SYSTEM PERFORMANCE OF A STIRLING ENGINE POWERED HEAT ACTIVATED HEAT PUMP

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ABSTRACT

The development of the first prototype heat activated heat pump (HAHP) jointly sponsored by the Gas Research Institute, the Department of Energy and the General Electric Company was completed. It is envisioned as a unitary heating and cooling product competing for the same market as is currently served by the gas year-round and gas furnace/electric air conditioning systems. Utilization of the system waste energy to supplement the heating capacity in addition to the heat pump effect would enable a HAHP system to significantly out-perform conventional heating/cooling devices.

This paper will present the results obtained in testing the first prototype of the complete HAHP system. The matching characteristics of the subsystem components, especially between the free piston Stirling engine and the free piston linear compressor, will be described. Optimization of the system matching techniques will be presented. The discussion will include potential improvements in the system performance based on the first prototype test results.

INTRODUCTION

A heat activated heat pump (HAHP) utilizing natural gas as the energy source is presently being developed by the Gas Research Institute, the Department of Energy, and the General Electric Company. It will be a unitary heating and cooling product for residential and commercial space conditioning applications.

Utilization of the waste energy of the system to supplement the heating obtained via the heat pump effect would enable an HAHP to significantly out-perform conventional gas furnace (heating)/electric air conditioner (cooling), or electric heat pump devices. Significant energy conservation for the nation and operating cost saving for consumers can be realized.

The concept evaluation and preliminary designs were completed in 1975. During this phase, several technical approaches to the implementation of an HAHP were evaluated. In addition, market cost and business evaluations were conducted to measure the feasibility and viability of the concept as a product. As a result of this effort, a Stirling/Rankine as the prime mover/refrigeration cycle, respectively, was selected as the most attractive approach (Reference 1),* and a decision was made to proceed with the Prototype Development. Superiority of a Stirling engine over other prime movers for HAHP systems was also independently confirmed by Maxwell and Didion (Reference 2).

The objective of the Prototype Development is to demonstrate that the projected performance can be realized. Two generations of prototype hardware configured for the residential market will be developed. Prototype 1 was the first attempt to test the Stirling/Rankine HAHP on a system level, with heat pump configured hardware. This system is comprised of hardware which is primarily developmental in nature (References 3 and 4). The Prototype 2 system will consist of more mature hardware since the results of the Prototype 1 component and system testing and any advanced component development analysis and testing will be incorporated into its design.

*See the References at the end of this paper.

