Design Method and Performance of Heat Pumps With Refrigerant-Filled Solar Collectors

M. P. O'Dell

J. W. Mitchell

W. A. Beckman

Solar Energy Laboratory,
University of Wisconsin-Madison,
Madison, Wis. 53706

A general procedure is presented for estimating the seasonal performance of solar-assisted heat pumps with refrigerant-filled collectors. The procedure accounts for variations in collector design and orientation and also for heat pump capacity and efficiency. The results from this design procedure for space heating applications are compared against those for both conventional heat pumps and liquid solar systems. The effects of collector area and design, heat pump size and degradation due to cycling, and energy storage are discussed. For space heating, uncovered collector heat pump systems have better performance than both conventional air-source heat pumps and covered collector heat pump systems over a wide range of collector areas. The cooling performance of a collector heat pump system is inferior to that of conventional heat pumps.

Introduction

One of the recent developments in the area of heat pump and solar technology is the use of refrigerant-filled solar collectors to replace the standard air source evaporator in a heat pump system. The rest of the system employs standard materials and components currently available in the refrigeration and air conditioning industry. An advantage from the heat pump standpoint is that the collector/evaporator can operate 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 halocarbon refrigerant which undergoes a phase change at a relatively low temperature. This yields a higher collector efficiency than that of a straight solar system. Thus there are reasons for expecting higher performance for these systems than conventional air-to-air heat pumps or conventional solar systems.

Although there are several such systems currently being marketed, there is no information available to compare the seasonal performance of these systems to conventional heat pumps or solar systems. Studies of refrigerant-filled collectors alone [1] and the resulting heat pump performance with elevated evaporating temperatures [2] have been done but were not used to estimate system performance. Design point analyses have been made [3], but the results do not indicate seasonal performance. Other studies of solar-assisted heat pumps have involved detailed simulations of series and parallel configurations which include an additional collector loop [4] but do not consider simultaneous solar and convection inputs. Several systems have been built and operated successfully in Australia, but their performance relative to conventional heat pumps has not been established [5, 6].

Contributed by the Solar Energy Division for publication in the JOURNAL OF SOLAR ENERGY ENGINEERING. Manuscript received by the Solar Energy Division February 18, 1983.

In this paper, the performance of heat pump systems with refrigerant-filled collectors in space heating applications will be explored. Performance comparisons will be made against systems of similar nature, including conventional air source heat pumps and active liquid solar collection systems. The effects of system parameters such as convection heat transfer coefficient, maximum COP, heat pump size, collector area, location, and collector slope will be included. Methods of optimizing system performance by storage or combined mode applications will be presented. Degradation due to heat pump cycling and system operation in the cooling mode is discussed.

System Configuration

The basic system is shown in Fig. 1. The evaporator is a refrigerant heat exchanger that absorbs available solar radiation and may have cover plates (covered collector), or it may be bare (uncovered collector). The ambient air acts an an additional 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. A refrigerant receiver and/or accumulator may also be included in the system to help in controlling the refrigerant distribution.

The energy rejected by the condenser contributes to load requirements through a refrigerant-to-air or refrigerant-to-water heat exchanger. Space heating requirements are typically met on a demand basis by heat pump systems. An energy storage media is not included in this configuration, although the effects of storage will be discussed.

System Performance Model

The flat-plate evaporator is modeled with the collector equation for useful energy gain [7].

