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A Magnetically Coupled Stirling Engine Driven Heat Pump:

Design Optimization and Operating Cost Analysis

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ABSTRACT

A preliminary design for a 2nd generation, gas-fired free-piston Stirling engine driven heat pump has been developed which incorporates a linear magnetic coupling to drive the refrigerant compressor piston. The Mark II machine is intended for the residential heat pump market and has 3 Ton cooling capacity. The new heat pump is an evolutionary design based on the Mark I free-piston machine which was successfully developed and independently tested by a major heat pump/air conditioning manufacturer. This paper briefly describes test results that were obtained with the Mark I machine and then presents the design and operating cost analysis for the Mark II heat pump. Operating costs by month are given for both Chicago and Atlanta. A summary of the manufacturing cost estimates obtained from Pioneer Engineering and Manufacturing Company (PEM) are also given.

INTRODUCTION

A compact, efficient Stirling engine-driven refrigerant compressor has been designed and analyzed as part of an ongoing program to develop a cost competitive natural gas-fired heat actuated heat pump (HAHP). The preliminary Mark II design has also been valued engineered so that manufacturing cost in production quantities has the potential to meet target values, allowing the unit to compete with existing heat pumps. The cost advantage of the new machine appears to be especially favorable in cold climates where heating mode operation predominates.

The design of both the Mark I and Mark II heat pumps is based on the use of a free-piston Stirling engine in which the refrigerant compressor piston is coupled to the power piston of the Stirling engine. A free-piston machine is especially well suited to the advanced heat pump application because of the continuously variable stroke available on both the displacer and the power piston, even though operating frequency

remains fixed at 60 Hz. Power output of the engine and resulting refrigerant flow rate are fully modulating and the need for on-off control is eliminated at most ambient conditions. The approach used at MTI for stroke control is to incorporate a linear motor to aid in driving the displacer. The motor's electric power consumption is very small except at the extreme operating points, i.e. the 115° ambient cooling condition and the 0° ambient heating point. The displacer linear motor also functions as a reliable starting device for the HAHP.

Free-piston Stirling engines have been under development in the United States for approximately 20 years. Refs. (1-3) provide an introduction to their operating principles and applications. Typically, the working fluid in the Stirling engine is high pressure helium which is hermetically contained. In most HAHP's studied to date (4-6), the heat pump working fluid is Freon-22. The Mark I HAHP utilized an oil-filled transmission with two diaphragms to separate the helium from the R-22. More recent designs such as the Mark II HAHP, utilize a linear magnetic coupling as a power transmission and fluid containment device. The hermetic shell of the coupling must be non-magnetic, thin-walled, and also strong enough to contain the high pressure helium. Refs. (7-8) describe the design of the Mark II magnetically coupled HAHP and also a similar machine designed by Sunpower, Inc.

Following the description of Mark I performance, this paper will discuss: 1) optimization results obtained for the Mark II design, 2) system analysis performed to calculate binned performance and annual operating cost, and 3) manufacturing cost estimates obtained through work with Pioneer (PEM). The Mark II HAHP is shown to have excellent potential as the primary component in an alternative residential heating and cooling system.

SUMMARY OF MARK I PERFORMANCE

Two Mark I heat pump systems, equipped with the direct acting transmission and tuned vibration absorbers, were tested during 1987 and early 1988. One of these units was tested at Lennox Industries and the other was used for continued development testing at MTI.

The Mark I "system package", assembled for testing at Lennox, was prepared at MTI and shipped to the Lennox Engineering Center in March, 1987. The unit was installed in an environmental test cell and connected to their existing 5-ton calorimeter loop. Figure 1 shows the Lennox installation. Testing at the Lennox facility involved over 140 hours of operating time and provided performance data at 72 test points ranging over equivalent ambient temperature conditions from 0° to 105°F. The measured preformance at Lennox was very good except at conditions involving low engine output power. This was particularly evident at the 47° ambient where the resulting thermal COP dropped from 1.62 at high capacity to 1.21 at a capacity near the load line. In part, this tendency is inherent in free-piston Stirling engines where maximum efficiency is achieved near the point of maximum output power. However, the Mark I as tuned for testing at that time also required large displacer damping power to be applied for modulation to low power (the displacer linear motor was acting as an alternator and damping power of about -275 W was used at fully modulated conditions). No credit was taken for this displacer damping power and therefore the engine's thermal efficiency was very low. A prime objective in the design of the Mark II is to obtain the required range of modulation while limiting the negative displacer motor power to about -50 W.

Testing and development of the Mark I continued until February of 1988. Improvements such as adding a lightly contacting split-ring seal to the displacer, running the machine at fixed 60 Hz frequency, and utilizing PTFE-based bearing materials were made. Performance of the Mark I continued to improve above that obtained in the Lennox testing. It will be useful to compare the final measured performance at the 95° ambient to goals that were set in 1981. Table I provides this comparison.

The laboratory reliability of the Mark I was more than adequate and its performance would have been attractive at all levels of modulation if better matching analysis, optimization, and tuning procedures had been employed. However, the manufacturing cost was found to be excessive. This was due primarily to the size and weight of the transmission housing, the use of diaphragms to contain the oil, and the need for an oil management system to recover and reinject any oil that leaks past the reciprocating shaft seals. A new lower end design, incorporating a linear magnetic coupling, was proposed to reduce manufacturing cost. This design evolved to one in which dry-lubricated bearings and seals were adopted so that no oil is needed in the transmission. The next section describes this design in detail.

MARK II HEAT PUMP DESIGN AND OPTIMIZATION

Figure 2 is a layout drawing of the preliminary Mark II HAHP design with magnetic coupling. Key components are labeled. Since the total mass of reciprocating components is approximately 20.2 lb_m, vibration from the unit would be quite severe unless appropriate measures are taken. The solution chosen in this design is to use a pair of tuned vibration absorbers mounted on torsion bars. These components are not shown in Figure 2 but their manufacturing cost was included in the totals discussed later.