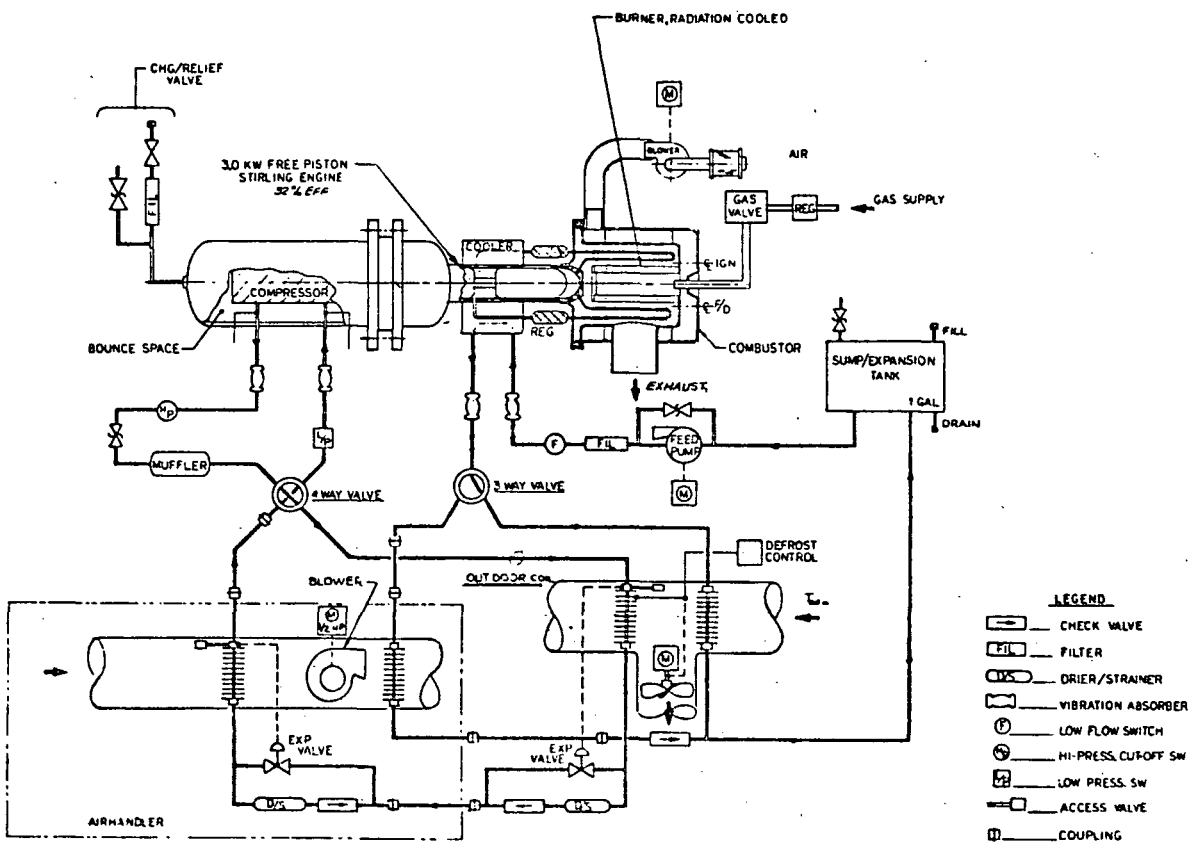


Fig. 1. - HAHP component definition

The present paper will present and discuss the test results of the first prototype HAHP system. The operational and system matching characteristics of the engine, compressor and the air handler subsystems will be discussed. System-level performance predictions will be compared to the experimental results. The discussion will include those potential improvements in the system performance which have been identified based on the analytical evaluations and empirical test results of the first prototype.

SYSTEM DESCRIPTION

The system requirements, operation and hardware designs were described in detail in previous papers (References 1 and 3). Fig. 1 schematically shows the components that comprise the HAHP system. The prime mover is a free piston Stirling engine (FPSE) with heat supplied by a gas-fired combustor. The engine provides linear motion to directly drive a double-acting, inertia type Rankine cycle vapor compressor. The combustor/engine/compressor assembly is depicted in Fig. 2.

This Stirling engine driven heat pump was designed to provide three tons of cooling at the 95°F ARI standard rating conditions and approximately 62,000 Btu/hr heating capacity at the 47°F ARI standard rating conditions. The COP's at these operating temperatures are 0.85 and 1.65, respectively, excluding the electrical power requirements to operate the fans, blower and pumps.

The thermal energy, liberated by the combustion process, is transferred to the FPSE working fluid via a nest of finned tubes which surround the radiation cooled transpiration burner. The engine converts approximately 1/3 of this thermal energy to mechanical power and drives the linear compressor. 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 operation mode and to the outdoor heat exchanger package during the cooling operation mode. The refrigeration loop is comprised of conventional HVAC components, with Refrigerant-22 as the working fluid. The requirements and goals for the HAHP subsystems described above are summarized in Table 1.

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	70%

*Based upon a 95°F ambient rating temperature

SYSTEM TEST

System level testing of the HAHP was conducted in an environmental chamber. Fig. 3 shows the assembled outdoor unit mounted inside the test chamber. The combustor/engine/compressor assembly is mounted vertically in the engine compartment. The other compartments house the controls, air/refrigeration heat exchanger, hydronic heat exchanger and refrigerant and hydronic loops. (References 1 and 3). The indoor unit is standard hardware except for the addition of a transition cap which houses the hydronic coil. This hydronic coil provides supplemental heating capability while dissipating the engine's waste heat during the heating mode of operation.

Pertinent operating parameters of the system and subsystems (including temperature, pressures, flow rates and dynamics of the three moving masses) were measured. The engine power output was measured as the total enthalpy gain of the refrigerant across the compressor. This measurement was substantiated with a force transducer measurement which was used in an earlier engine test when the FPSE was coupled with a facility load helium pump.

An energy balance was determined for each test run to ensure that all measurements were accurately interpreted. The range of test conditions for the system testing are tabulated in Table 2.

