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SOLAR ASSISTED HEAT PUMPS: A POSSIBLE WAVE OF THE FUTURE

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16. Abstract
Heat pumps have been used sparingly in residential construction and less frequently in commercial construction probably due to higher capital costs, maintenance costs, and a lack of understanding of the advantages. With the higher costs of electric power and the widespread interest to use solar energy to reduce the national dependence on fossil fuels, heat pumps are examined to determine their suitability for use with solar energy systems.

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INTRODUCTION

Electrically driven heat pumps employing ambient air as the heat source in wintertime and the heat sink in summertime have been used but sparingly in residential construction and even less frequently in commercial construction. In the past the lack of heat pump installations could be attributed to:

1. Their slightly higher capital cost compared to a gas furnace and standard air-conditioning. This is a result of lower production volume, the addition of reversing valves and extra controls, and the need to match condenser and evaporator more closely.

2. Their higher maintenance costs, particularly those resulting from the fact that compressor failure is more frequent in heat pump service than in air-conditioning service. This is simply a reflection of the fact that the pressure ratios and hence the temperature ratios across the machine must be higher in wintertime than in summertime so that there are more frequent valve failures.

3. The higher operating cost which occurs whenever system coefficient of performance drops below about 1.5 as it does in most installations at ambient temperatures below 35-38^{o}F. ($\sim$ 2 - 3^{o}C)

4. The general lack of understanding of heat pump operation, advantages, and disadvantages on the part of the public at large.

There have also been a number of heat pump installations employing a water well as the heat source or sink. These installations have generally lower operating costs than units using the ambient air as the source/sink because the temperature of the ground water is, on the average, closer to the inside temperature desired for the building than is the air temperature. Since the temperature difference over which one must "pump" heat is less with a water source/sink unit, the machine does not work as hard and therefore consumes less power. One has, however, higher capital costs with such a machine because of the need to drill a well and provide a pump and because the condenser/evaporator must be replaced more frequently under the corrosive action of the ground water.

With the current widespread interest in utilizing solar energy to reduce the national dependence on fossil fuels, heat pumps are again being examined to determine their suitability for use with this heat source. Two types come immediately to mind: an absorption system and one employing a mechanical compressor. A rather detailed comparison by a student of the present writer showed that:

1. Both systems use about the same amount of electrical power in the wintertime.

2. The absorption system uses about 20% less power for summertime air-conditioning but all of the usage comes during times of peak electrical demand whereas the mechanical system operates only during off-peak hours.

3. The capital cost of the absorption system is about 20% greater.

The higher capital costs for the absorption system are associated with the need for a large cooling tower and the insulation needed to permit the solar panels to operate efficiently 120^{o}F (66.7^{o}C) above ambient temperature.

Basically, the rationale for a mechanical heat pump operated in this fashion is the following:

The major cost in any solar heating system is the cost of collectors. This is because solar energy is so diffuse ($\sim$ 1000 BTU/ft^2-day) (0.02722 $\frac{\text{kwH}}{\text{M}^2\text{-day}}$) that it requires a large collector area ($\sim$ 1000 ft^2) (92.9 M^2) to supply the heating needs of the average home. To reduce collector unit cost while operating at high efficiency in order to keep collector area at a minimum, it is necessary to operate the collector at near-ambient temperatures. The cost of insulating the panels for operation 110^{o}F (61.1^{o}C) or more above ambient can increase the panel cost by about 50% compared with the insulation needed for efficient operation only 25^{o}F (13.9^{o}C) above ambient. Twenty-five (25)^{o}F (13.9^{o}C) above ambient, of course, is insufficient to provide heat for a home directly -- it must be pumped "uphill" as much as 55^{o}F (30.6^{o}C). The trade off is then between the more complex vapor compressor and its controls and power consumption on the one hand and the additional insulation for the solar panels and a simple circulating pump on the other hand. At this point it appears that a cost savings of \$2-\$3/ft^2(\$21.5 - \$32.3/M^2) for solar panel insulation is achievable by going to the heat pump system. In systems where solar energy provides almost all the heating requirement for the house, this saving is more than sufficient to pay for the vapor compressor and its controls.

The cheapest solar panel one can readily envision is constructed of two aluminum sheets bonded together with preformed heat transfer passages between them. This construction is employed in household refrigerators, for example. Aluminum is, unfortunately, subject to corrosion if water is used as the heat transfer agent. Aluminum, on the other hand, is very compatible with fluorocarbon refrigerants. Many household refrigerators run 20 years or more without trouble.