Three engine components with relatively high cost are the heater head, the regenerator, and the magnetic coupling. A discussion of materials chosen for these components is given below. The current design utilizes high temperature, iron-based superalloys for the heater head (Incoloy 800 heater tubes and XF818 casting alloy for the head). Although far less expensive than the nickel-based Inconels, these materials still allow 750°C heater head temperature. Brunswick feltmetal is specified for the regenerator because of its ready availability and low cost relative to stacked screen (future development of the foil-wrapped regenerator may lead to its specification in place of feltmetal). The magnetic coupling utilizes Magnequench MQ II magnets (neodymium-iron-boron) which are projected to cost around $20/lb.

The magnetic coupling does not provide a rigid connection between the power piston and the refrigerant compressor piston. Coupling flexibility can be modeled to first order as a linear spring and damper with stiffness between 2000 and 5000 lb/in., depending on the design. Because of this flexibility, an additional degree-of-freedom is added to the system and the thermodynamic/dynamic analysis now includes 5 dynamic degrees-of-freedom. These are: displacer motion, motor coil current, motor eddy currents, power piston motion, and compressor piston motion. As part of the engine/compressor matching analysis described below, compressor piston stroke is determined prior to the engine dynamic analysis and, therefore, only four state variables need to be found. The system is assumed to behave harmonically; therefore closed form solutions for the four remaining degrees-of-freedom can be obtained. These equations are programmed in a "function" subprogram that is iterated with the engine's thermodynamic model. This is done automatically until specified convergence criteria are met. Lower end dynamics, i.e. power piston, magnetic coupling, and compressor piston, are represented by the schematic in Figure 3.

Engine/compressor matching analysis was performed to determine the heat pump behavior at several heating and cooling mode operating points. Capacity equals the load line requirement at all the ambients between 95° to 80° in cooling mode and 17° to 47° in heating mode. Compressor conditions at each operating point (flowrate, suction pressure, and discharge pressure) were identified using system model results provided by Lennox. Then the required power, stroke, offset, and equivalent spring rate were determined by modeling the compressor using MTI's CYLINDER code. Once these four quantities are known at each point, matching analysis is used to determine the behavior of the engine. Several adjustments affecting system tuning are made until the behavior is acceptable at all operating points. This procedure was performed for several different engine configurations as part of the optimization study described below.

The optimization study involved starting with an engine design that had been thermodynamically optimized at low power and then investigating how various changes affected the efficiency and motor power requirement at all operating points of interest. Parameters that were varied include heater tube ID, regenerator porosity, regenerator length, power piston area, and displacer area. Figure 4 shows the cycle efficiency and motor power results for three different engine configurations that resulted from numerous trials analyzed with the thermodynamic/dynamic engine code. Case 3 was chosen as the Mark II baseline design because the motor power requirements are quite small and it also has the advantage of 26% lower piston mass than Case 2 (resulting in lighter vibration absorbers) and 25% shorter regenerator. Table II gives the matching analysis results at 8 key operating points for the baseline engine design.

MARK II OPERATING COST

The next step in Mark II performance analysis is to calculate the annual operating cost (gas and electric) in two population centers with different climates and utility rate structures. Chicago and Atlanta were chosen. Figure 5 shows the load line for a typical 4-bedroom house in either city and also gives the number of hours per year at each ambient temperature in Chicago and Atlanta. Gas and electric rates for 1988 were obtained and entered into a spread sheet cost analysis program. Operating costs must be computed by month because the utility rates are formulated in that way and rates typically vary depending on season.

Heat pump system performance, including electric parasitics, is summarized in a bin analysis program that allows ready analysis of different engine operating characteristics. Using engine performance data from Table II, and information concerning combustor efficiency, cycling loss behavior, and electric parasitics, the program computes average gas and electric consumption rates in each temperature bin. Heat pump COP (total heat delivered divided by gas firing rate), and system COP including parasitics are also calculated. Figure 6 is a plot of both heat pump and system COP vs ambient temperature. The calculated average gas and electric consumption rates in each bin are transferred to the monthly cost analysis programs to compute operating cost.

The operating cost analysis for the baseline design showed that total annual operating cost is $789.95 in Chicago and $548.12 in Atlanta. The operating costs by month are shown in Figures 7 and 8. Table III gives the breakdown of annual gas and electric costs and also summarizes the changes in cost for several perturbations of engine performance. The perturbations analyzed include reduced thermodynamic cycle efficiency, reduced combustor efficiency, and modified power consumption of the combustion air blower. Table III shows that the penalty associated with reduced combustor efficiency is much greater than that for reduced cycle efficiency. The reason for this is because, in heating mode, any increase in engine rejected heat is recovered and transferred to the indoor coil while any increase in combustor heat loss is simply lost to ambient.

The cost of electric parasitics is quite large in comparison to the cost of natural gas, especially in Chicago. The electric parasitics assumed for the baseline system (in addition to displacer motor power) are as follows:

1) combustion air blower - constant consumption at 233 W (airflow is modulated via bypass or gate valve),
2) coolant circulating pump - constant consumption at 50 W,
3) indoor air fanfully modulating at 301 W maximum to 25 W minimum, and
4) outdoor air fan - 2 speed at 207 W maximum to 41 W minimum.

The estimates for combustion blower and coolant pump power are quite conservative. The combustion blower power can probably be reduced through use of more sophisticated modulation controls and/or combustor design improvements. The modulating indoor fan and 2-speed outdoor fan are expected to be standard practice for future high efficiency systems.

The last entry in Table III indicates the savings in annual operating cost associated with a modulating combustion air blower. The improved blower motor is assumed to consume one half of 233 W at the lowest firing rate modeled, 2.72 kW, and consume 233 W at the 95° point where firing rate is 9.06 kW. The annual savings associated with this change is $55.74 in Chicago and $31.39 in Atlanta. Clearly, additional design or controls work should be pursued to minimize the combustion air blower power consumption.

