Table 1. A description of these subsystems and the results of subsystem testing are presented in the following sections.

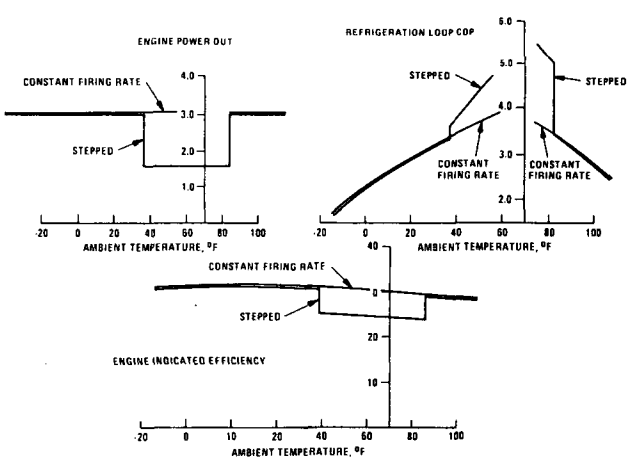

Fig. 1 - Stepped operating mode: definition and component performance

Table 1 - Subsystem Requirements and Goals*

Combustor:	Overall Efficiency	80%
	Air Preheat	800°F
	% Excess Air	15%
Engine:	Power Output	3 KW
	Cycle Efficiency	32%
	Hot Side Temperature	1200°F
	Coolant Inlet Temperature	140°F
Refrigeration Loop:	COP	3.5
	Condensening Temperature	120°F
	Evaporating Temperature	45°F
	Compressor Isentropic Efficiency	75%

*Based upon a 95°F ambient rating temperature

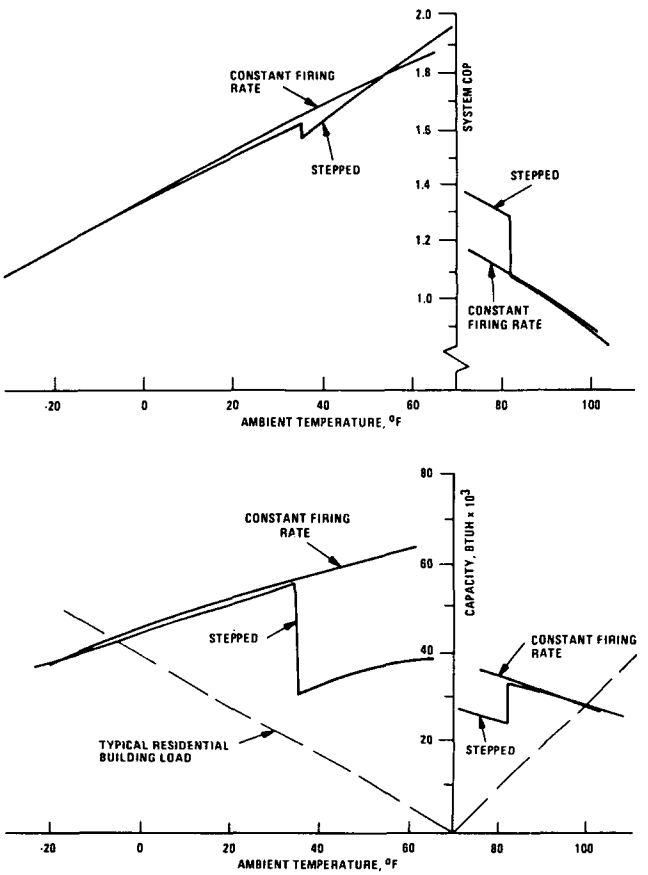

Fig. 2 - HAHP system performance: stepped mode operation

SUBSYSTEM DESCRIPTION

COMBUSTOR - A schematic of the cylindrically shaped sheet metal combustor assembly incorporating a radiation-cooled porous frit burner is illustrated in Figure 3. The actual combustor components are shown in Figures 4 and 5. Combustion air passes through an integral exhaust gas/inlet air heat exchanger and is preheated prior to being mixed with the gaseous fuel. The fuel/air mixture enters the burner through an inner-core