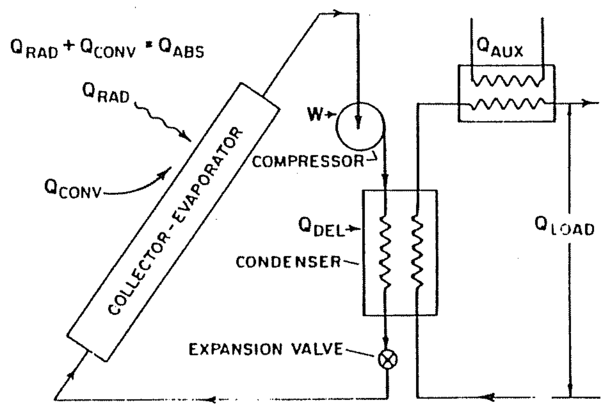

Fig. 1

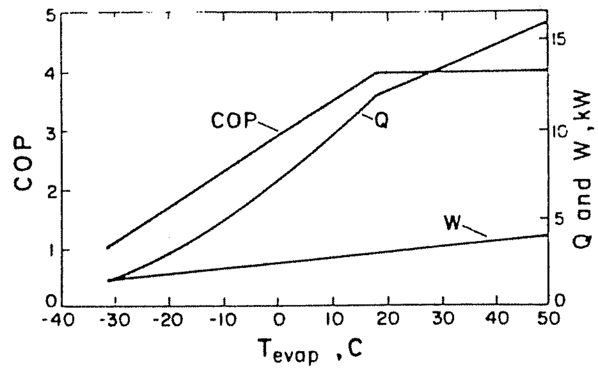

Fig. 2

$$Q_u = A_c F_R[(\tau\alpha)I_t - U_L(T_{f,i} - T_a)] \quad (1)$$

For a refrigerant-filled collector the collector heat removal factor, F_R, is assumed to be unity. This is a reasonable assumption since the convection heat exchange coefficient inside the collector/evaporator is very high due to boiling of the refrigerant, which contributes to a very high collector plate efficiency. In addition, the refrigerant is saturated and at a constant temperature, so the temperature gradient in the flow direction is insignificant. With F_R equal to unity, the analysis will yield results which are slightly greater than what is achievable.

The fluid inlet temperature to the collector is the evaporator temperature T_e, and the useful energy collected is the absorbed energy for the heat pump Q_{abs}. Equation (1) can be rewritten as

$$Q_{abs} = A_c[(\tau\alpha)I_T - U_L(T_e - T_a)] \quad (2)$$

The loss coefficient U_L is the sum of the collector top and bottom loss coefficients and is treated in the same manner as conventional solar collectors [7, 8]. The bottom of the covered collector is insulated while both sides of the uncovered collector are uninsulated. U_L for the uncovered collector is due primarily to convection and is a strong function of wind velocity. The wind convection coefficient is determined by [7]

$$h_{wind} = 3.74 V_{wind}^{0.6} (\mathrm{W/m^2C}) \quad (3)$$

For the uncovered collector with both sides exposed

$$U_L = U_T + U_B = 2h_{wind} \quad (4)$$

The heat pump performance models include relations for heating COP and capacity (Q_{del}). The COP is dimensionless and is defined by

$$\mathrm{COP} = \frac{Q_{del}}{W'} \quad (5)$$

where W' is the electrical input to the heat pump. The energy delivered to the load, Q_{del}, is the sum of energy absorbed by the evaporator and the heat pump electrical input

$$Q_{del} = Q_{abs} + W' \quad (6)$$

In residential heat pump applications, the indoor supply temperature to the condenser is relatively constant and does not affect pump performance nearly as much as the outdoor ambient conditions. Therefore, it is assumed that the heat pump COP and capacity are functions only of the saturated evaporating temperature T_e. The difference between conventional and collector/evaporator heat pumps lies in the characteristics of the outdoor heat exchanger. Manufacturer's catalog data for conventional air-to-air heat pumps as commonly used in residential applications were used as a base. However, these performance data are given as a function of outdoor ambient temperature, so for collector heat pumps it is necessary to estimate the saturated evaporating temperature that corresponds to performance data available in the catalog tables.

For a conventional refrigerant-to-air heat exchanger, the heat exchanger effectiveness is assumed to be constant over the range of operating conditions. The effectiveness ϵ defined as the actual energy transferred to the evaporator divided by the theoretical maximum energy transfer, is given by

$$\epsilon = \frac{Q_{abs}}{(\dot{m}C_p)_{air}(T_a - T_e)} \quad (7)$$

If the effectiveness at any operating condition is set equal to the effectiveness at some base condition for which the difference between the ambient and evaporator temperatures is known, the following relation can be derived

$$\frac{T_e - T_a}{(T_e - T_a)_o} = \frac{Q_{del}}{Q_{del,o}}\left[\frac{1 - \dfrac{1}{\mathrm{COP}}}{1 - \dfrac{1}{\mathrm{COP},o}}\right] \quad (8)$$

Nomenclature

A_c = collector area
C_D = degradation coefficient
COP = coefficient of performance
F_{np} = nonpurchased fraction
F_R = collector heat removal factor
h_w = wind convection coefficient
I_T = incident solar radiation
$(\dot{m}c_p)_{air}$ = air mass flow rate times heat capacity
Q = hourly energy transfer
T = temperature
U_B = bottom loss coefficient
U_L = collector loss coefficient
U_T = top loss coefficient
$(UA)_h$ = building conductance-area product
W' = heat pump electrical input
ϵ = heat exchanger effectiveness
$(\tau\alpha)$ = collector transmittance-absorptance product

Subscripts

a = ambient
abs = absorbed
aux = auxiliary
cyc = cyclic
del = delivered
e = evaporating
f,i = collector fluid inlet
L = heating load
nom = nominal heat pump capacity
o = base condition
p = plate
r = room
sa = sol-air
ss = steady state
u = useful by the collector