It will be useful to compare the baseline system operating costs to similar costs that were calculated for an advanced electric heat pump tested by Westinghouse/DOE in 1985 and 1986. This machine used a dual stroke compressor and other high efficiency components; field test data was taken and published in terms of EER for all the bins of interest. Using the MTI monthly cost analysis programs, the annual operating cost of the Dual-Stroke Electric was found to be $1481.98 in Chicago and $861.87 in Atlanta. The operating cost advantage of the baseline Mark II is therefore $692.03 in Chicago and $313.75 in Atlanta. When one considers the expected manufacturing cost premium of the Mark II baseline system, the unit appears to be competitive in Chicago but may not be competitive in Atlanta.

MANUFACTURING COST ANALYSES

For a FPSE HAHP system to be economically viable in a residential application, it was determined that the system manufacturing cost could not exceed $1250 (Figure 9 defines the system components to be included). This amount was a result of an economic analysis done in the 1985-86 time period (ref ?). At that time, cost goals of $250 each for the combustor, engine, transmission/compressor, heat recovery system, and controls/accessories were arbitrarily set.

To determine manufacturing cost, analyses were performed on all the subsystems shown in Figure 9. For manufactured hardware items (the combustor, the FPSE, the transmission/compressor assembly, and the combustion air blower), layout drawings and detailed parts and assembly drawings were produced. These were sufficiently detailed to allow experienced manufacturing cost estimators to accomplish the following as required:

1) identify raw material requirements
2) establish raw material costs
3) establish piece part manufacturing process
4) prepare piece part routing sheets (including operation times)
5) estimate operating times (labor minutes)
6) establish machine overhead rates
7) estimate scrap rates
8) estimate tooling costs

The cost estimating work was accomplished by personnel at Deere & Co. and Pioneer Engineering and Manufacturing Co (PEM) at different times during the course of the program.

For the controls, auxiliaries, and heat recovery system, detailed schematic and block diagrams were produced. These were accompanied with either a complete part description or component part specification. This information allowed PEM to establish the manufacturing and procurement costs of these subsystems.

Several design iterations on the major hardware items (engine, transmission and compressor) were made with each one resulting in a significant percentage manufacturing cost reduction from the preceding design. The following table summarizes the total engine/transmission/compressor assembly manufacturing costs at various selected time periods.

DESIGN	TOTAL MFG. COST
MARK I (exclusive of regenerator, displacer motor, & final assy & test)	$5381
MARK II	1711
MARK IIVE Engine and Lower End	1226
MARK IIVE with MARK 11B Lower End	1014
MARK II BASELINE (Current design shown in Fig 2)	665

From this table it is seen that much has been accomplished towards achieving the original cost goal of $500 for the combined engine/ transmission/compressor assembly. The table summarizes the manufacturing costs of the major subsystems in terms of original goals, current estimates for the Baseline Mark II, and current projections. The projections are based upon estimates of reductions that should result from changes that are planned to be made in the next design iteration.

CONCLUSION

The results of the work reported in this paper indicate that the Mark II FPSE HAHP system has significantly better performance potential than the Mark I (from power modulation and annual operating cost viewpoints) and that it has the potential for a manufacturing cost on the order of $1000. Thus, it appears that with further development, the Mark II HAHP can be an economically viable alternative residential heat pump system.

ACKNOWLEDGEMENT

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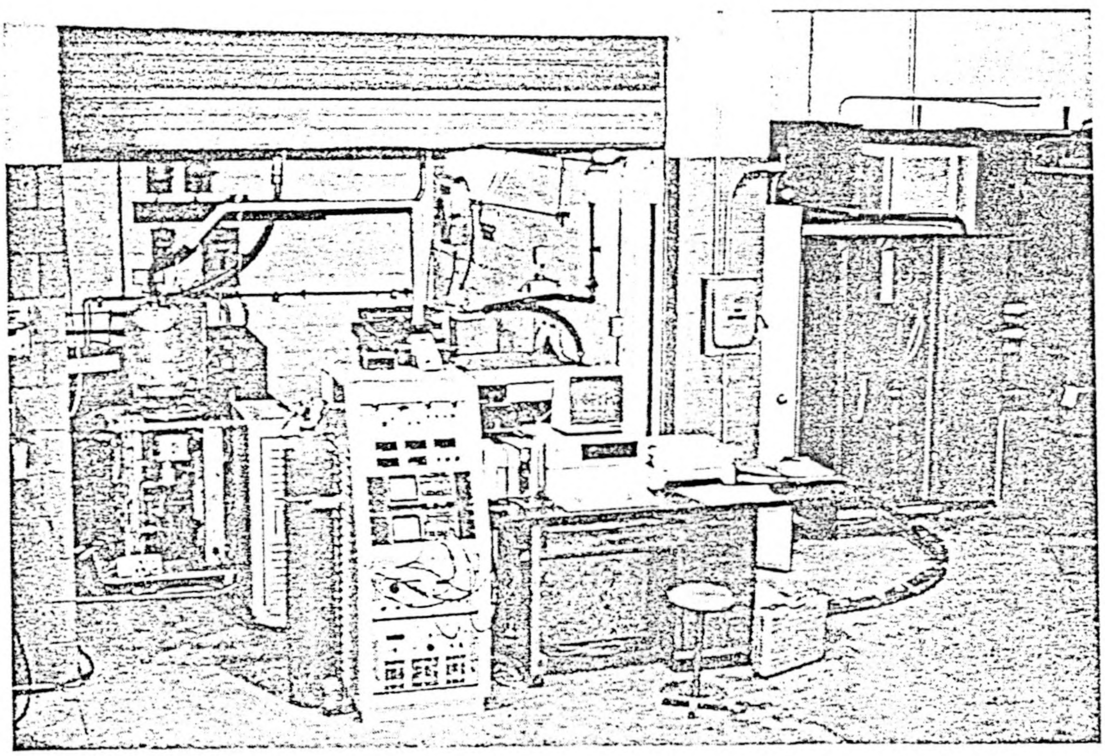