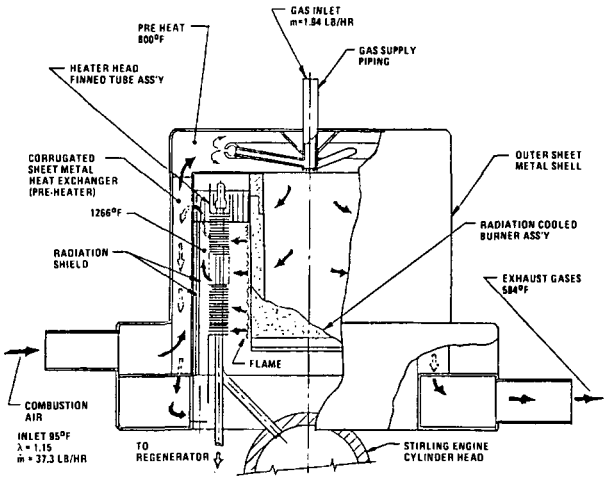

Fig. 3 - Combustor assembly schematic

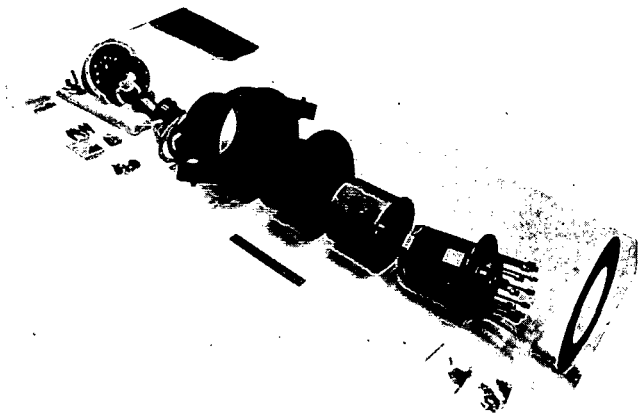

Fig. 4 - Combustor assembly hardware

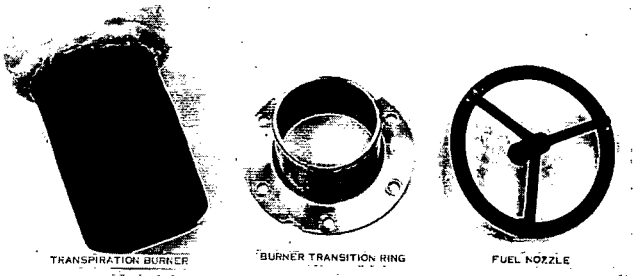

Fig. 5. - Radiation cooled transpiration burner components

and is burned at the outside of the porous frit cylinder. The hot gases pass through the finned tube heat exchanger transferring thermal energy to the Stirling engine's helium working fluid and are subsequently ducted into the exhaust gas side of the inlet air preheater. All combustion products are collected in a manifold and exit the combustor via an exhaust duct.

The compact combustor assembly results from the successful development of a high energy density burner. The energy released at the flame holder surface area will exceed 1600 Btu/hr/in^2. The nominal temperature of the air/fuel mixture entering the flame holder will be 800°F. Tests to date indicate preheat temperatures exceeding 1000°F are possible without pre-ignition problems. The combustor design requirements and measured test results are summarized in Table 2.

FREE PISTON STIRLING ENGINE - The free piston Stirling engine is a single cylinder machine with the displacer and power piston contained within the cylinder in an in-line configuration. The engine housing assembly is the pressure vessel member and contains the heater head, regenerators and cooler subassemblies. Figure 6 illustrates the engine housing assembly.

The heater head is a finned tube configuration interconnecting the cylinder head and the regenerator assemblies. There are twelve heater head tubes with each one connected to a single regenerator. The regenerators contain packed wire mesh material and perform a recuperative function on the helium working fluid. Each regenerator is connected to an individual cooler tube which interfaces with the cylinder near the midplane. All cooler tubes are housed within a sheet metal envelope which comprise the engine's low temperature heat exchanger subassembly.