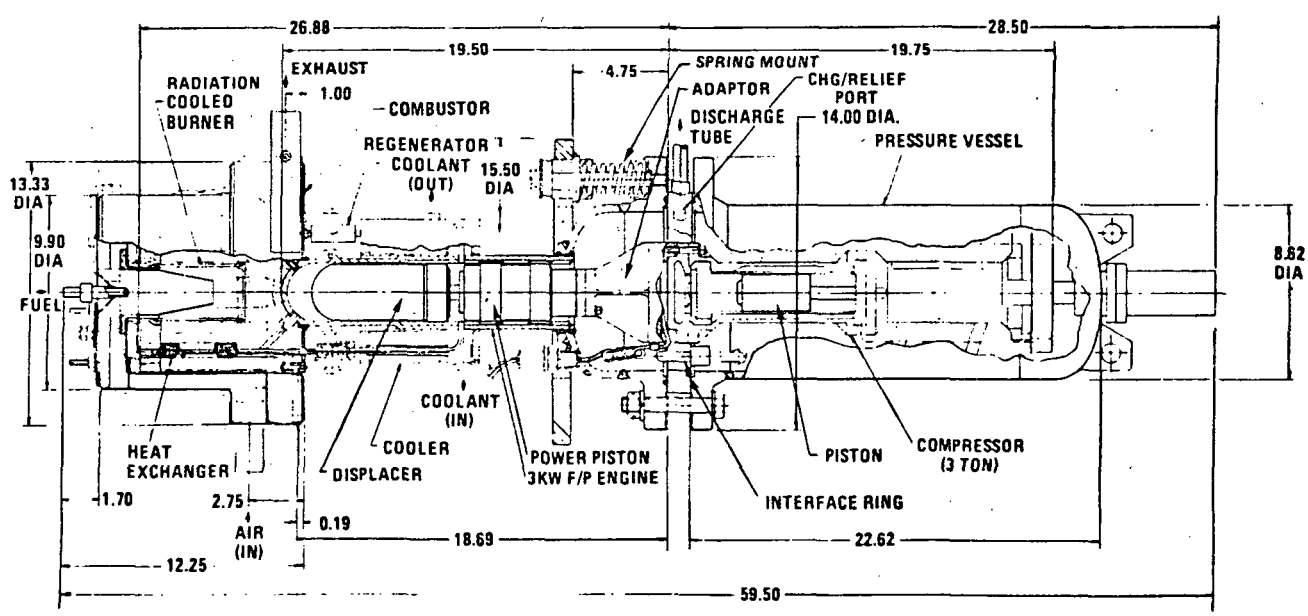


Fig. 2. - Combustor/engine/compressor assembly

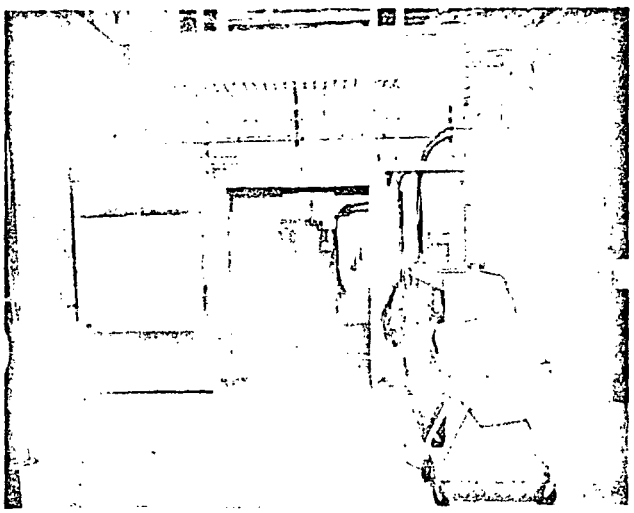


Fig. 3. - Heat pump test facility

Table 2. Test Conditions

Firing Rate (KBTUH)	20 - 65
Engine Charge Pressure (psia)	600 - 1160
Engine Operating Frequency (Hz)	18 - 26
Engine Power Piston Stroke (in.)	0.9 - 1.8
Ambient Temperature (°F)	35 - 100
Outdoor Air Flow (cfm)	3500 - 4300
Indoor Air Flow (cfm)	1400 - 2000
Hydronic Flow (gpm)	1.0 - 2.5

SUBSYSTEM PERFORMANCE

Component and subsystem testing on combustor, engine, compressor and air handlers were conducted and the results were confirmed by the system test data. Fig. 4 shows the combustor test results. The results indicate that combustor efficiencies in excess of 80% can be achieved over the operating range of the HAHP.

Fig. 5 is the performance map of the free piston linear compressor. A 66% isentropic efficiency was obtained at the design point of 95°F ambient temperature. Analysis has shown that further improvement in efficiency can be achieved by reducing the internal flow leakages, friction and throttling losses in the discharge and suction ports as indicated in Fig. 6.

The performance of the free piston Stirling engine that was obtained from both component and system level testing is shown in Fig. 7. The overall engine performance level was compromised by incorporating design flexibility into the Prototype 1 hardware to permit parametric testing. Losses associated with the hardware configuration result in thermal efficiencies measured in the 20-26% range. Table 3 illustrates the engine energy balance for a typical set of conditions both as predicted and tested. Major heat losses which contributed to the low efficiency for the first prototype engine were regenerator heat losses and working fluid leakage. With improved designs of the regenerator housing, engine piston seals, and insulation, an engine efficiency level exceeding 30% can be realistically projected. Detailed discussion of the engine performance and operating characteristics is given in a separate paper (Reference 5).