Fig. ~~3-1~~ 1 Mark I System Package Installed in Lennox Test Facility

TABLE I Mark I Performance Compared to Original Targets

PARAMETER		FPSE/HP TARGETS	MARK I FPSE/HP MEASURED DATA
Date		1981	Scan 32, 1-6-83
Ambient Temperature	(°F)	95	95
Coolant Temperature	(°F)	80	83
Capacity	(RT)	3.0	3.0
Displacer Motor Power	(watts)	<500	612
Engine Efficiency	(%)	27.5	25.3
Hydraulic Trans Effic	(%)	82.7	83.7
Compressor Isentropic Efficiency	(%)	83.2	81.5
Lower End COP		3.63	3.78
HP Thermal COP		1.00	0.96

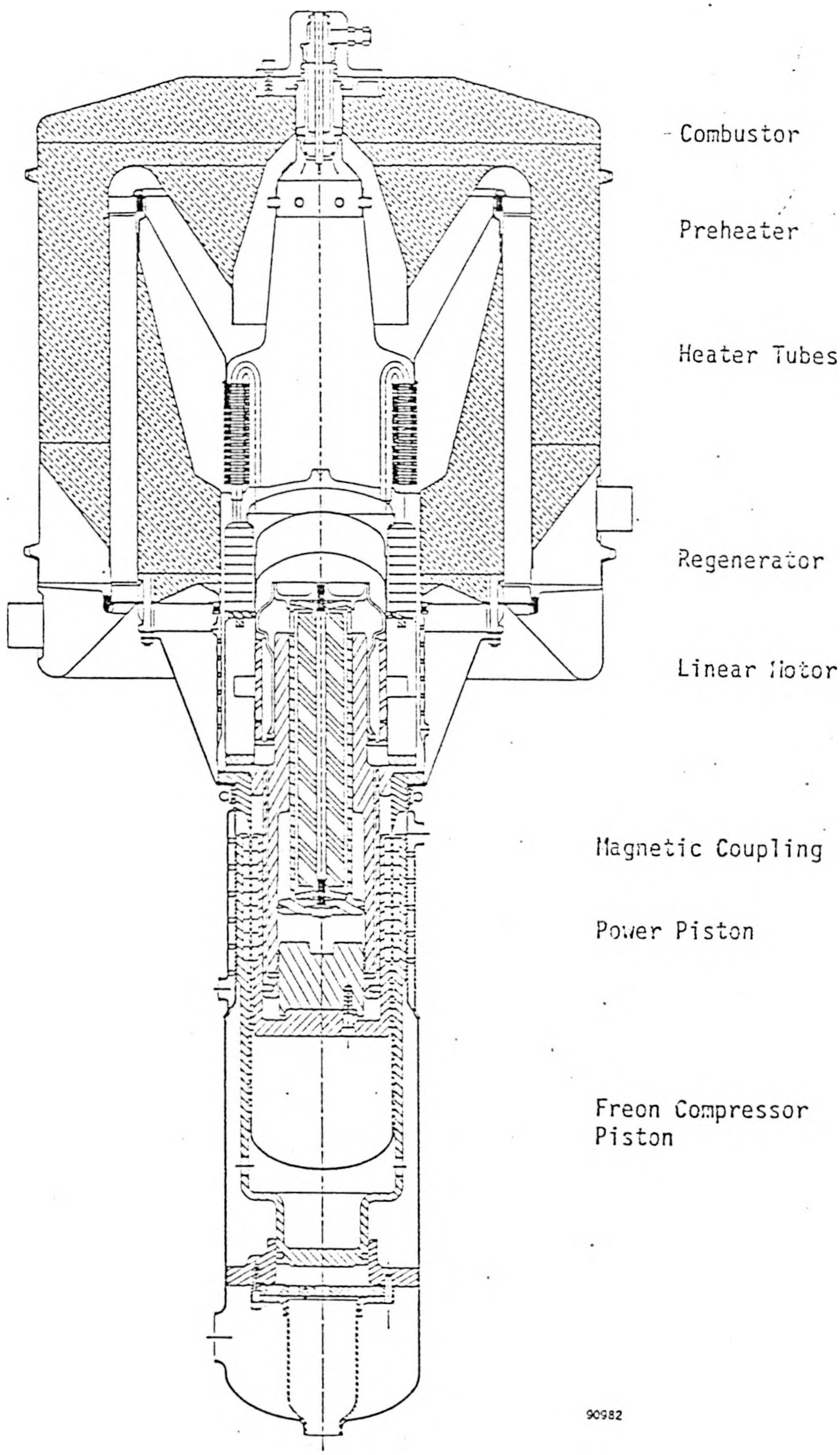
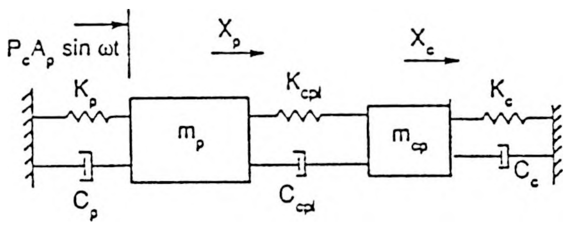


Fig. 2 Layout of Mark II Baseline Engine/Compressor



Couples to Thermodynamics via:

$$P_c A_p = (P_{lp} X_p + P_{ld} X_d) \times A_p$$

m_p = 10.58 lb_m
m_{cp} = 4.37 lb_m
K_{cpl} = 3000 lb/in.
C_{cpl} = 264.8 lb-sec/in.

m_{cp} Chosen to Minimize Coupling Force

m_p Is Determined So That:

$$\Delta P_{\text{Lower Gas Spring}} = \Delta P_{\text{Engine}} \text{ at } 32° \text{ Point}$$

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Fig. 3 Dynamic Model of Lower End

