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Solar Heat Pump Systems with Refrigerant-Filled Collectors

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ABSTRACT

The heat pump system with a refrigerant-filled evaporator consists of a standard air-to-air or air-to-liquid heat pump that utilizes a solar panel as the evaporator. A combination of solar energy and convection heat transfer acts as the "free" energy absorbed by the collector/evaporator. In this paper, the seasonal performance of such systems for industrial applications will be presented. Performance of collector/evaporator heat pumps will be compared with alternative heat pump and solar systems. The benefits of covered and coverless collector/evaporators will be discussed. Results to date have shown that refrigerant-filled collector heat pumps do not perform as well as conventional heat pumps at small collector areas but have as much as 15% performance improvement over conventional heat pumps at an appropriate collector area.

INTRODUCTION

One of the recent developments in the area of heat pump and solar energy technology is the use of a refrigerant-filled collector to replace the evaporator in a heat pump system. The rest of the system employs standard materials and components currently available in the refrigerating and air conditioning industry. The advantage of the collector-evaporator from the heat pump standpoint is that the evaporator operates at a temperature higher than ambient due to solar heating, which increases the heat pump COP. From a solar viewpoint, the working fluid for the collector is a refrigerant that undergoes a phase change at a relatively low temperature, which yields a higher collector efficiency than that of a straight solar system. Thus, there are reasons for expecting high performance from these systems.

Although several such systems currently are being marketed, no information is available to compare the performance of these systems to conventional heat pump or solar systems. Studies of refrigerant-filled collectors have been done.[1,2] Other studies of solar-assisted heat pumps have involved detailed simulations of series and parallel configurations, which include an additional collector loop.[3] In this paper, the performance of heat pump systems with refrigerant-filled collectors will be explored.

SYSTEM CONFIGURATION

The basic system is shown in Fig. 1. The evaporator is a refrigerant heat exchanger that acts as a solar collector during periods of sunshine. The ambient air acts as an additional convective heat source or sink, depending on whether the refrigerant temperature is higher or lower than ambient. The thermal expansion valve regulates refrigerant flow to the collectors in order to maintain constant superheat at the heat pump compressor inlet.

The energy rejected by the condenser contributes to load requirements through a refrigerant to-air or refrigerant-to-water heat exchanger. Loads may range from residential space heating to industrial process heat supply.

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MODEL DEVELOPMENT

The flat-plate evaporator model is based on the conventional flat-plate equation[4]

$$Q_u = A_C F_R[(\tau\alpha)I_T - U_L(T_{f,i} - T_a)] \tag{1}$$

where

Q_u is the useful energy collected, A_c is the collector area, F_R is the collector heat-removal factor, $(\tau\alpha)$ is the transmittance-absorptance product, I_T is the incident solar radiation, U_L is the loss coefficient, and $T_{f,i}$ and T_a are the fluid inlet and ambient air temperatures, respectively.

For a refrigerant-filled evaporator, the collector heat-removal factor, F_R, is unity. The fluid inlet temperature is the evaporator temperature, T_{evap}, and the useful energy collected is the absorbed energy for the heat pump, Q_{abs}. Eq 1 can be rewritten as

$$Q_{ABS} = A_C[(\tau\alpha)I_T - U_L(T_{evap} - T_{amb})] \tag{2}$$

The evaporator equation can be written to utilize the collector "sol-air" temperature, defined as

$$T_{sa} = (T_a + \frac{(\tau\alpha)I_T}{U_L}) \tag{3}$$

The loss coefficient, U_L, is the sum of the collector top and bottom loss coefficients, U_T and U_B. The bottom of the collector is not exposed to solar radiation.