EXHIBIT HARDWARE

Based on these considerations and little additional analysis the author and several senior mechanical engineering students from North Carolina State University constructed the system shown in Figure 1. Note that the panels are set at the optimum angle for winter heating at this latitude. The effort was made to keep component costs at a minimum. Designed to supply about 12,000 BTU/hr (3.517 kw), the system employs 88 square feet (8.175 M^2) of collector surface. The surface is painted with a special low IR-emission paint from Dow Chemical and panel is covered with 20-mil-thick (1/2 mm) Dupont "Tedlar". The wooden frames - built from 2" x 4" (5.08 cm x 10.16 cm) studs - are insulated on the backside with sprayed-on polyurethane foam. Material cost for the panels was $\$4/ft^2$ ($\$43.05/M^2$). A used automotive air-conditioning compressor was reconditioned for use as the compressor in the system. It was run by a 1 1/2 hp (1.1 kw) electric motor. Two automotive air-conditioning condensers connected in series were placed horizontally in a water tank made from two 55-gallon (208.56 liter) drums and served as the system condenser. The water heated by the freon condensation can be piped to convectors around the house.

No means for circulating the water in the condenser was provided. During field trials this proved to be the limiting factor in the system. By admitting 1.5 gallons (5.685 liters) of water per minute at 76^{O}F (24.9^{O}C) to the tank and allowing the excess hot water to drain out, it was possible to give up 14,400 BTU/hr (4.22 kw) to the water at a condensing temperature of about 180^{O}F (82.2^{O}C) using freon 114 as the working fluid. The tests, conducted in Albuquerque, New Mexico on August 15, 1975, revealed that with the high levels of solar insolation encountered there it was necessary to cover approximately 45% of the solar panels to keep the freon entering the compressor below 125^{O}F (51.7^{O}C) on a 90-95^{O}F ($\sim$ 34^{O}C) day. The power drawn from the line (input to the motor) was about 3740 BTU/hr ($\sim$ 1.10 kw) at the time. Thus, the system was operating with a coefficient of performance of better than 3.5.

Total <u>material</u> cost of the solar system was about $900.00. It is estimated that with an improved condenser one could use this scheme to produce about 45-gallons (170.6 liters) of 150^{O}F (65.6^{O}C) water per hour in the summer at an electrical cost of less than 8¢. Produced with a conventional resistance type electric water heater, the heating of the water would cost about 27¢. With mass purchasing of materials and an efficient assembly line, it should be possible to manufacture a unit of this size for a direct cost of about $1,000.00.

If the system were to be scaled up to provide a whole house with hot water for heating in the winter time and chill water for cooling in the summertime, one would probably wish to change the system refrigerant from F-114 to F-22. F-114 was chosen for the present application because the solar panels were not designed to withstand high pressures (>80psi) (553 kPa). At the panel temperatures expected during summer heating tests ($\sim 120^{o}F$) ($48.9^{o}C$) F-22 can be expected to reach pressures of 260 psig (1797.12 kPa) whereas at the same conditions F-114 reaches pressures of 50 psig (345.6 kPa). At $40^{o}F$ ($4.44^{o}C$), (the winter heating condition) F-114 boils at atmospheric pressures while F-22 is at a comfortable 80psig (553 kPa). The advantage of F-22 under the latter circumstances is that a given <u>size</u> compressor can move about 6 times as much mass per revolution, thereby reducing the capital cost for a given level of refrigerant flow (and heat transfer).

Upon return from Albuquerque, the condensing portion of the system was replaced by a commercial water-cooled condenser with the original water tank now serving only as a thermal storage device. Water from this tank was then force-circulated through one of the original condensers which had been sealed into an air duct. Moving air was supplied by a 1 hp squirel-cage blower. These steps permitted the solar panels to operate at their full effectiveness. Figure 2 shows the system operating at the North Carolina State Fair during the week of October 20-25, 1975.