The engine power piston is an aluminum cylinder

Table 2 - Combustor Design Point Definition and Operating Results

	Predicted Design Point Definition	Test Results
Burner Energy Release - (Btu/hr)	42,000	42,000
Effective Combustor Efficiency - (%)	80	85
Air Preheat - (°F)	800	815
Excess Air - (%)	15	15
CO Emission Level - (%)	0.04	0.04
Air Side Pressure Differential - (in of H_2O)	1.0	1.0

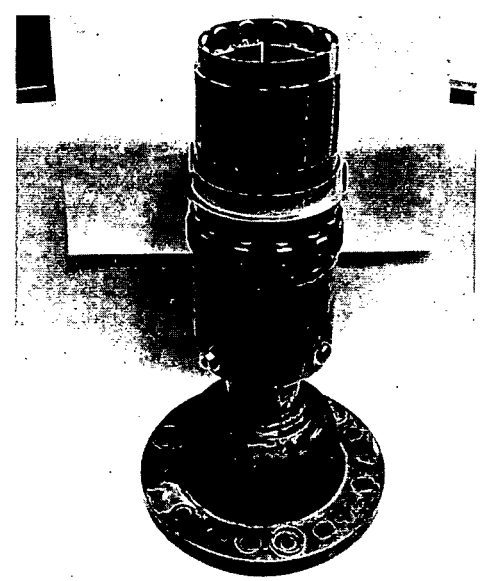

Fig. 6 - Engine housing assembly

fabrication. The inside diameter of the piston is fitted with a bearing sleeve which mates with the displacer assembly. The outside diameter is machined to accept a carbon graphite material bearing sleeve. The piston, bearing sleeve and liner components are illustrated in Figure 7.

The displacer, along with its seal ring and rod, are shown in Figure 8. The displacer is a hollow assembly that is designed to be guided within the engine housing cylinder via the displacer rod/power piston internal bearing configuration. The seal ring is fabricated from a carbon graphite material as is the piston bearing sleeve.

An adapter couples the power piston to the driven hardware, the inertia refrigeration compressor, which is depicted in Figure 9; this adapter also houses the linear variable displacement transducer which measures the motion of the displacer relative to the power piston. Mounting of the adapter to the power piston completes the displacer gas spring cavity within the power piston.

The design parameters for the free piston Stirling engine are shown in Table 3.

The component testing of the free piston Stirling engine was performed in parallel with the compressor evaluation. The prime combustor hardware provided the input energy, and the output power was absorbed by a gas compressor. This loading device exhibited critical damping characteristics which limited the output capacity to 78% of predicted. The engine performance predictions and test conditions are presented in Table 4.

Actual piston motion was measured relative to ground by using a linear variable displacement transducer (LVDT) attached to the compressor. Pressure transducers were installed in the engine cold working space and the bounce space to measure the dynamic pressure levels. Thus it is possible to dynamically record the data and determine frequency, strokes, piston/displacer phase angle, piston work and engine working pressure levels.

Table 3 - Free Piston Stirling Engine Design Parameters

Working Fluid	Helium
Charge Pressure - (psia)	870/1160
Power Level - (kW)	3.0
Design Efficiency - (%)	32
Heater Head Temperature Level - (^{0}F)	1200
Coolant Inlet Temperature Level - (^{0}F)	140
Power Piston Storke - (in)	1.7
Frequency - (Hz)	26

Table 4 - FPSE Performance Predictions and Test Conditions

Parameters	Predicted Values	Test Conditions
Charge Pressure	60-80	40-80
Firing Rate (KBTUH)	40	20-40
Frequency (Hz)	27	18-26
Power Piston Stroke (in.)	1.7 max.	0.9-1.4
Displacer/Piston Phase Angle (degrees)	30 nominal	55-75
Output Power (Watts)	1800	1400