This allows Eq 2 to be written as

$$Q_{ABS} = A_L U_L(T_{sa} - T_{evap}) \tag{4}$$

For an uncovered collector, the loss coefficient is a strong function of wind velocity. The convection coefficient, due to wind, is determined by[4]:

$$h_{wind} = 3.74\ V_{wind}^{0.6}\ W/m^2C \tag{5}$$

The top and bottom loss coefficients may also include cover plates and conventional insulation. These are treated in the same manner as for a conventional collector [see Refs 4 and 5]. Other equations used in determination of heat pump capacity and COP are curve fits to conventional manufacturers data. Relations of the form

$$\begin{aligned} Q_{DEL} &= f(T_{evap}) = b_1 + b_2 T_{evap} + b_3 T_{evap}^2 \\ COP &= \frac{Q_{DEL}}{W} = a_1 + a_2 T_{evap} \end{aligned} \tag{6}$$

were used, where a_1, a_2, b_1, b_2 and b_3 are constants for a particular unit at a particular delivery temperature. Fig. 2 shows the corresponding performance curves used for the simulations reported here. The COP is limited to a maximum value of 4 to prevent unrealistic system performance.

The energy delivered to the load is given by the sum of the absorbed energy and work

$$Q_{DEL} = Q_{ABS} + W \tag{7}$$

Any shortfall in meeting the load is made up with an auxiliary heat source with COP of 1. Thus

$$Q_{LOAD} = Q_{DEL} + Q_{AUX} \tag{8}$$

The performance totals for any time period may be reduced to a measure that is the ratio of free energy obtained divided by the load. The "nonpurchased fraction" is:

$$F_{np} = 1 - \frac{(W + Q_{AUX})}{Q_{LOAD}} \tag{9}$$

The sol-air temperature concept allows hourly weather data to be converted into two-dimensional bins, which store the number of hours corresponding to an ambient temperature range and a sol-air temperature range. Performance can be obtained by adding all of the contributions from each bin to the corresponding total for the period of interest. Fig. 3 shows typical bin data for a particular collector in Madison, WI during the daytime conditions. The left-most bin indicates times at which the sol-air and air temperatures are equal. The remaining bins show the heating effect of solar radiation.

An example of the effect of sol-air temperature on heat pump COP is illustrated in Fig. 4, which shows that the heat pump performance increases as both sol-air temperature and collector area increase.

RESULTS

The seasonal performance of a refrigerant-filled, collector heat pump system has been evaluated using computer simulations.

The load is taken as representative of an industrial process load and is 24 MJ/hour (23,000 Btu/hr) for 12 hours during daytime, each day of the year, in both locations. The results are on a seasonal basis.

Figs. 5 and 6 show the total energy delivered to the industrial load by the refrigerant-filled collector systems as a function of collector area. The two locations are Madison, WI, and Albuquerque, NM. The top curve is for a coverless collector, and the bottom is for a covered collector. The difference between values on these curves and the total annual load of 105 MJ (100,000 Btu) indicates the amount of auxiliary energy necessary to meet the load.

It is apparent that for a given area, the coverless collector supplies more energy to the load. The curves show the decreasing advantage of collector area as the energy delivered approaches the value of the load. For the situation in Madison, the collector area needed to meet 90% of the load is 13 m^2 (140 ft^2) for the coverless collector and 40 m^2 (430 ft^2) for the covered collector.

In Figs. 6 and 8, comparisons are made between the four types of heating systems described in Tab. 1. The analysis is made again with the industrial load, and the yearly nonpurchased fraction is plotted against collector area for all systems, with the exception of the conventional heat pump system. The nonpurchased fraction for a straight heat pump system is a horizontal line since the heat pump size is constant.

The nonpurchased energy fractions for refrigerant collectors, with and without covers, increase rapidly at low collector area and are better than that for the conventional heat pump at large areas. The performance of the conventional solar system is poorest at small collector areas but is better than the other systems at larger areas.