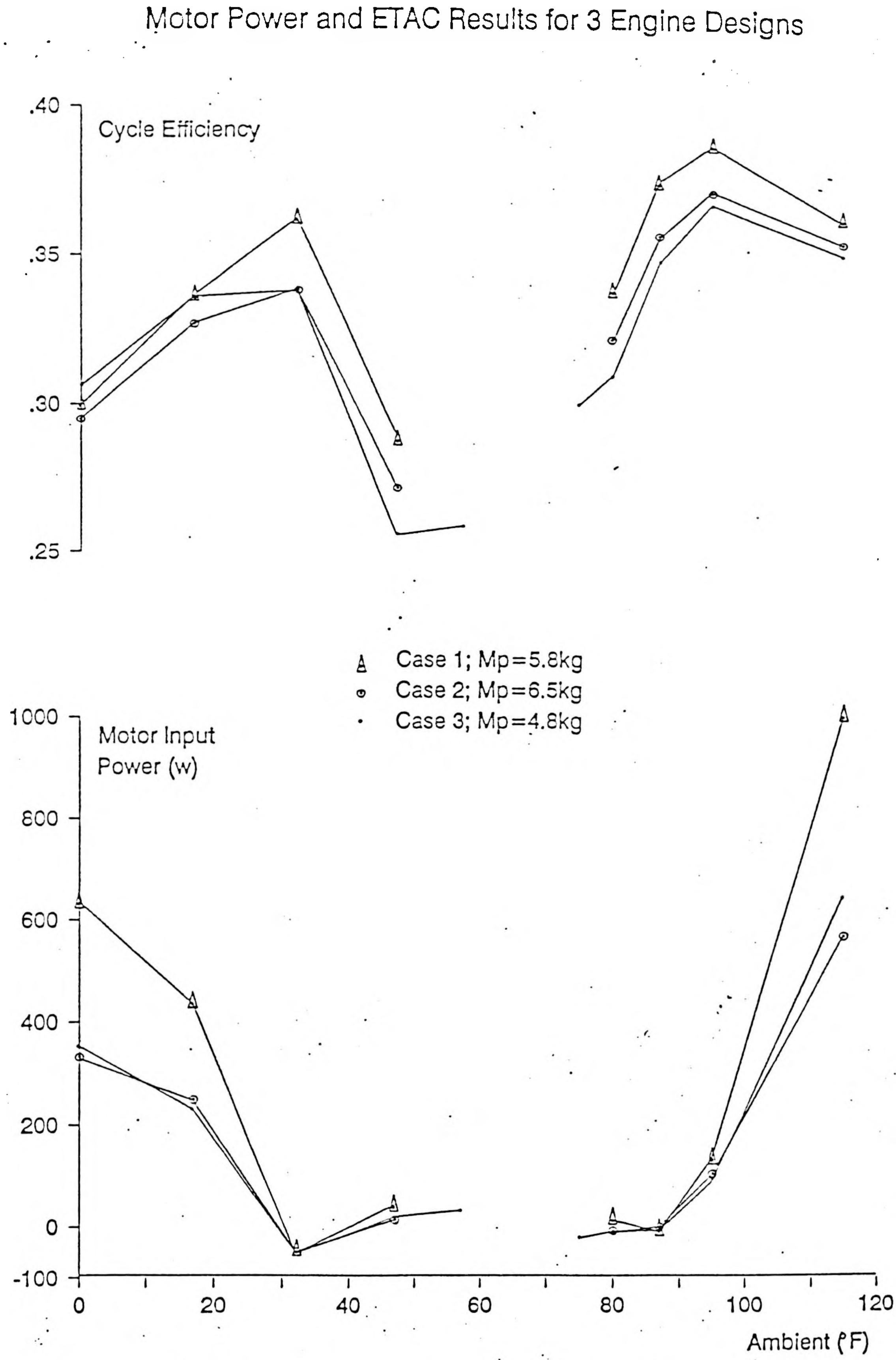

Fig. 4 Engine Optimization Results

Table II

Table ~~4-7~~. Mark II Baseline Engine Matching Performance

Piston Mass = 4.8 kg; ΔP_{engine} = 5.90 bar (95°F Point); Regenerator Length = 1.708 in.; Porosity = 0.90; Heater Tube ID = 5 mm; X_p = Piston Stroke; X_d = Displacer Stroke; ϕ_D = Displacer Phase Angle; PC = Compressor Space Pressure Amplitude; $\eta_{pV} = pV_{piston}/Q_{head}$; $\dot{m}$ = ~~Refrigerant~~ Mass Flow Rate

Ambient (°F)	$\dot{m}$ (lb/hr)	Capacity (Btu/hr)	X_p (mm)	X_d (mm)	ϕ_D (deg.)	PC (bar)	pV_{piston} (W)	η_{cycle}	η_{pV}	Transmission Loss (W)	P_{motor} (W)
115	493.2	30,400	24.76	22.32	40.3	5.95	3071	0.3495	0.3456	344	648.9
95	516.2	35,940	20.06	16.28	60.2	5.90	2517	0.3668	0.3458	232	92.5
87	359.0	26,740	16.29	11.56	60.5	4.85	1470	0.3472	0.3186	152	-6.0
80	239.7	18,605	13.43	8.63	56.2	3.90	845	0.3095	0.2790	101	-8.1
75	227.0	17,960	13.10	7.73	60.8	3.94	767	0.3048	0.2726	97	-20.4
57	149.4	18,250	11.75	7.90	42.9	3.08	528	0.2585	0.2331	76	32.7
47	118.0	14,390	12.36	7.11	44.1	3.38	489	0.2556	0.2256	84	20.9
32	208.8	27,800	17.31	8.93	74.3	5.68	1262	0.3378	0.3017	172	-50.9
17	279.1	40,310	19.96	12.61	108.0	8.01	2115	0.3358	0.3240	251	230.8
0	187.0	31,430	20.06	11.34	120.6	8.42	1697	0.3061	0.2998	259	353.5