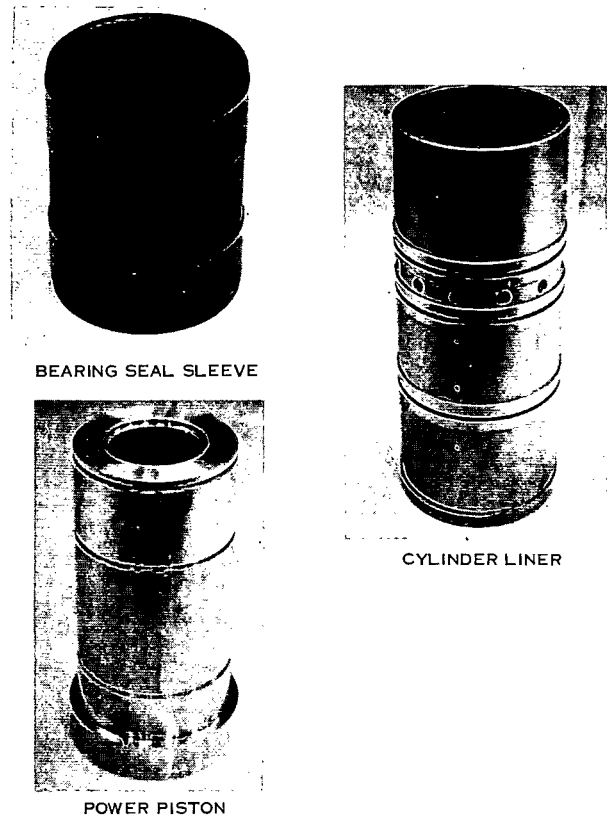

Fig. 7 - Engine piston components

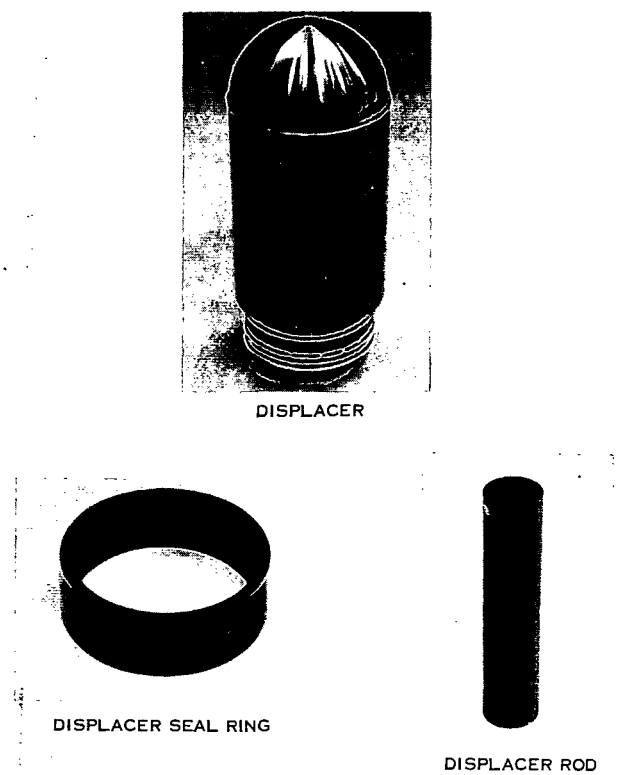

Fig. 8 - Engine displacer components

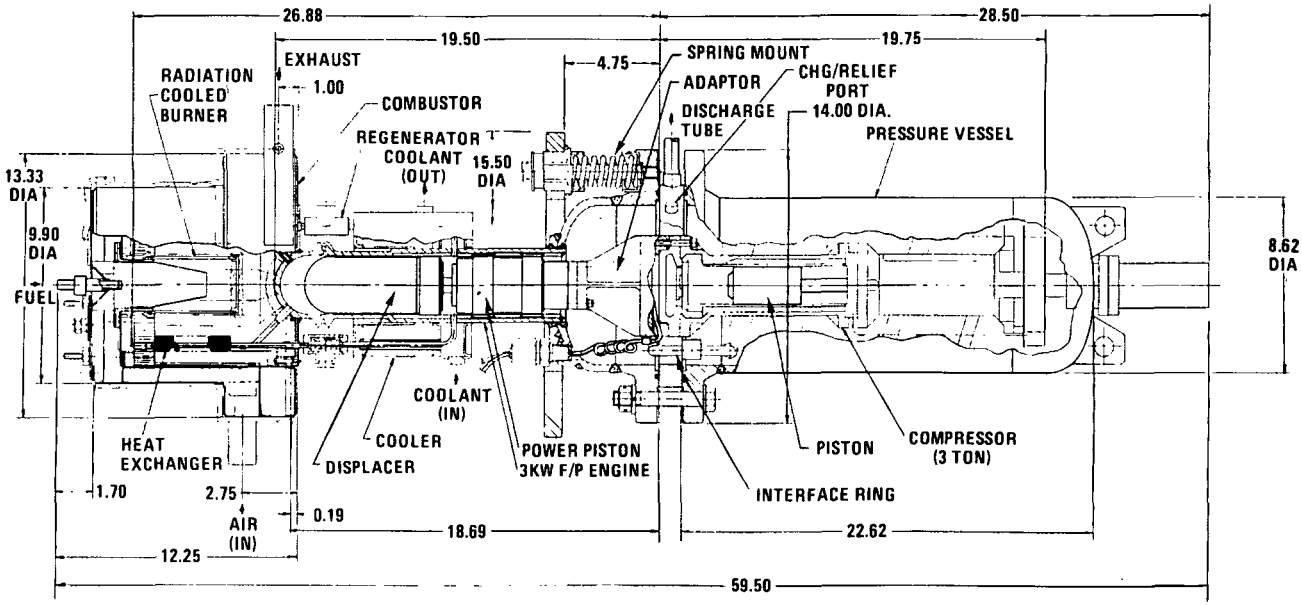

Fig. 9 - Engine/compressor assembly

All component test objectives were achieved. Proof of the design concept was clearly demonstrated, and the understanding of the engine/load interaction was established. The gas springs were optimized for the given configuration, and the engine performance was evaluated, thus verifying the empirical and analytical correlations.

INERTIA COMPRESSOR - The inertia compressor assembly is a two-cylinder, two-piston single acting unit shown in Figure 9. The cylinders are an in-line configuration and are mirror images of each other about the midplane. Both the upper and lower cylinder assemblies are shown in Figure 10. The cylinder assemblies are fabricated from aluminum and contain a cast iron sleeve. Each cylinder is instrumented with a pressure transducer.