The performance of the uncovered system is consistently better than the covered system in both locations. The difference is greatest for collector areas of between 10 m^2 (108 ft^2) and 40 m^2 (430 ft^2), which is the range of areas for currently installed applications of this capacity. The improved performance of the uncovered collector over the covered collector system was not anticipated. The covered system was expected to have the advantage of being able to operate at higher temperatures with a higher system COP during periods of solar radiation. The covered system does operate better during those periods of the day when solar insolation is higher; however, with low levels of radiation, the primary energy source is ambient air. The lower loss coefficient of the covered collector will force the evaporator temperature to be lower than that for the uncovered collector in order to absorb energy, with a consequent reduction in COP. There are significant periods with low solar radiation during the year, and as a result the annual performance of the covered collector system is lower.

The performance of the conventional solar collector heating system differs significantly between the two locations. This reflects the large difference in radiation levels between Madison and Albuquerque. However, the higher incident radiation levels in Albuquerque compared to Madison are not reflected in the performance curves of collector heat pump systems. This implies that radiation is not the primary heat source for these systems. Fig. 9 shows the effects of removing the solar radiation term from the contribution to the performance of the uncovered collector heat pump system in the Madison simulation. The substantial contribution by convective heat transfer between the ambient air and the collector is evident. This shows that solar radiation accounts for only about 10 percent of the nonpurchased energy delivered to the load.

CONCLUSIONS

The performance of refrigerant-filled collector heat pumps is poorer than conventional air source heat pumps at small collector areas but is better at higher collector areas. For an appropriate collector area range the collector heat pump system can have a 10% to 15 percent performance improvement over conventional heat pumps.

At large collector areas, a conventional solar heating system with storage will contribute a larger share of the load than the collector and conventional heat pump systems.

The uncovered collector heat pump system outperforms the covered collector heat pump system at all collector areas for daytime only loads.

REFERENCES

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3. T.L. Freeman, J.W. Mitchell, and T.E. Audit, "Performance of Combined Solar Heat Pump Systems, Solar Energy 22: 125 (Nov. 1979).

4. J.A. Duffie and W.A. Beckman, Solar Energy Thermal Processes (New York: Wiley, 1980).

5. TRNSYS, A Transient Simulation Program, Engineering Experiment Station Report #38, Solar Energy Laboratory, Univ. of Wisconsin-Madison.

ACKNOWLEDGEMENT

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TABLE 1

System Simulation Parameters

System Type	F_R-U_L $\left(\frac{W}{m^2C}\right)$	$\left(\frac{Btu}{hr\text{-}ft^2F}\right)$	$F_R(\tau\alpha)$	Collector Slope
Covered Collector	3.0	0.5	.72	60
Uncovered Collector	20.0	3.5	.80	60
Conventional Heat Pump	-	-	-	-
Conventional Solar	4.22	0.74	.70	60

Conventional Solar Storage: 350 KJ/°C-m^2 (17.1 Btu/F-ft^2)
Heat Pump Size: 25.3 MJ/hr (24,000 Btu/hr)

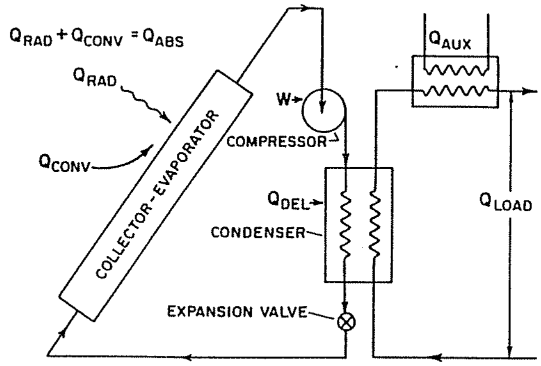

Figure 1. *Schematic of solar heat pump system with refrigerant-filled evaporator*