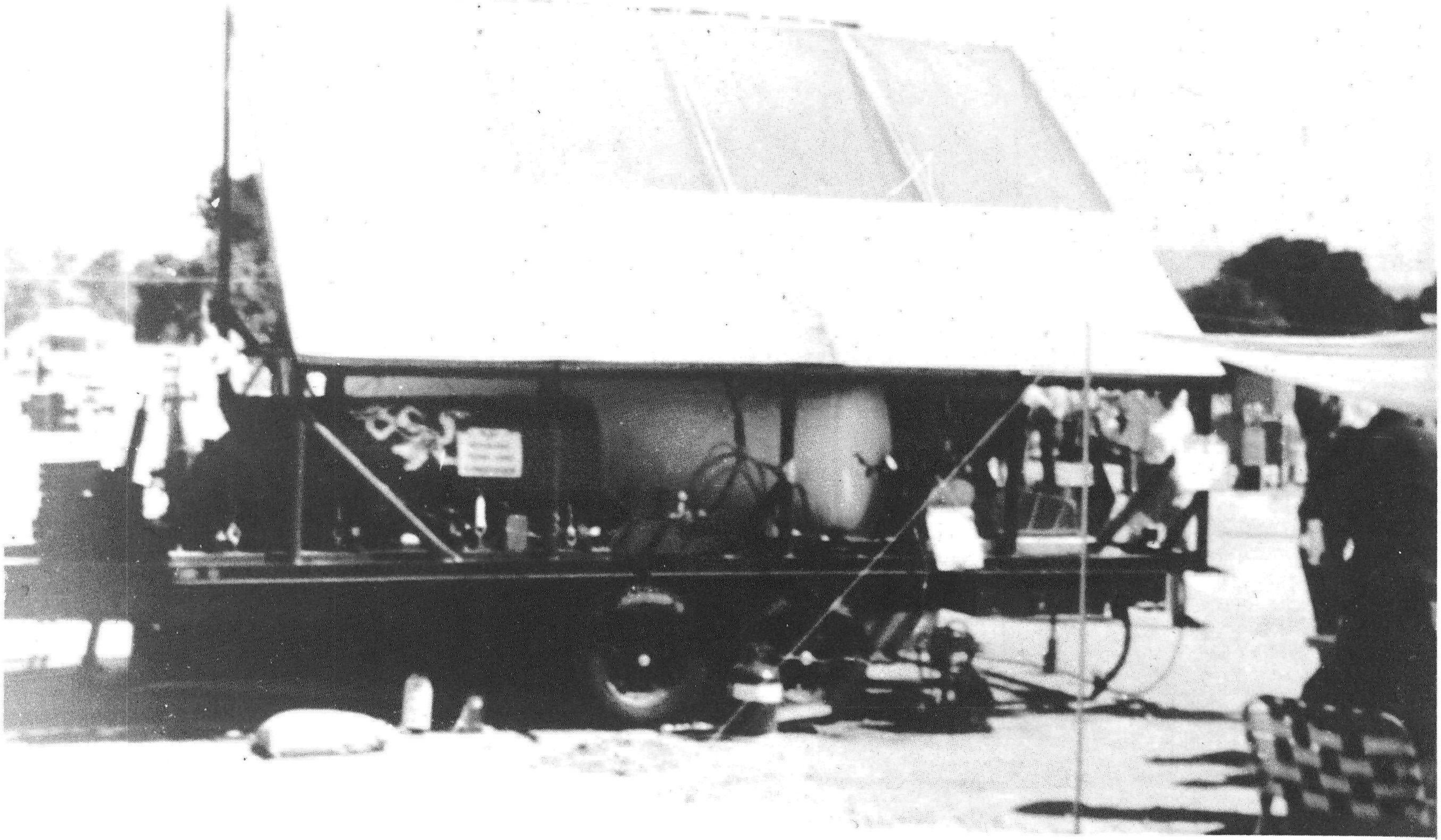

Figure 1

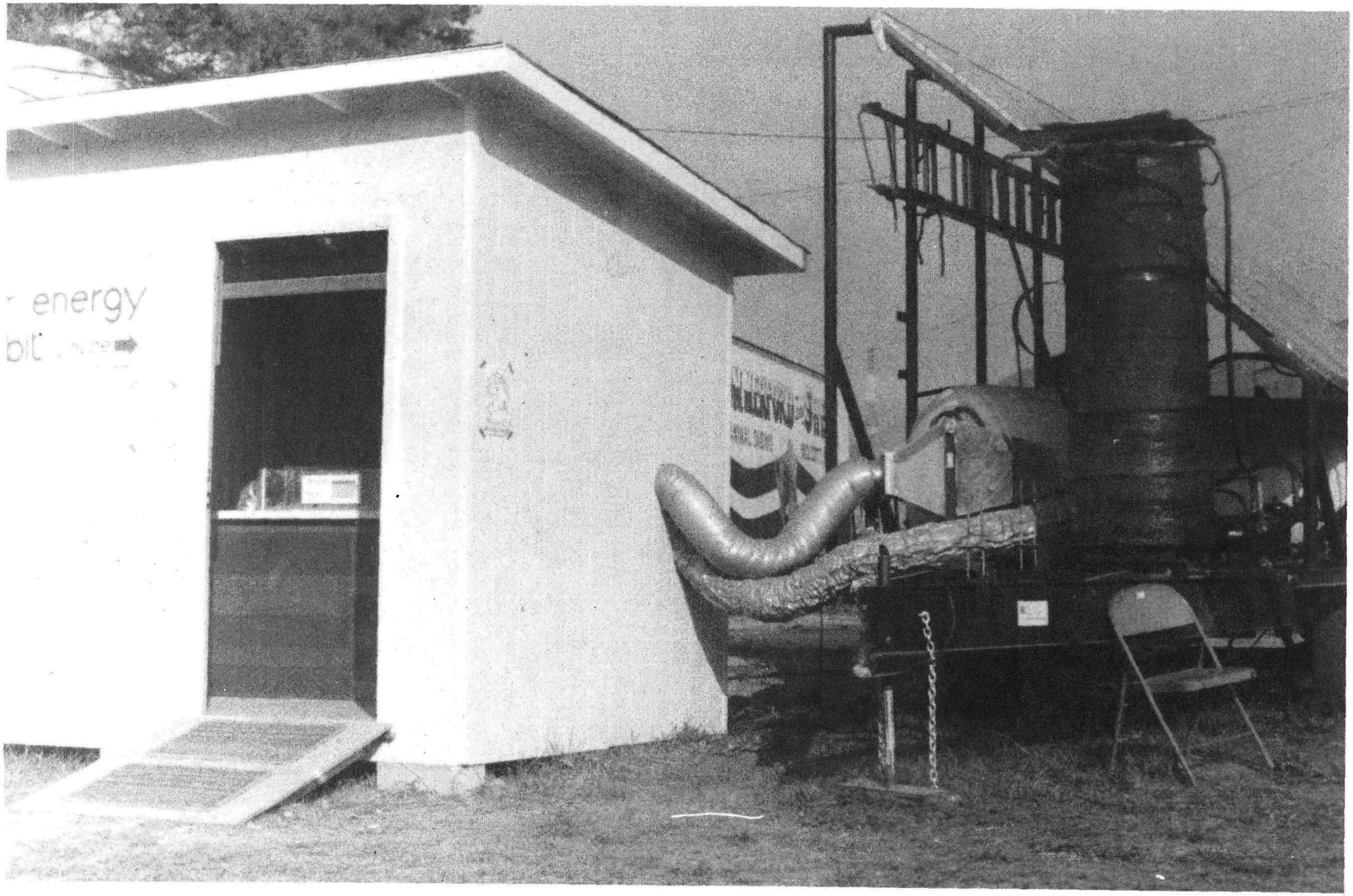

Figure 2

RECENT SYSTEM MODIFICATIONS AND TEST RESULTS

The system as shown at the N. C. State fair was modified extensively during the winter of 1975-1976. As a result of much over-the-road travel several joints had to be remade. Pressure and temperature instrumentation were added at all the points in the cycle as well as refrigerant flow rate, condenser calorimetry, and solar insolation. An oil separator and a suction line accumulator were also added.

The system was ready to begin operation by early spring. Shortly thereafter, however, a failure in one of the condenser joints, ultimately necessitating replacement of the condenser, delayed operation until the return of warm weather. The warmer temperatures and higher solar insolation rates lead to compressor overheating and ultimately compressor failure. It became obvious after examining the test data that the small compressor was unable to move enough gas to cool the panels during the higher ambient temperatures and the high levels of solar insolation then existing.