The valve assembly, piston and cylinder head are shown in Figure 11. Each valve nests into the cylinder liner and separates the suction side refrigerant from the discharge refrigerant. The pistons are joined together via a connecting rod and close fitting wrist pins. The volume contained between the valves and the outer faces of the pistons are the compressor working volumes. The volumes between the midhousing plate and the connecting rod side of the pistons comprise the gas spring volumes.

The inertia compressor was assembled and tested separately from the free piston Stirling engine. The compressor was mounted to a motor driven crank arm mechanism and was driven at various frequencies and preset amplitudes which correspond to the predicted engine stroke levels. The refrigerant load was provided by a unitary packaged commercial heat pump. The compressor

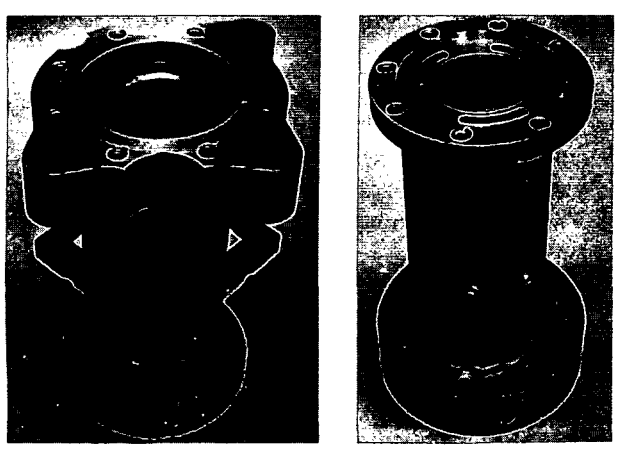

Fig. 10 - Inertia compressor housings

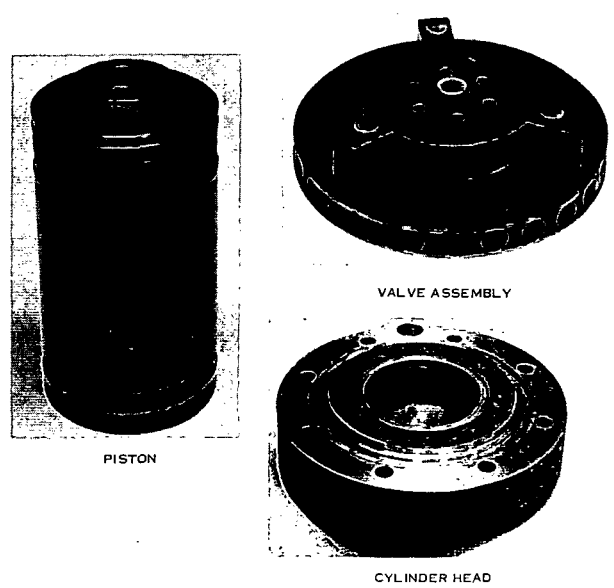

Fig. 11 - Inertia compressor components

was fully instrumented, including pressure transducers in both cylinders and in the gas spring volumes. A typical predicted versus measured energy balance of the compressor is presented in Figure 12.