It is also noted that the free piston Stirling engine is a "constant" frequency but variable stroke machine where engine stroke is dictated by the load conditions. Engine performance is highly sensitive to changes in stroke. This sensitivity is clearly illustrated in Fig. 7. For a given heat input level, the temperature level of the heater tubes vary inversely with stroke. Engine efficiency (and therefore, the power output) vary directly with the temperature of the heater head.

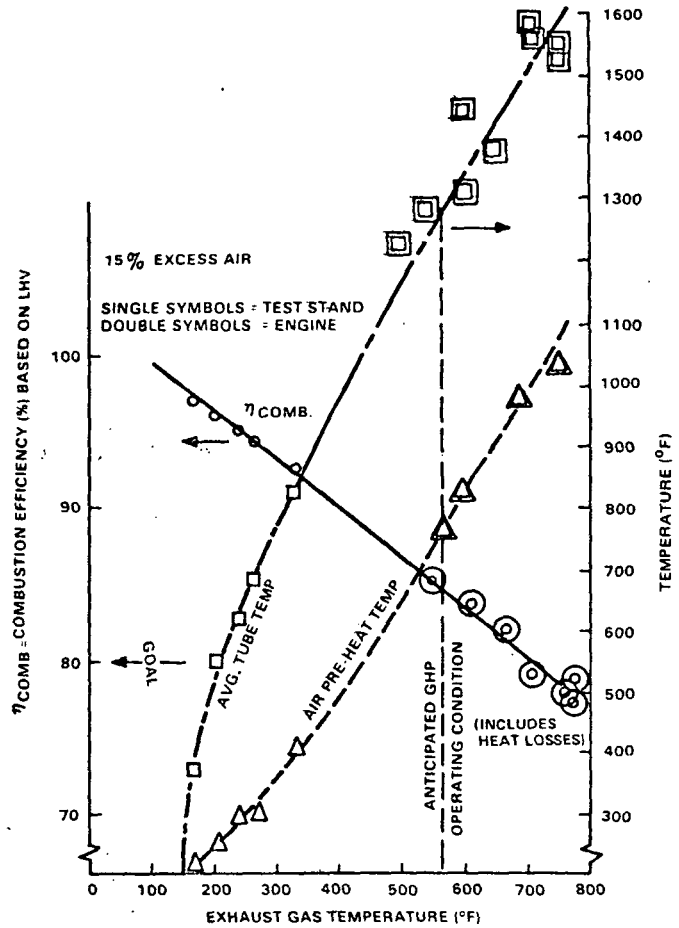


Fig. 4. - Combustor performance test results

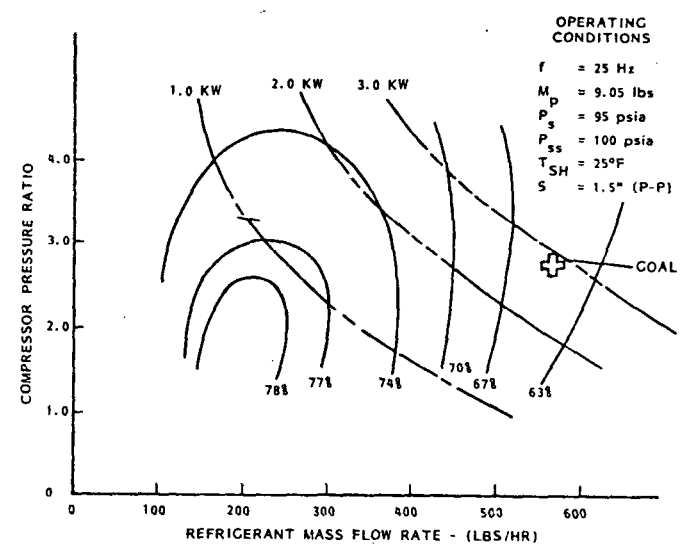


Fig. 5. - Prototype no. 1 compressor performance

SYSTEM MATCHING

System level testing demonstrated a stringent requirement on the proper matching among all subsystems in the heat pump application. To achieve a high engine efficiency, the natural frequency of the linear compressor must be properly matched with the engine operating frequency. For a given air handler/refrigerant loop design, the compressor suction and discharge pressures are primarily determined by the outdoor temperature. The resulting load variation, as a function of ambient