Accordingly, a five ton, F-22,hermetically-sealed compressor (the type ordinarily used in 230V single phase central air conditioners for the house) was used to replace the automotive compressor. With this compressor installed the following data were obtained in early August 1976:

Ambient Temperature	= 88 F	31.1^{o}C
Solar Insolation on Panels	= 22,130 BTU/HR	6.4859 kw
Panel Temperature	= 93^{o}F at Bottom	38.89^{o}C
	94^{o}F at Top and Center	
Heat Output in Condenser	= 28,210 BTU/HR	8.26788 kw
Water Temperature In	= 69^{o}F	20.56^{o}C
Water Temperature Out	= 91^{o}F	32.78^{o}C
Regrigerant Flow Rate	= 455 lbs/HR	206.57 kg/HR
Electrical Power into Motor	= 9543 BTU/HR	206.57 kg/HR
Work into Gas at Compressor	= 7246 BTU/HR	2.12368 kw

Heat Losses

Compressor Outlet to Condenser Inlet	=	684 BTU/HR	.2 kw
Panel Outlet to Compressor Suction	=	319 BTU/HR	.09349 kw

Compressor Conditions

Suction =	86^{o}F,	17.5 psig	121 kPa	30^{o}C
Discharge =	198^{o}F,	111 psig	767 kPa	92.2^{o}C

Overall C.O.P.	= 2.96
Efficiency, Line-to-Gas	= 77%
Solar Panel Efficiency	≃ 100%

Two comments on these results are in order.