CONTROLS - The functions of the control system are to provide for proper sequencing of events so that normal heat pump operation can be satisfied - the primary control function - and to detect abnormal operation and implement the shutdown procedure - the safety control function. The primary controls include the indoor thermostat, the on-off sequencing during start-up and shutdown of the HAHP, and the defrost control for the outdoor refrigerant coil. The safety controls include flame safety, engine overstroke protection, detection of low fluid flow in the hydronic loop, and detection of high/low refrigerant pressure. These developmental controls utilize both electronic and electro-mechanical logic to provide safe, semiautomatic operation of the total system.

AIR HANDLERS - The design of the air handler packages are similar to those of a conventional split system heat pump. Both air handlers utilize a compartmentized approach which permit test flexibility, simplicity and excellent accessibility for component adjustment or replacement. The indoor unit is standard hardware except for the addition of a transition cap which houses the hydronic coil. This hydronic coil provides space heating capability while dissipating the engine's waste heat during the heating mode of operation. The indoor unit is illustrated in Figure 13.

The outdoor unit is schematically shown in Figure 14, and the actual hardware is presented in Figure 15. The design utilizes a three compartment approach. They are:

1) The air side heat exchanger compartment

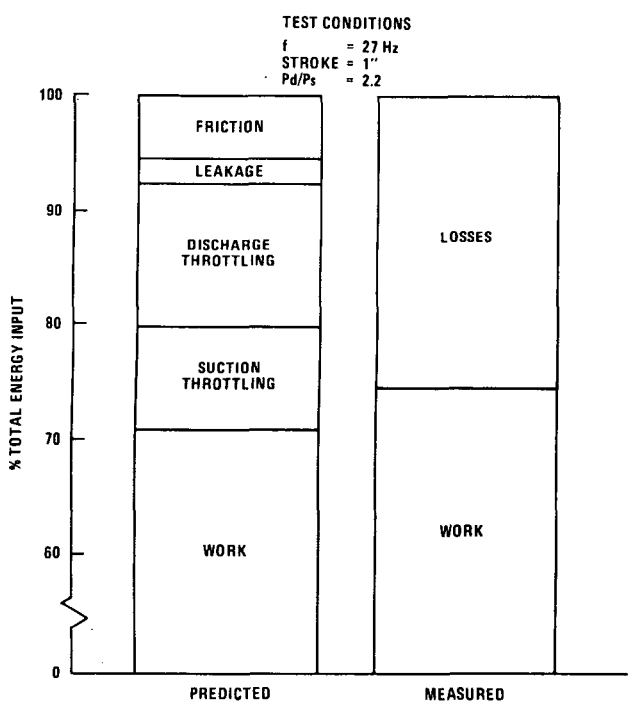

Fig. 12 - Prototype 1 inertia compressor energy balance

2) The refrigerant, hydronic and control subsystems compartment

3) The combustor/engine/compressor assembly compartment

Compartment No. 1 employs low pressure drop, single bank heat exchangers with an upward discharge, top mounted fan system. The hydronic heat exchangers are mounted in the upper portion of the compartment. Two parallel flow sections comprise the refrigerant