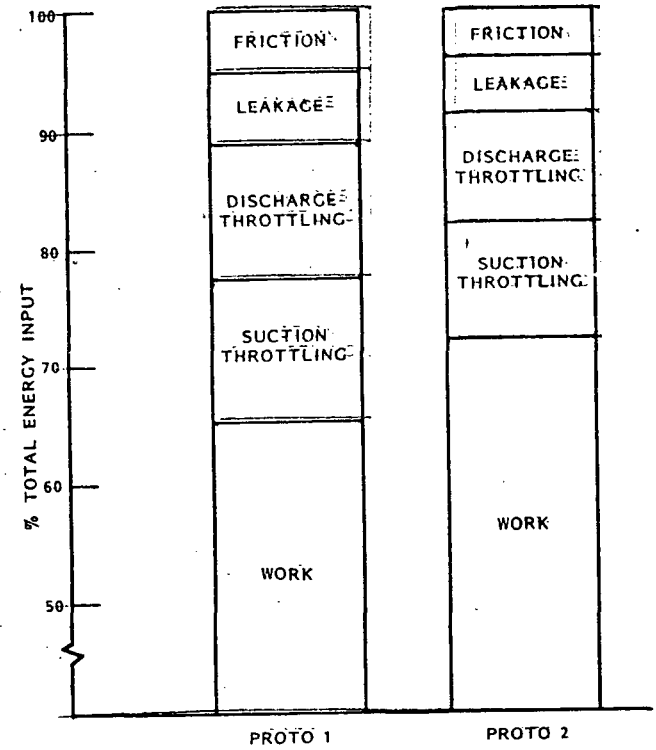


Fig. 6. - Linear compressor performance

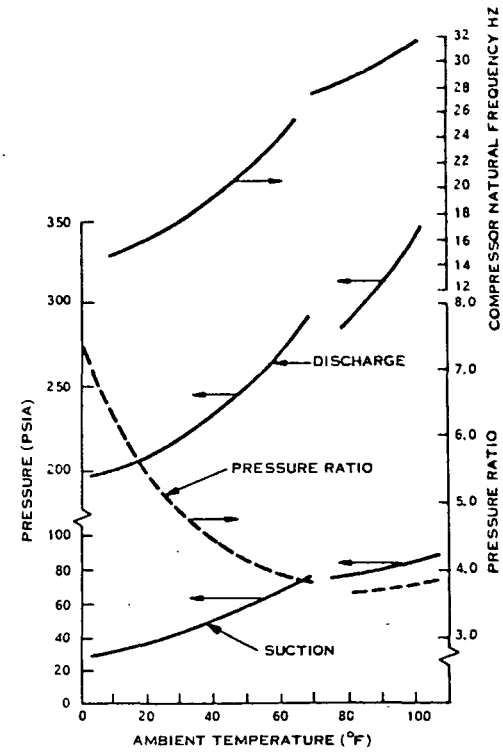
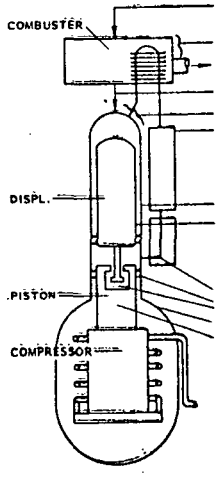


Fig. 8. - Compressor pressure characteristics

Table 3. Stirling Engine - Energy Balance



		PREDICTED UNITS IN WATTS		TEST		PROJECTED
TOTAL HEAT INPUT			12110	12922		12110
INSULATION LOSS			328	391		392
EXHAUST			1884	1935		1873
INPUT TO ENGINE			9898	10596		9845
INSULATION LOSS			55	363		243
RECENERATOR LOSSES						
A) AXIAL WALL COND	276	1513			180	517
B) AXIAL MATRIX COND	27				27	
C) RADIAL COND	910				310	
D) FLOW LEAK	300				0	
CYLINDER WALL COND			69			69
DISPLACER LOSSES						
CONDUCTION	11	203			11	203
RADIATION	41				41	
SHUTTLING	151				151	
HEAT TO COOLER			6690	6823		5662
FLOW LEAKAGE				821		293
GAS SPRING LOSS			163	163		163
HEAT CONDUCTION				280		705
ENGINE POWER			3000	2144		3287
ENGINE CYCLE EFFICIENCY			30%	20%		32%