1. The energy obtained from the solar panels verifies the basic premise of the design that very high efficiencies, approaching 100%, can be obtained by operating the panels only a few degrees above ambient temperatures.

2. The condenser is obviously undersized for this heat load. As a result, the compressor is forced to operate at high discharge pressures and temperatures. More work must be done by the compressor per unit of refrigerant mass flow in order to reach these levels and the overall system C.O.P. is therefore somewhat lower than it would be if the discharge temperature were in the range of $150^{O}F$ ($65^{O}C$). Note that a drop in gas temperature of $58^{O}C$ ($105^{O}F$) takes place in the condenser where one would normally expect only a $10^{O}C$ change in order to release the heat picked up by the gas in the panels and in the compressor.

As a consequence of these results the condenser was replaced by a unit rated at 17.6 kw (60,000 BTU/HR). With this change the following results were obtained:

Ambient Temperature	= 75^{o}F	23.9^{o}C
Solar Insolation on Panels	= 24,648 BTU/HR	7.224 kw
Panel Temperature	= 87^{o}F - 89^{o}F	$\sim$ 31^{o}C
Heat Output in Condenser	= 31,462 BTU/HR	9.22 kw
Water Temperature In	= 68^{o}F	20^{o}C
Water Temperature Out	= 93^{o}F	33.9^{o}C
Refrigerant Flow Rate	= 473 lbs/hr	214.7 kg/hr
Electrical Power Into Motor	= 8874 BTU/HR	2.6 kw
Work Into Gas at Compressor	= 7333 BTU/HR	2.149 kw
Heat Losses		
Compressor Outlet to Condenser Inlet	= 235 BTU/HR	0.0689 kw
Panel Outlet to Compressor Suction	= 284 BTU/HR	0.0832 kw

Compressor Conditions

Suction	14 psig, 82^{o}F		96.77 kPa, 27.8^{o}C
Discharge	62 psig 181^{o}F		428.5 kPa, 82.8^{o}C

Overall C.O.P.	= 3.55
Efficiency Line-to-Gas	= 82.6%

Note that the larger condenser resulted in improving the C.O.P. from 2.96 to 3.55. This is evidenced by the fact that to deliver 11.8% more heat, 7% less power is required. The pressure ratio across the compressor was also reduced from 3.876 to 2.72 and the refrigerant flow rate was increased 4%. The fact that the compressor discharge temperature is still relatively high means that an even larger condenser would be desirable although the decreased discharge pressure indicates that about 33% of the heat pickup in the compressor is now a result of the refrigerant cooling the motor windings in the hermetically-sealed compressor. Ideally, one could expect a C.O.P. as high as 5.5 if the gas reached a temperature of only 150^{0}F at 62 psig discharge.

The apparatus in its current state is shown in Figures 3 and 4. Note how the line lengths have been shortened to reduce heat losses and pressure drops.

Additional tests are planned for the fall and winter (1976-1977) to investigate the following items:

1. Panel efficiency with cover removed as a function of the temperature difference between ambient and panel.

2. Operation at design ambient temperature with F-114 and/or F-22.

Results of these tests will be reported at a later date.