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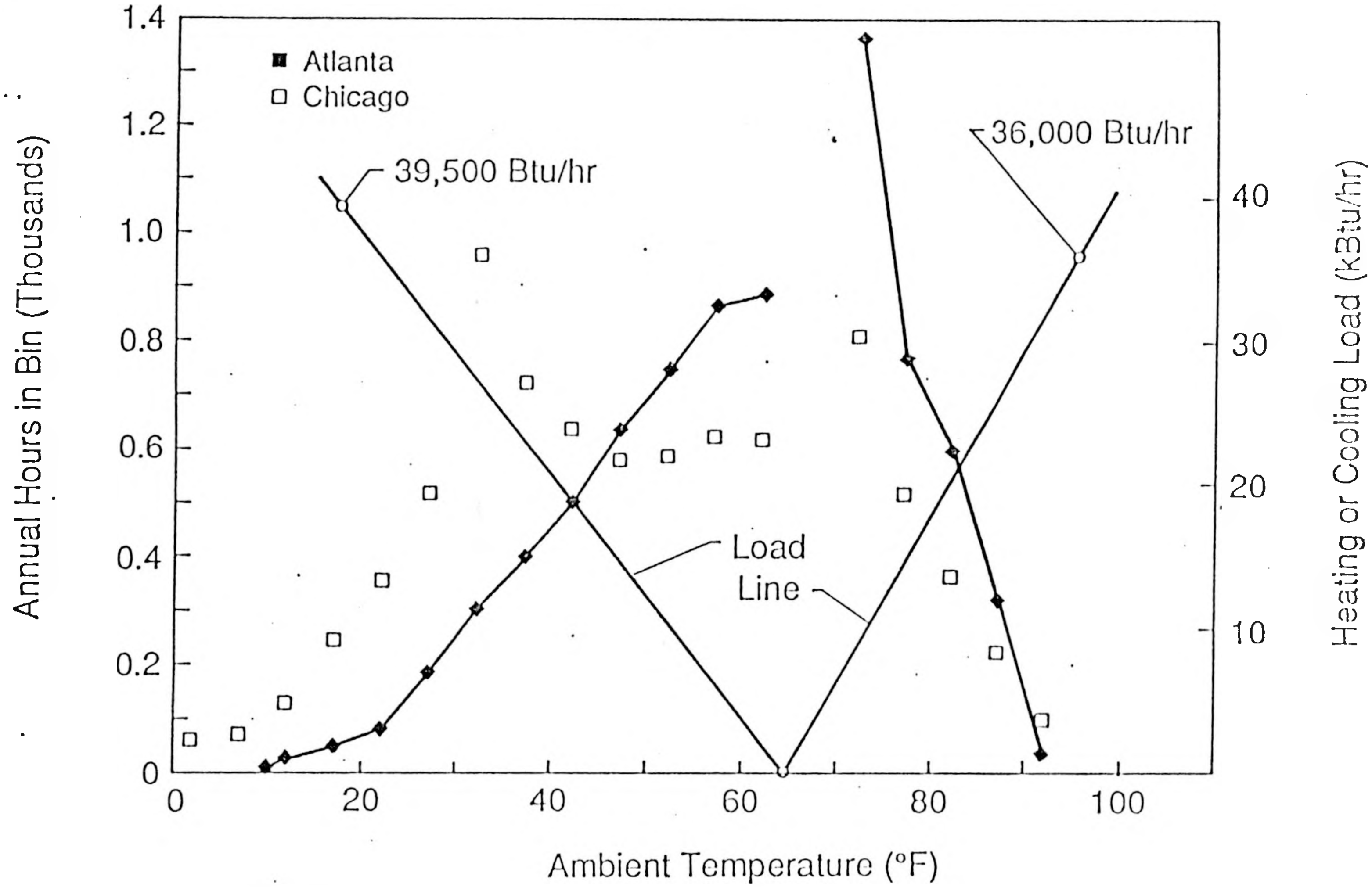

Fig. 5 Load Line and Weather Data for Chicago and Atlanta

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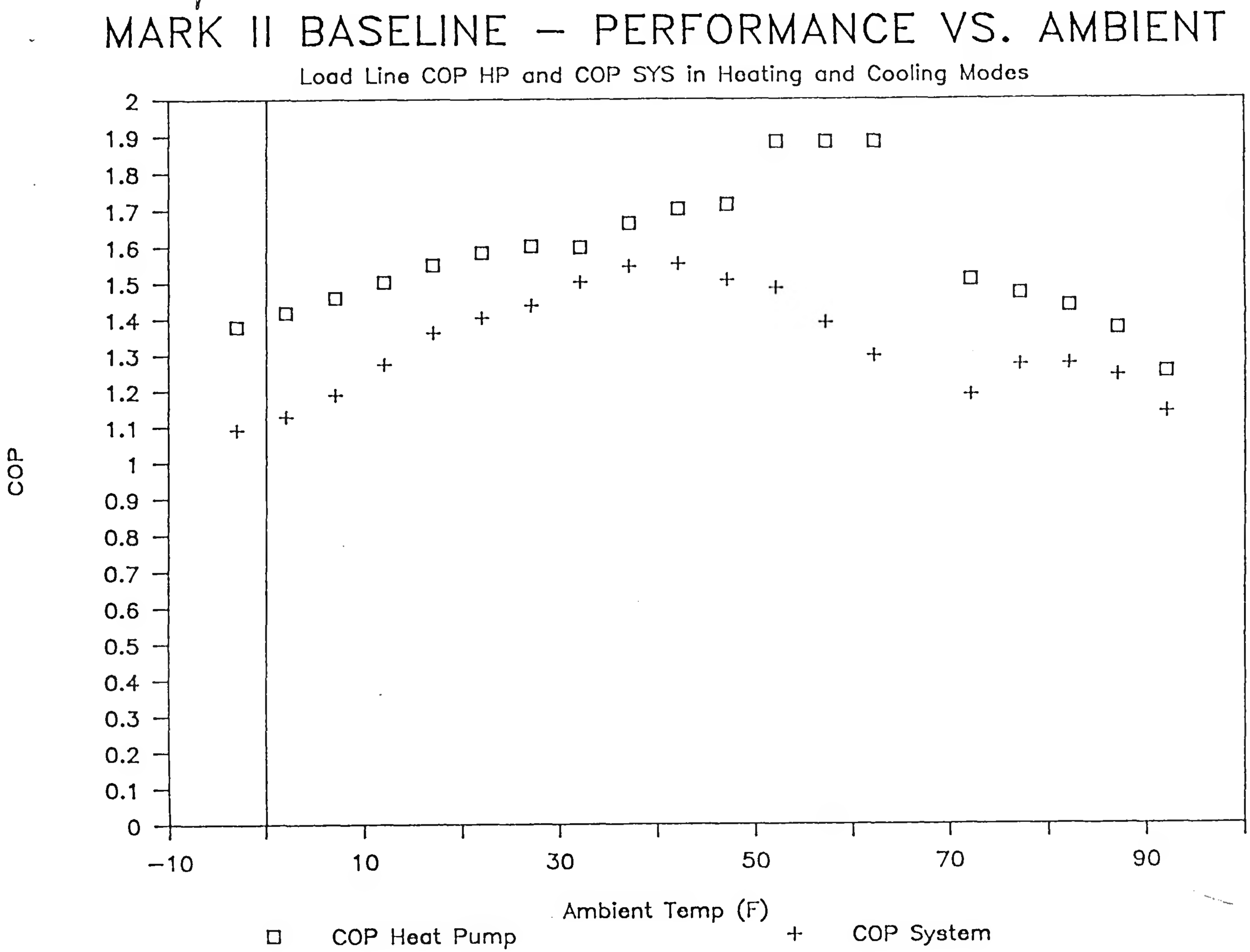
Fig. 6