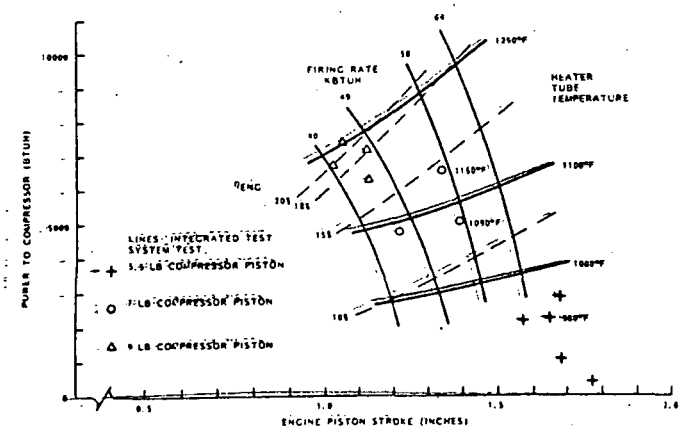


Fig. 7. - Free-piston Stirling Engine performance

temperature (pressure ratio), is shown in Fig. 8. Consequently, the most effective approach to obtain proper system matching is to modify the hardware designs of either the engine and/or compressor. In the first prototype system tests, system matching was accomplished by optimizing the compressor piston weight.

The performance maps of the compressor and the engine are superimposed in Fig. 9. The first, second and fourth quadrants show the compressor performance and operating characteristics rearranged from Fig. 5. The third quadrant exhibits the engine performance similar to Fig. 7. The system operating point is determined by matching the engine power output and the compressor power input as indicated by a closed rectangular line connecting the four quadrants of Fig. 9. For example, at the given engine operating frequency, a larger engine stroke would be required using a 5.5 lb. compressor piston than would be required for a heavier piston in