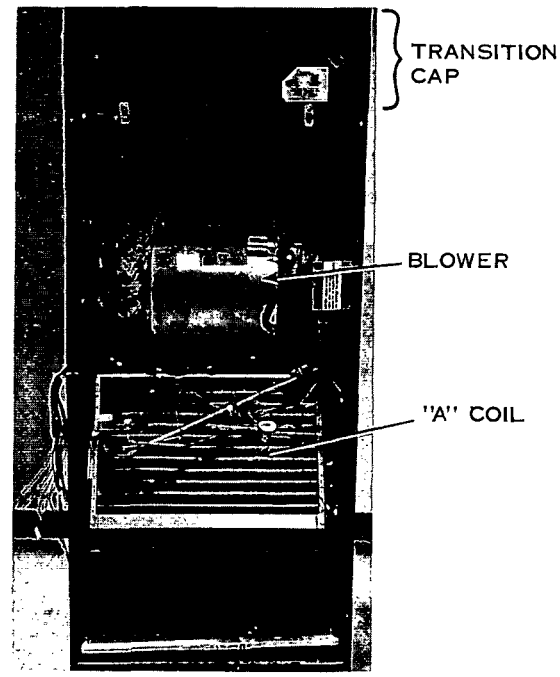

Fig. 13 - Modified air handler assembly

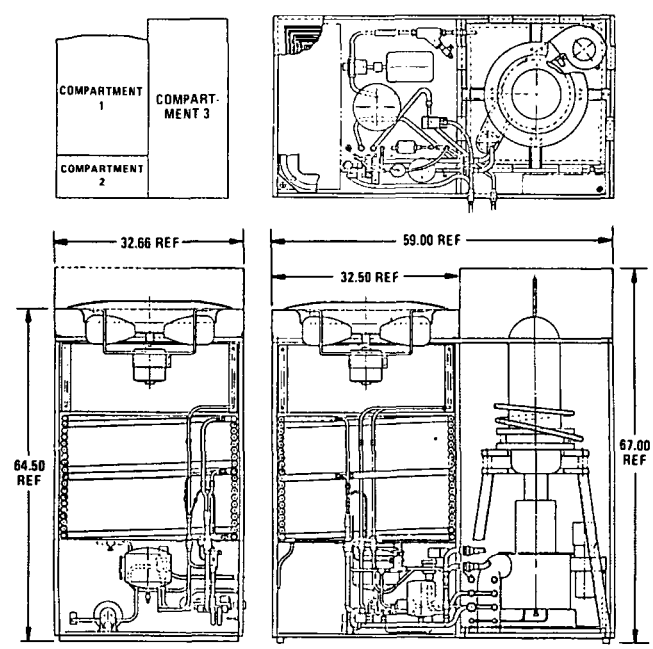

Fig. 14 - Prototype 1 outdoor unit hardware arrangement

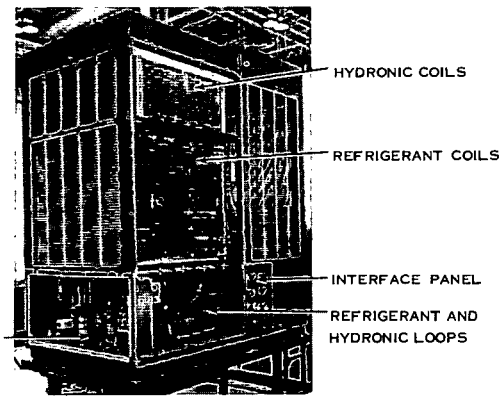

Fig. 15 - Outdoor air handler - Prototype 1 assembly

heat exchanger.

One side of Compartment No. 2 contains the refrigerant components. These include the 4-way switchover valve, thermal expansion valve, filter/dryer, distribution system and the interface connections into Compartment No. 1. The hydronic loop pump, sump, 3-way switchover valve, etc., are positioned adjacent to the refrigerant loop. All controls other than the gas controls are housed in a separate watertight box which opens to the outside through a separate panel.

Compartment No. 3 houses, in addition to the combustor/engine/compressor, the combustion air blower, the gas controls, exhaust duct and the indoor unit interface panel. During component checkout and verification testing of the air handlers, a conventional electric driven compressor was employed to simulate the Stirling engine/inertia compressor. A summary of the predicted versus measured capacities of the various heat exchangers as a function of ambient temperatures is presented in Table 5.

ENGINE/COMPRESSOR ASSEMBLY

The assembly of the combustor and the free piston Stirling engine (FPSE) coupled to the inertia compressor is depicted in Figure 9. Figure 16 illustrates the assembled hardware mounted in the facility test stand. The major axis of the assembly is vertical with the single cylinder Stirling engine nested atop the combustor assembly and below the compressor. The inertia compressor is hard coupled to the engine power piston and totally contained within the engine bounce space pressure vessel. A three point spring suspension system mounted to the facility stand supports the engine/compressor/pressure vessel assembly, thus providing excellent isolation of the reciprocating restoration forces from the heat pump frame.