MARK II BASELINE — PERFORMANCE VS. AMBIENT
Load Line COP HP and COP SYS in Heating and Cooling Modes
COP
2
1.9
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
−10
10
30
50
70
90
Ambient Temp (F)
COP Heat Pump
COP System

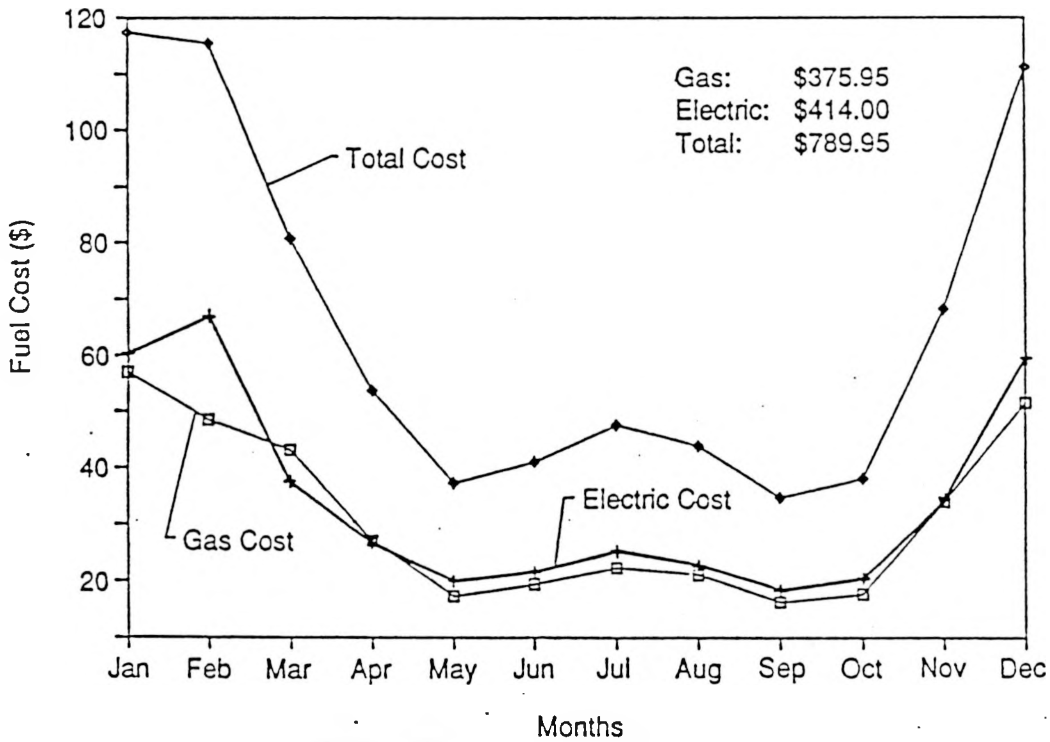


Fig. 7 Monthly Operating Cost in Chicago

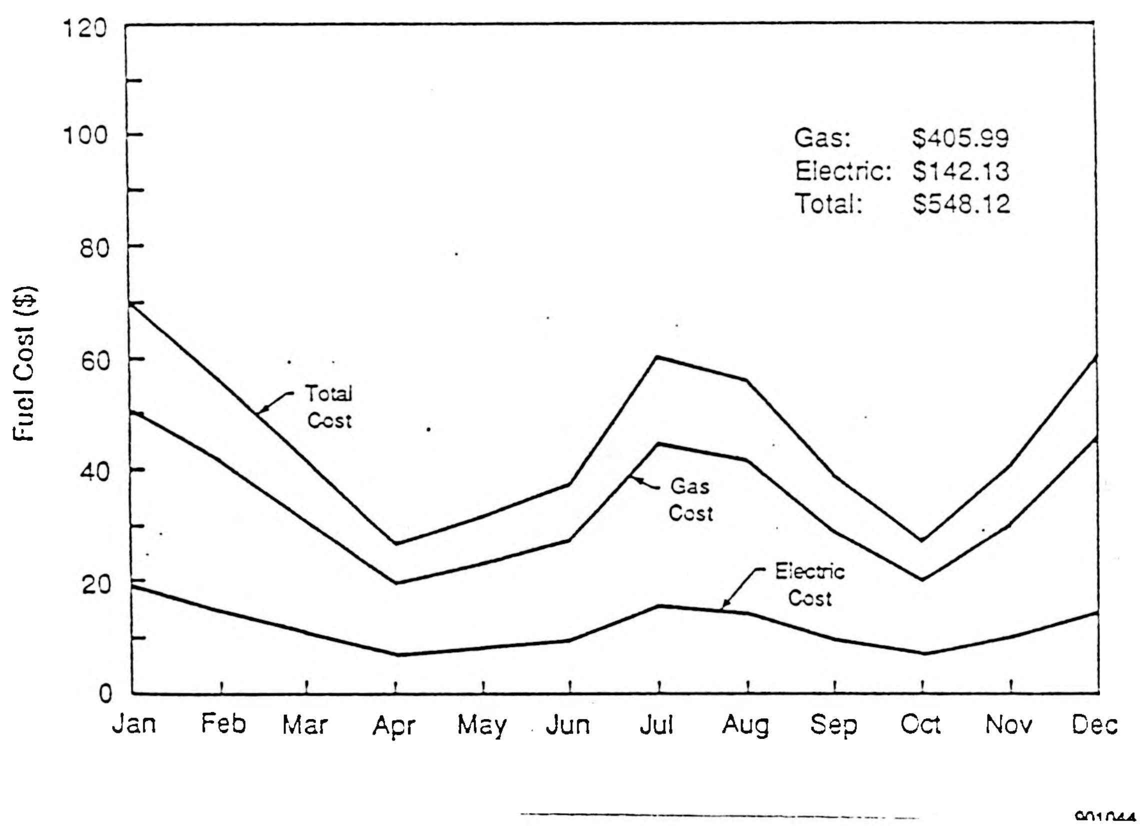


Fig. 8 Monthly Operating Cost in Atlanta

III

Table ~~5-2.~~ Fuel Costs for Various Efficiency Changes

Configuration	City	Gas Cost ($)	Electric Cost ($)	Total ($)	Change from Baseline ($)
Corrected Baseline	Chicago Atlanta	375.95 405.99	414.00 142.13	789.95 548.12	0 0