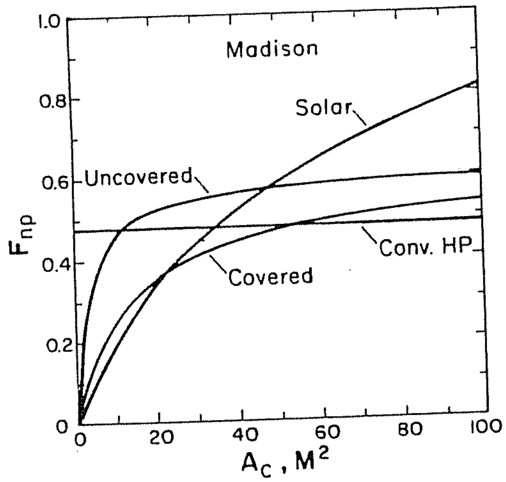

Fig. 3

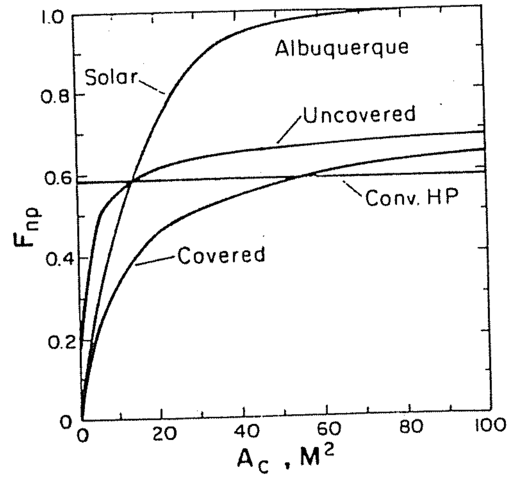

Fig. 4

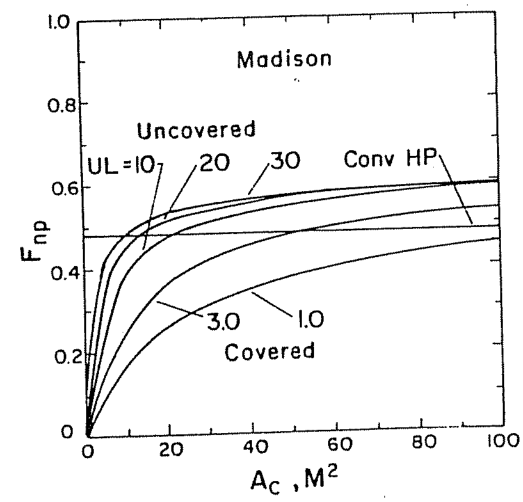

Fig. 5

Catalog data for COP and capacity for any operating conditions can then be used to determine the corresponding evaporator temperatures. The performance curves for COP, Q_{del}, and W as functions of evaporator temperature used to obtain heating performance results are shown in Fig. 2. The COP is limited to a maximum value of 4.0 to prevent unrealistic system performance.

In space heating applications, the load is determined by the usual steady-state expression

$$Q_L = (UA)_h (T_r - T_a) = Q_{del} + Q_{aux} \tag{9}$$

The performance totals for any period of interest may be reduced to the ratio of the "free energy" obtained by the heat pump evaporator divided by the load. This "nonpurchased friction" is

$$F_{np} = 1 - \frac{\int (W + Q_{aux}) dt}{\int Q_L dt} = \frac{\int (Q_{del} - W) dt}{\int Q_L dt} \tag{10}$$

The method for estimating long-term performance of collector heat pumps is based on the standard bin method for conventional heat pumps. Equation (2) can be rearranged as

$$Q_{abs} = A_c U_L \left[\left(T_a + \frac{(\tau\alpha) I_T}{U_L} \right) - T_e \right] = A_c U_L (T_{sa} - T_e) \tag{11}$$

Where collector properties and weather conditions have been combined into a "sol-air" temperature, defined as

$$T_{sa} = \left(T_a + \frac{(\tau\alpha) I_T}{U_L} \right) \tag{12}$$

Horizontal solar radiation data can be converted to incident radiation on the collector surface (I_T) as described in [7].

For fixed values of U_L and $(\tau\alpha)$, the sol-air temperature concept allows hourly weather data to be converted into two-dimensional bins that contain the number of hours corresponding to an ambient temperature range and a sol-air temperature range. The load is determined by the ambient temperature at the midpoint of the bin ambient temperature range. System performance is obtained by using the midpoint sol-air temperature of the same bin for which the load is determined. Heat pump performance data for COP and Q_{del} as functions of T_e, along with equations (6) and (11) are solved simultaneously. Performance totals are obtained by summing all of the contributions for each bin. This same bin method is also used to generate conventional heat pump performance by ignoring the sol-air temperature and using the ambient temperature bin data for both performance and load calculations. In all of the results of the study the sol-air temperature bin data were obtained on a monthly basis with a 2°C range used for the ambient temperature and sol-air temperature bins.