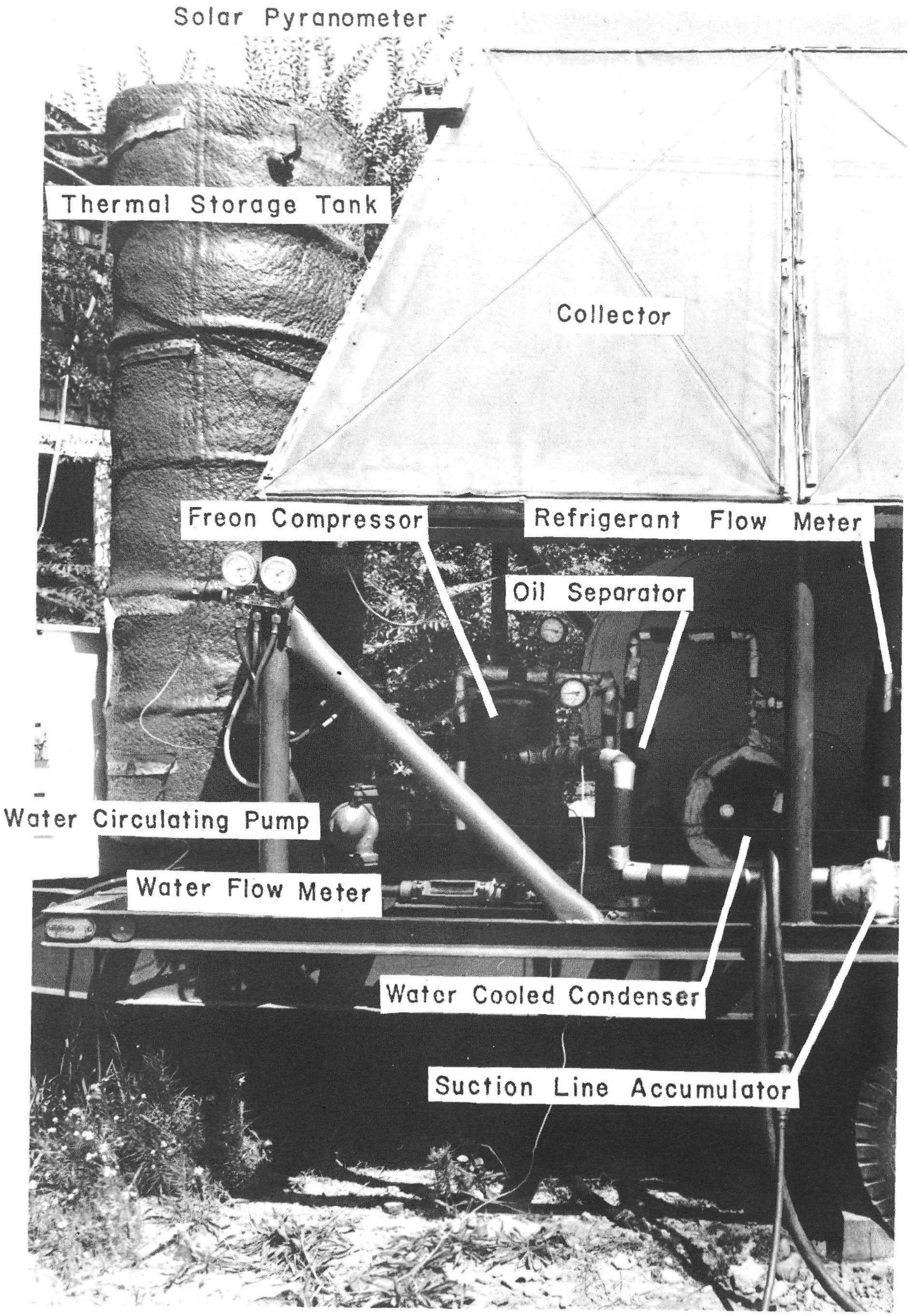
Solar Pyranometer
Thermal Storage Tank
Collector
Freon Compressor
Refrigerant Flow Meter
Oil Separator
Water Circulating Pump
Water Flow Meter
Water Cooled Condenser
Suction Line Accumulator