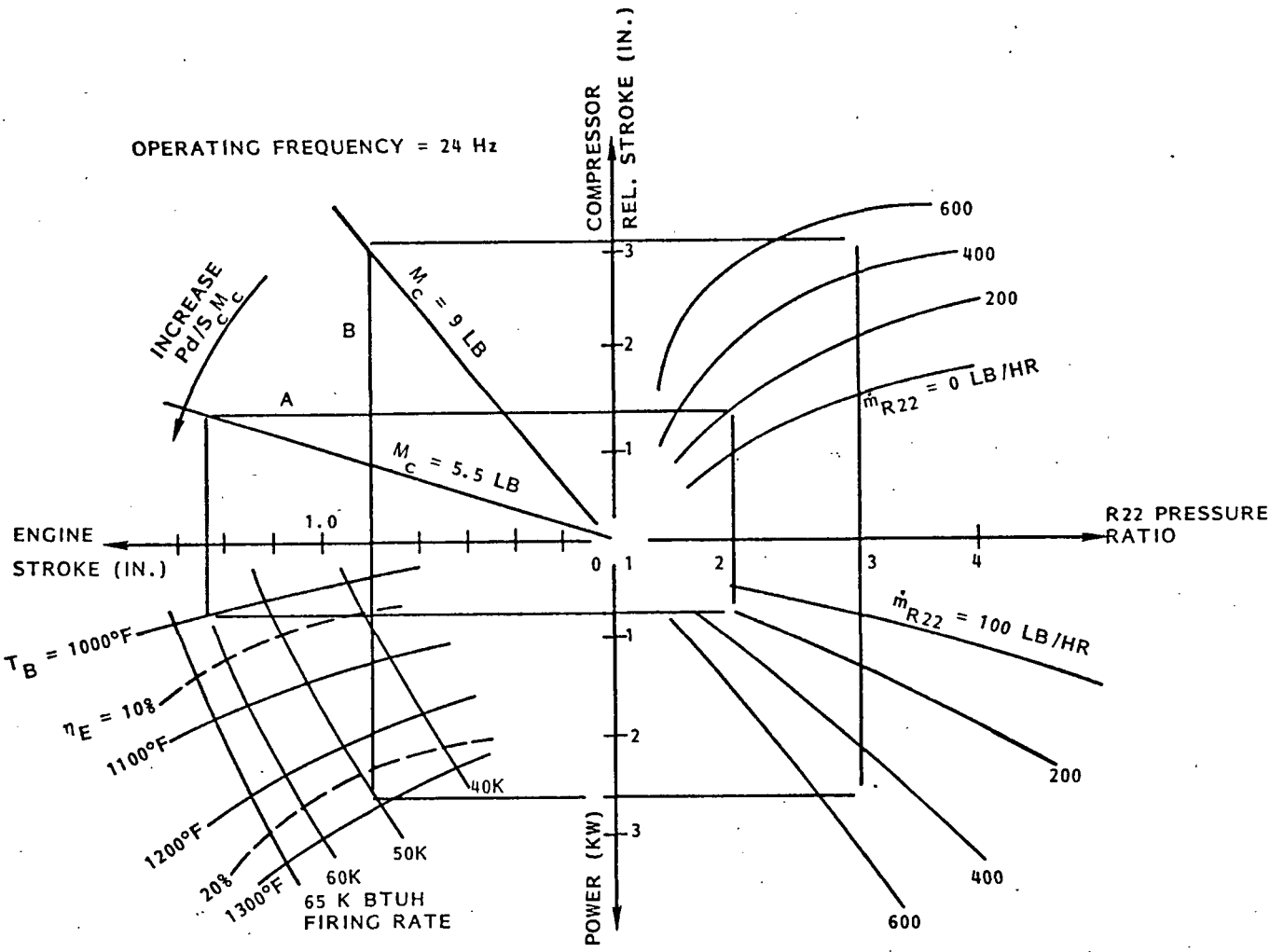


Fig. 9. - Prototype 1 matching characteristics

order to obtain an identical compressor compression ratio (i.e., compressor relative stroke). As a result, the equivalent operating condition using a 5.5 lb. compressor piston is seen to form the series of closed lines A in Fig. 9 as compared to the closed lines B for the heavier compressor piston. A large engine stroke using the light compressor resulted in a low engine heater head temperature and a low engine efficiency, low freon flow and reduced compressor pressure ratio. Using a heavier compressor piston to more closely match the resonant condition between the compressor natural frequency and the engine operating frequency, enabled the engine to operate at a higher heater head temperature and improved system performance. This illustrates the importance of proper system matching between the free piston Stirling engine and a linear compressor, and the type of component modifications that can be made to achieve that matching.

SYSTEM PERFORMANCE

System tests were conducted in both the cooling mode and the heating mode. Fig. 10 depicts the system energy balance for a typical cooling mode set of test conditions.

In addition to a lower than anticipated engine efficiency, the system performance was further penalized by the excessive level of superheat experienced in the compressor suction lines. As indicated in Fig. 10, the heat conduction and the flow leakage from the engine working space to the bounce space, which contains the compressor, represented a significant portion of the available total energy which could have produced useful work. A portion of this energy was transferred to the freon suction gas resulting in a substantially higher level of superheat than initially anticipated and thus reduced the capacity of the compressor. Fig. 11 shows the effect of the superheat on the compressor performance. Consequently, the overall system performance was lower than the product design goals. This superheat problem will be corrected in the next prototype unit by incorporating an improved type of insulation on the freon suction lines and by reducing the quantity of hot working fluid that leaks from the working space to the bounce space.

Measured system capacity and COP are shown in Figs. 12 and 13 respectively for a constant firing rate of 40 KBTUH. Improvement in system performance has been progressively accomplished with design refinements resulting in 80% of the design goals being realized. An improvement in engine performance represents the most promising area to address in order to yield a more efficient heat activated heat pump product. As discussed earlier, an engine efficiency of 30% is considered to be achievable in the next prototype unit. This improvement in engine performance will then enable the system performance to meet and/or exceed the design goals.

CONCLUSIONS

The subsystem level testing indicated that, except for the free-piston Stirling engine, the performance goals of the prototype components were essentially met. The reduced engine performance resulted in reduced system performance. Methods to improve the engine efficiency have been identified and implemented into the design of the next prototype. The potential improvements have been verified analytically and the product performance goals are achievable.