$0.9 \times \eta_{cycle}$	Chicago Atlanta	400.33 436.14	404.38 140.14	804.71 576.28	+14.76 +28.16
$0.75 \times \eta_{cycle}$	Chicago Atlanta	445.20 495.33	385.50 136.36	830.70 631.69	+40.75 +83.57
$0.9 \times \eta_{combustor}$	Chicago Atlanta	415.44 448.29	414.00 142.13	829.44 590.42	+39.49 +42.30
$0.75 \times \eta_{combustor}$	Chicago Atlanta	491.02 532.65	414.00 142.13	905.02 674.78	+115.07 +126.66
Modulated Blower Power Assumed	Chicago Atlanta	375.95 405.99	358.26 110.74	734.21 516.73	-55.74 -31.39

90TR25

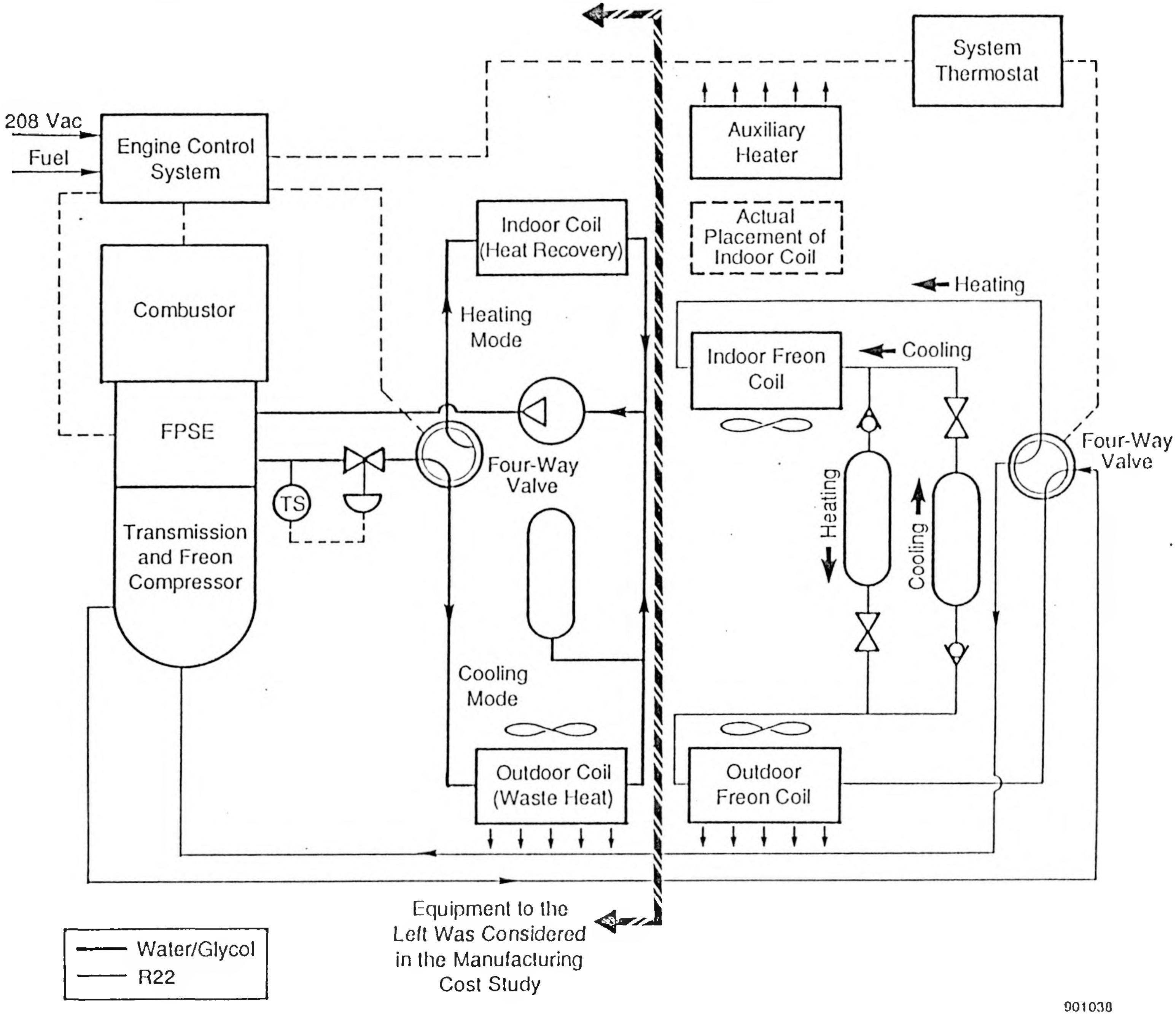


Fig. 9 Heat Pump System Schematic