Checkout testing of the integrated FPSE/Inertia Compressor assembly has been completed. Performance mapping is currently underway and scheduled for completion by mid 1978.

CONCLUSIONS

The subsystem level testing which has been performed to date has shown that the product performance goals which were established as a guide for the subsystem design are

Table 5 - Comparison of Test Data and Systems Program Predictions

Case	Condenser Measured Btuh	Capacity Predicted Btuh	Evaporator Measured Btuh	Capacity Predicted Btuh	Hydronic Measured Btuh	Capacity Predicted Btuh
95°F Day	48,900	48,600	36,400	34,300	22,700	22,500
105°F Day	48,400	48,200	34,400	32,800	23,100	23,100
45°F Day	36,500	38,400	27,800	26,900	23,500	22,200
60°F Day	45,000	45,100	35,000	33,900	22,300	22,200

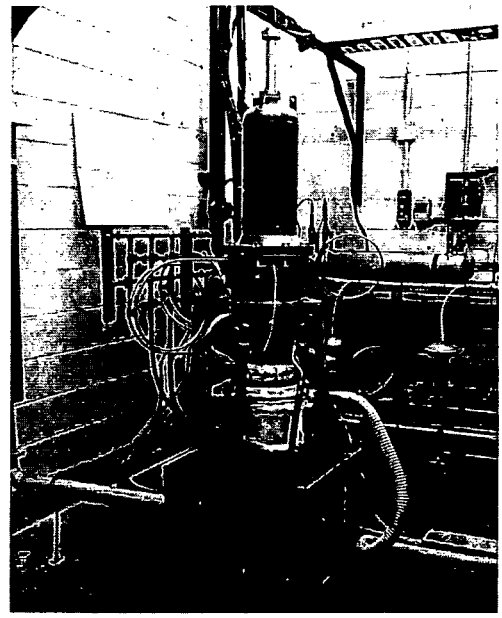

Fig. 16 - Assembled hardware mounted in the facility test stand

achievable. The measured performance levels for the first prototype subsystems, though slightly below the product goals, agree well with analytical predictions. The results of the Prototype 1 tests, the product goals, and the performance level predictions for the Prototype 2 subsystems are presented in Table 6. Based on the Prototype 1 observations it now appears that product goals can be surpassed in Prototype 2 hardware as shown in Table 6. The impact of these different subsystem performance levels on system performance are presented in Table 7. The system performance is presented in terms of the Overall (Heating and Cooling) Seasonal Performance Factor (SPF) for a Philadelphia location. This SPF includes the electricity to power the fans, pumps and blowers and is based upon primary fuel utilization (i.e., electricity is reflected back to the power plant).

Significant savings in energy can be obtained through the use of heat activated heat pumps as is evidenced from the Seasonal Performance Factors presented in Table 7. Considering just the first development unit, the SPF, prediction is 50% higher than present day, conventional HVAC systems. The Stirling/Rankine HAHP, therefore has the potential for becoming an attractive HVAC product and provides an opportunity to conserve the nation's energy.

A system test employing the Prototype 1 components

Table 6 - Subsystem Performances Measured and Predicted

Subsystem	Product Goals	Prototype 1 Test Results	Protype 2 Prediction
Combustor	80%	84%	85%
Engine	32%	30%	35%
Refrigeration Loop COP	3.5	3.2	3.5

Table 7 - Sensitivity of Systems Performance to Variations in Subsystem Level Performance

Basis for Calculation	Overall SPF
Product Goals	1.18
Prototype 1 Test Results	1.04
Prototype 2 Predictions	1.30

will be started by the third quarter of 1978. Development of a second cycle of prototype hardware will also be started in 1978. This hardware will be employed in an HAHP demonstration program to obtain performance information of prototype configured hardware under various climatic conditions of the U. S. and to provide visibility of the Stirling/ Rankine HAHP in the HVAC industry and in the consumer market.

REFERENCES

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2. "Heat Pump Technology: A Survey of Technical Developments, Market Prospects and Research Needs," Report No. CONS/ 2127-d, July 1977.