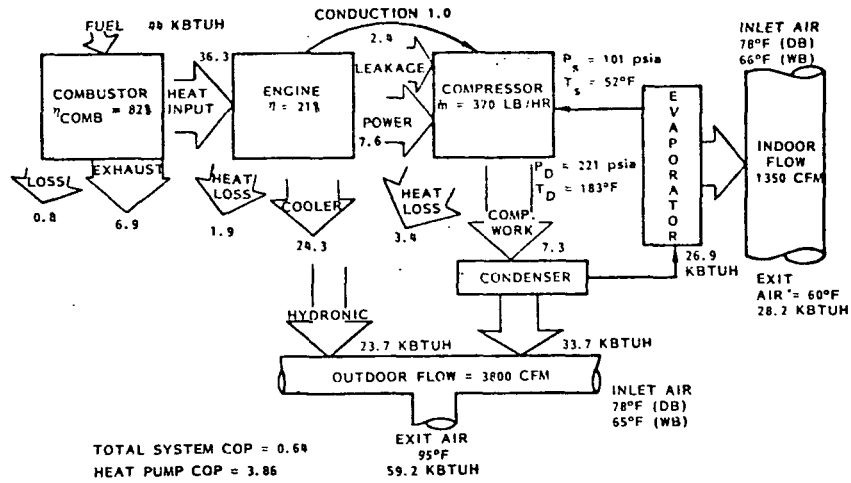


Fig. 10. - System energy balance for a typical cooling mode operation

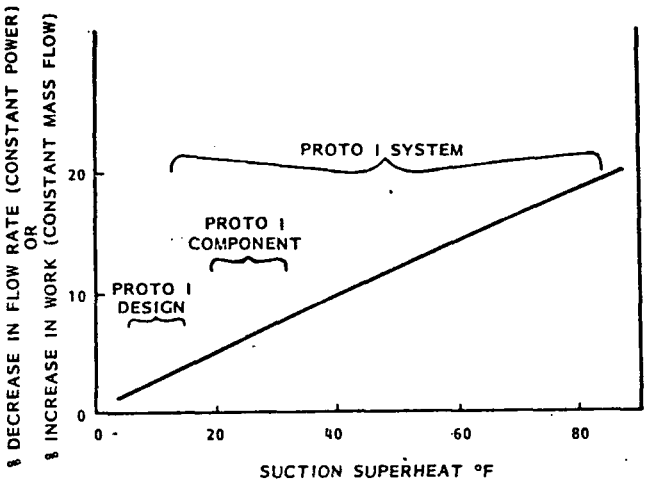


Fig. 11. - Effect of superheat on compressor work

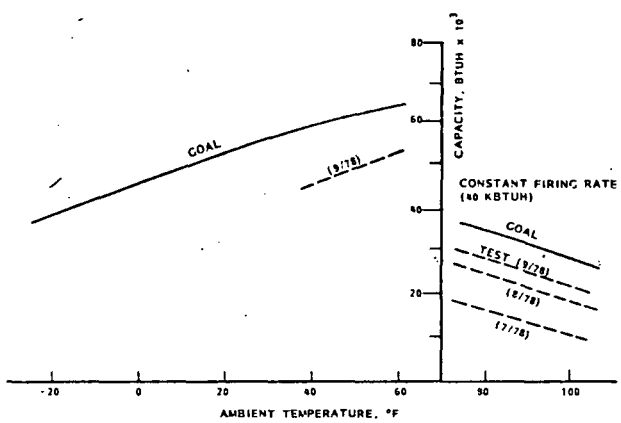


Fig. 12. - System capacity at a constant firing rate

The first prototype HAHP system has served to define the system operating characteristics and to identify pertinent parameters which influence system matching. With an over 30% engine efficiency predicted for the second prototype engine, the system performance of the HAHP product will result in a product that will both meet its design goals and offer substantial savings in energy consumption.

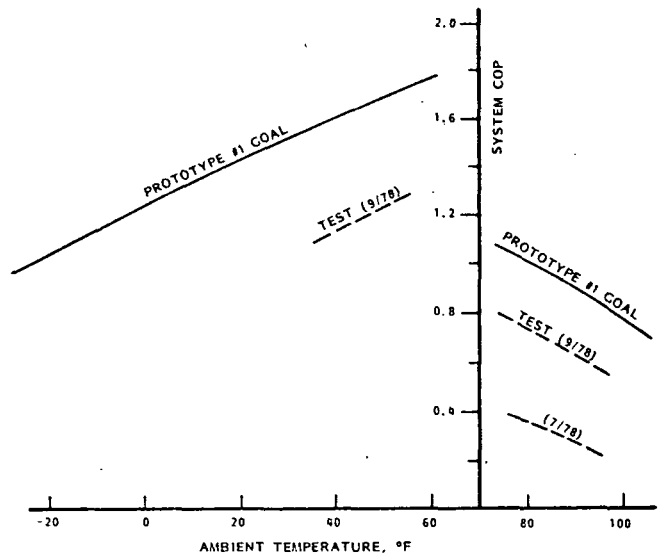


Fig. 13. - System COP at a constant firing rate

REFERENCES

1. W.L. Auxer, "Development of a Stirling Engine Powered Heat Activated Heat Pump," Paper No. 779065, 12th Intersociety Energy Conversion Engineering Conference.
2. B.R. Maxwell, and D.A. Didion, "An Experimental Evaluation of Engine-Driven Heat Pump Systems," Winter Annual Meeting, ASME, December, 1978.
3. W.D.C. Richards and W.L. Auxer, "Performance of a Stirling Engine Powered Heat Activated Heat Pump," Paper No. 789453, 13th Intersociety Energy Conversion Engineering Conference.
4. T.J. Marusak and W.S. Chiu, "The Matching of a Free Piston Stirling Engine Coupled with a Free Piston Linear Compressor for a Heat Pump Application," Paper No. 789272, 13th Intersociety Energy Conversion Engineering Conference.
5. W.S. Chiu and W.B. Carlson, "Performance of a Free Piston Stirling Engine for a Heat Pump Application," to be presented at the Stirling Engine Session, 14th Intersociety Energy Conversion Engineering Conference, 1979.