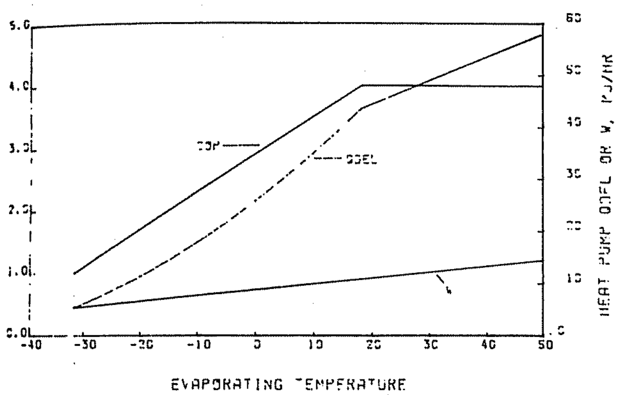

Figure 2. *Heat pump performance as a function of evaporator temperature*

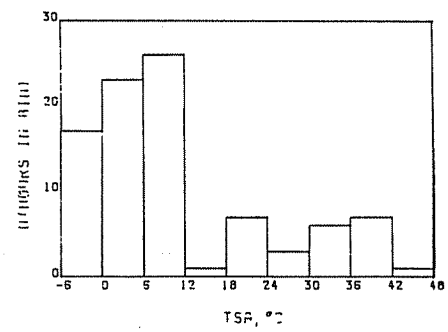

Figure 3. *Sol-air temperature distributions for March in Madison, WI*

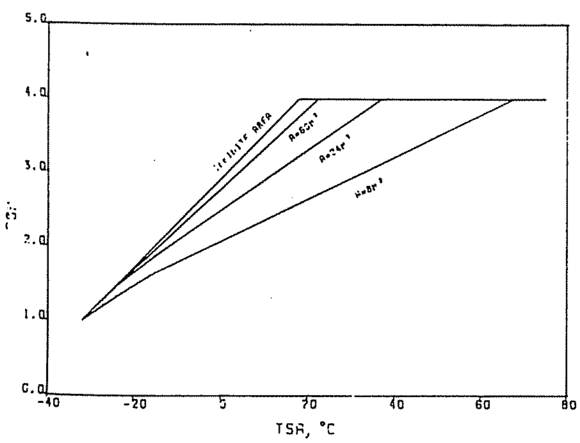

Figure 4. *Heat pump COP as a function of sol-air temperature*

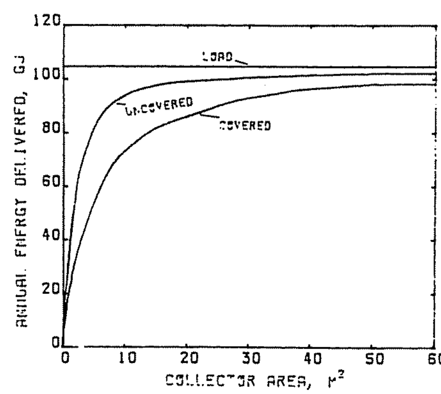

Figure 5. *Energy delivered by heat pump system as a function of collector area for Madison, WI*

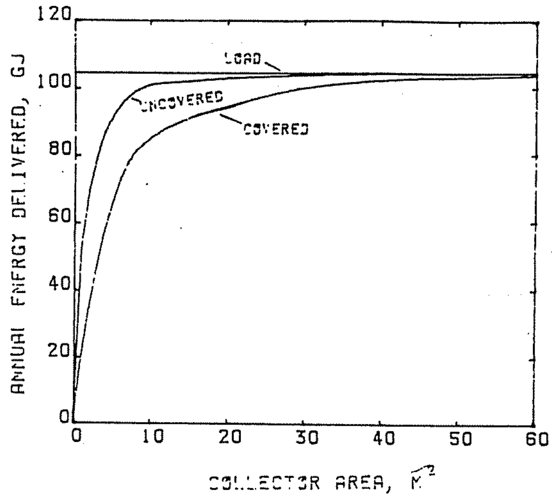

Figure 6. Energy delivered by heat pump system as a function of collector area for Albuquerque, NM