Figure 3

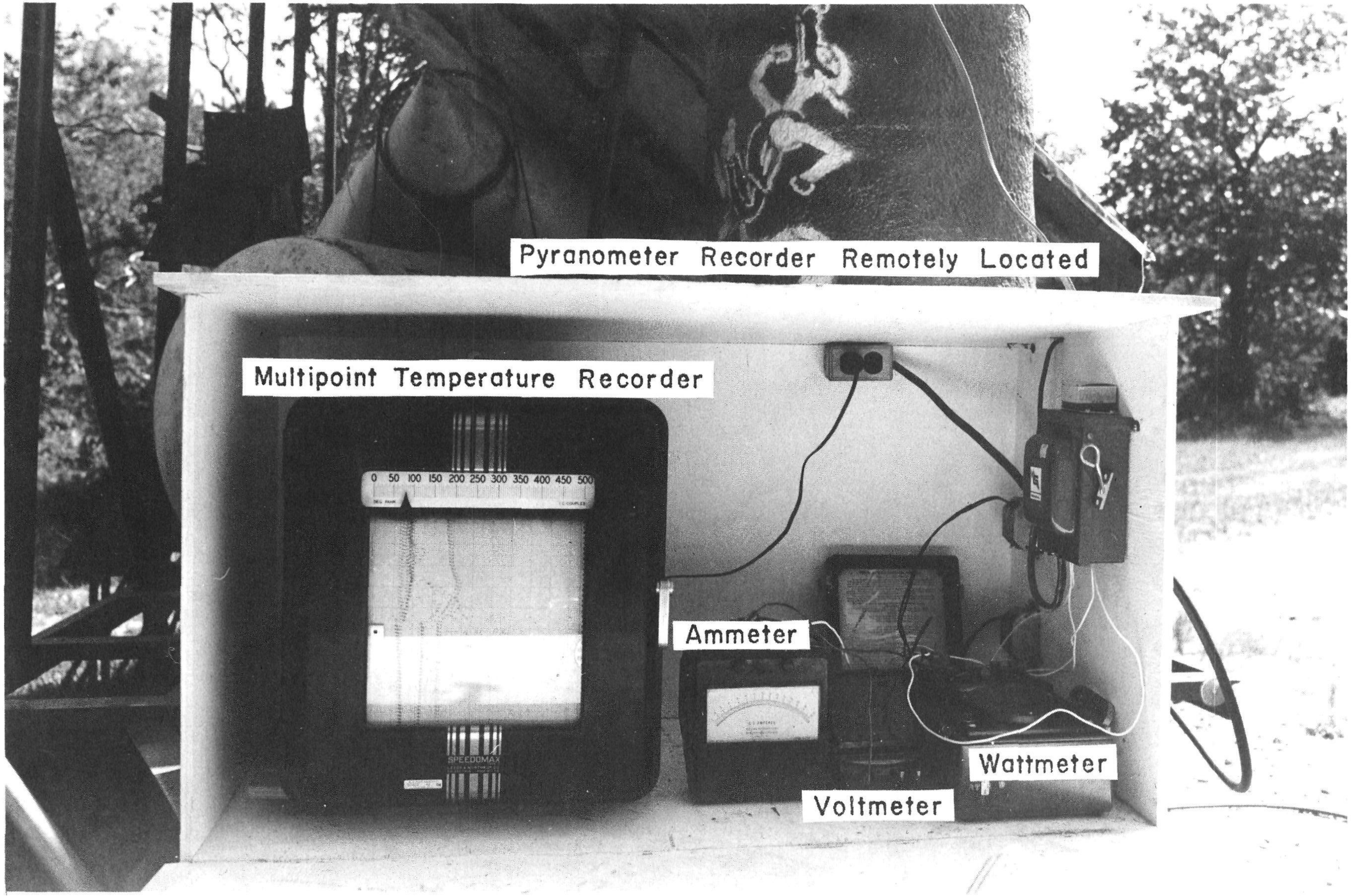
Pyranometer Recorder Remotely Located
Multipoint Temperature Recorder
0 50 100 150 200 250 300 350 400 450 500
SPEEDOMAX
Ammeter
Wattmeter
Voltmeter

Figure 4

CONCLUDING REMARKS

In scaling the system up to a house size unit one would probably wish to use 32 panels (704 ft^2, 65.4 M^2) and five 60,000 BTU/HR (17.58 kw) compressors in parallel. This would permit maximum collection of about 153,000 BTU/HR (44.84 kw) of solar energy and a total heat delivery to storage of about 200,000 BTU/HR (58.6 kw) at a cost of 48.2¢/HR based on electrical costs 3.5¢/kw-H. For comparative purposes one might mention that an oil furnace burning 40¢/gallon oil at 50% efficiency would require $1.33 worth of oil to deliver the same heat. The reason for suggesting five separate compressors is that most homes can only be supplied with single phase power and 5 HP is generally the largest motor made in single phase size. Further, such units are mass-produced while the larger units are produced at much smaller rates. Thus the cost of the five smaller machines is roughly comparable to that of one larger machine. Most important, however, is the degree of control provided by being able to operate from one to five units depending upon the panel temperature. A high degree of control is necessary because of the extreme variation in solar insolation which occurs during the course of a day. Also, the use of multiple units permits continued operation while one of the units is being repaired. At current prices it is estimated that one could construct the panels and buy the compressors for $4441.00. Interconnect plumbing, wiring, controls, heat exchanger storage tank, regrigerant, and circulating pumps are additional.

Ideally one would wish to employ a very large storage volume. so that one could store the excess heat collected during November and early December for use when the heat collected may be insufficient as it may in late December and January. Operation in this manner also tends to relieve the peak load requirements on the electrical utility.

DESIGN EXERCISE FOR APPLICATION
OF
SOLAR-ASSISTED HEAT PUMP TO A HOME

In an effort to determine how a system employing this concept might perform using commercial hardware, design calculations were carried out for a home in the Raleigh, NC area. The assumptions are listed below. Note however that this design is but one of many possible arrangements employing different collector sizes and orientation, different compressor sizes, and different storage volumes, all matching the same long term heating and cooling requirements. Because of the very large number of variables involved, it has not been possible to arrive at a cost-effective optimum. The following is therefore presented in the interests of stimulating further investigation and not as a finished design in any sense. To facilitate its use and understanding by engineers and others in the heating and air conditioning industry, the calculations are shown in U. S. Customary Units.

Assumptions:

House in Raleigh, NC with 3000 ft^2 floor space

Usual winter daytime minimum: $20^{o}F$; nighttime: $10^{o}F$

Usual summer daytime high: $95^{o}F$; nighttime: $75^{o}F$

Max. heat load for A/C, summer: 90,000 BTU/hr

Winter heating required for $20^{o}F$ night; 90,000 BTU/hr

Total heat load (24 hrs) max summer

or winter: 1,600,000 BTU

Hours of daylight, summer: 16

Hours of daylight, winter: 8

Solar Collector-radiator area 3000 ft^2, 0^{o} incidence with ground

Heat collected over 8 hrs. during winter = 1,600,000 BTU

80,000 lbs of water in tank, buried and insulated

Water from tank is circulated in coil in heating - cooling duct

by circulating pump

Interior heat exchanger can pickup or deliver 100,000 BTU/hr

with a 15^{o} ΔT

For cooling, coil is supplied with water from $35^{o}F$ to $50^{o}F$

For heating, coil is supplied with water from $95^{o}F$ to $110^{o}F$

Insulation on collector is sufficient to keep collector surface $25^{o}F$

above ambient while withdrawing 200,000 BTU/hr.

Cooling

Heat pickup during daylight hours: 1,200,000 BTU
water originally at 35^{o}F

$$\text{Temperature rise in water} = \frac{1{,}200{,}000}{80{,}000} = 15^{o}F$$

Final water temperature = 50^{o}F

Heat Pump Operating Conditions (on R-22)

Heat Rejected	205,000 BTU/HR	203,000	199,500	194,000
Condensing Temp	130^{o}F	120^{o}F	110^{o}F	100^{o}F
Evaporating Temp	50^{o}F	45^{o}F	40^{o}F	35^{o}F
Power Consumed	19,000 watts	17,875	15,700	14,425
C.O.P.	3.01	3.34	3.51	3.95

During eight hour operating cycle heat rejected is 1,600,000 BTU.

Power required between 9:00 PM and 5:00 AM averages 17.0 KW + 1.835 KW
between 5:00 AM and 9:00 PM averages 1.835 KW

Consumption is 176.2 KWH per day with peak occuring 9-11 PM. Consumption for conventional Air conditioning with equal efficiency is about 15% greater (because of higher heat rejection temperatures) with the peak load coming 3-5 PM.

Thus this scheme has two advantages:

(1) Shifts the air-conditioning peak load to a time when many industrial and commercial users have closed down for the day, thus reducing the power company's need for generating capacity.

(2) The total power required is reduced about 15%.

For cooling, the covers of the solar panels must be removed to enhance convective as well as radiative heat transfer. Calculations indicate that in this mode the panels are approximately twice as effective in rejecting heat as they are in collecting heat in the winter (with glass or tedlar covers).

Heating

Heat loss during hours of darkness: 1,200,000 BTU
water originally at 110°F

Final water temperature = 95°F

Heat Pump Operating Conditions

Heat Into Water	298,600*	291,200	284,100	277,000 BTU/HR
Condensing temp.	95°F	100°F	105°F	110°F
Evaporating temp.	45°F	45°F	45°F	45°F
Power Consumed	14,300	15,050	15,750	16,475 WATTS
C.O.P.	6.1199	5.6708	5.287	4.9277

*If collector cannot supply this much heat, evaporating temp will drop to match heat available. Example: for 200,000 BTU/HR pickup from collector, evaporation temperature will be 33°F. Power consumed will also drop to about 13,700 watts and heat into water will be 246,500 BTU/HR. C.O.P. = 5.2734

Compressor Required: 4 CYLINDER

Bore: 2 1/2"

Stroke: 2"

2380 CFH @ 1750 RPM (R-22)

Power required between 8:30 AM and 4:30 PM averages 15.5 KW + 1.835 KW
between 4:30 PM and 8:30 AM averages 1.835 KW

Consumption per day with solar-assisted heat pump is 167.85 KWH. Consumption with electric resistance heat to develop the same heat would be 670 KWH, a factor of 4.0 times as much.

Comparison with Natural Gas for Heating

Home gas furnaces are assumed to be 85% efficient. Conversion of natural gas to electricity and delivery to home is assumed to be done at an efficiency of 33%.

To heat the home, assume a requirement of 2.18×10^6 BTU per day. Home gas furnace requires 2.565×10^6 BTU of gas input per day. Solar-Assisted Heat Pump requires input of 570,000 BTU per day or

1.712×10^6 BTU of gas input at the power plant. Even considering the inefficiency of converting gas energy to electricity, the solar assisted heat pump decreases the demand for gas for home heating by 33%.

Compared with burning gas at home or buying electricity for the solar-assisted Heat Pump, electricity now costs about 3 times as much per input BTU. But gas heating requires about 4.7 times as much input so that gas will be about 1.5 times as expensive.

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