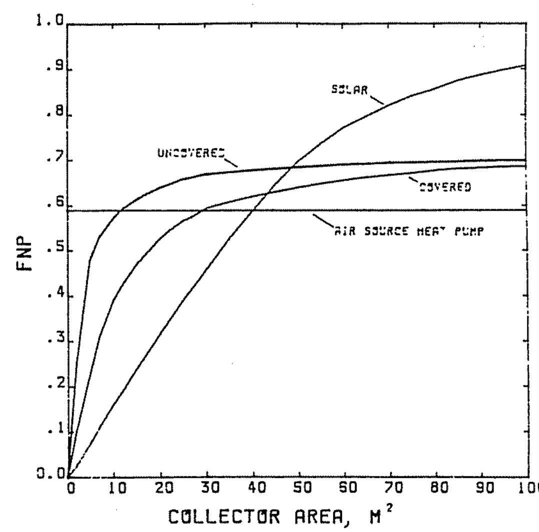

Figure 7. Fraction of nonpurchased energy supplied as a function of collec area, Madison, WI

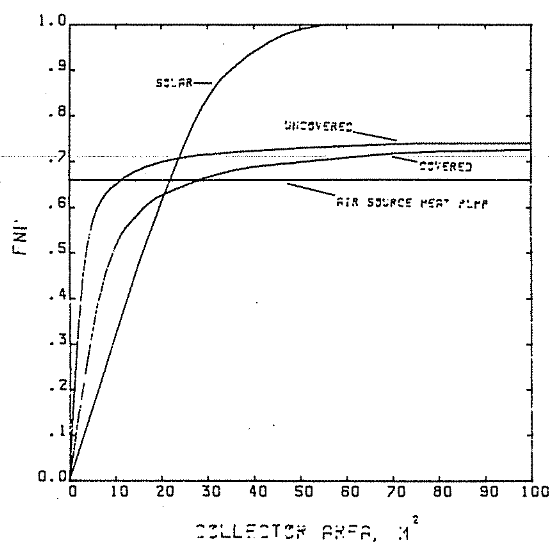

Figure 8. Nonpurchased energy fraction as a function of collector area, Albuquerque, NM

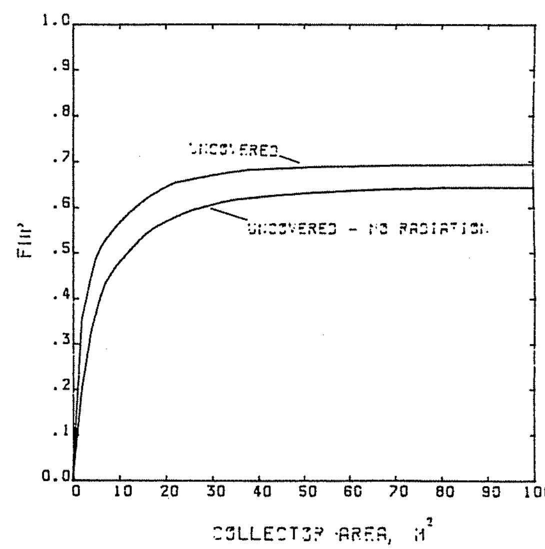

Figure 9. Nonpurchased energy fraction as a function of collector a. Madison, WI

DISCUSSION

W.D. Cooper, DuPont Co., Wilmington, DE: A solar air source heat pump has operated at a laboratory to heat hot water:

R22 Refrigerant
Bare Panel - 57 f^2
Comp. Display - 1.361 c.i./rev
Operating conditions
Disch. Pressure 300 psig

Outdoor Temp. °F	I	Ps psig	Capacity BtuH	COP
43	256	70	14500	3.37
50	15	61	9900	2.49
79	220	90	18000	3.69

J.W. Mitchell: These results show that the heat pump COP does increase with solar insolation. The improvement in COP between the 43°F and 50°F temperatures of about 35 percent is in agreement with out model results. Again, it is importnat to note that there are results for daytime performance, and that the improvement over the entire heating season will be less.