Base Case Space Heating Applications

The various systems were evaluted for heating performance in Madison, Wisconsin and Albuquerque, New Mexico. The parameters used in these base case performance comparisons are shown in Table 1. The weather data were obtained from SOLMET TMY tapes [9].

The performance of the conventional liquid based solar heating system with storage was determined by the f-chart [10] method. A higher loss coefficient was used for the liquid solar system than the covered collector heat pump system because of the higher average plate temperature at which the solar system operates [7].

Figures 3 and 4 show performance of the four heating systems described in Table 1 in Madison and Albuquerque. The yearly nonpurchased fraction is plotted against collector area. The nonpurchased fraction for the uncovered collector system increases rapidly at small collector areas and exceeds that of the conventional heat pump above a collector area of 11-13 m^2. The performance of the conventional solar system is the lowest at small collector areas, but is better than the other systems at large collector areas. The covered collector system performance for the space heating load is lowest at small collector areas and does not equal that of the conventional heat pump until the collector area exceeds 50 m^2. The performance of the covered collector system is consistently lower than the uncovered collector system, especially for collector areas below 40 m^2.

The greater performance of the uncovered over the covered

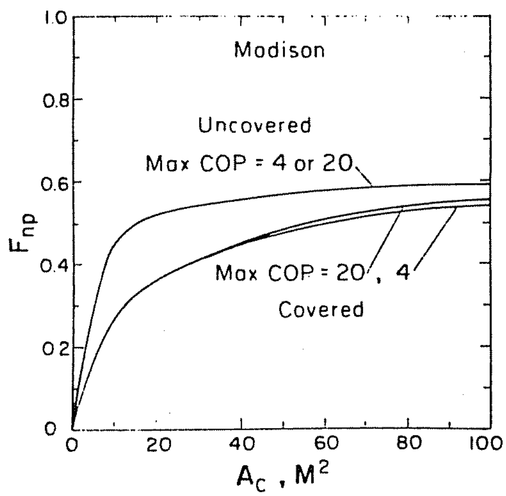

Fig. 6

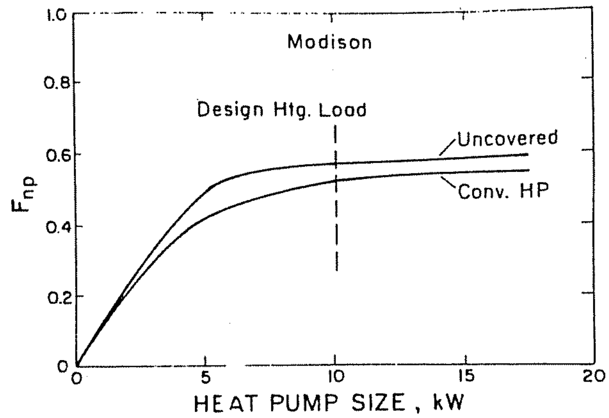

Fig. 7

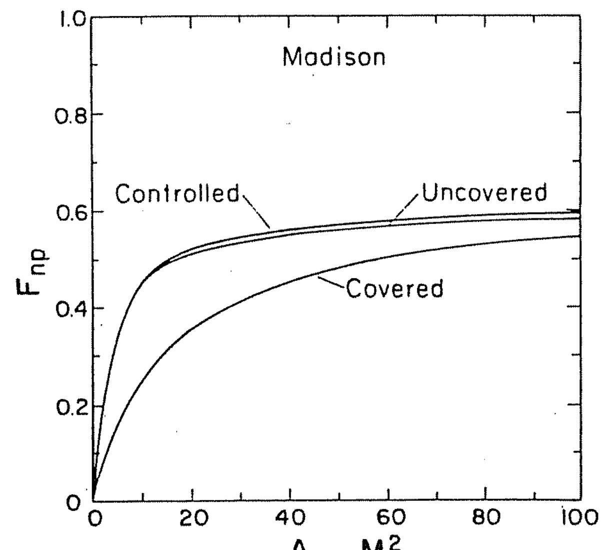

Fig. 8

systems was not anticipated. The covered system was expected to have the advantage of being able to operate at higher evaporator temperatures with a higher system COP during periods of solar radiation. The covered system does operate better during periods when solar insolation is high. However, with low levels of radiation, the primary energy source is heat transfer from ambient air. The lower heat transfer coefficient of the covered collector results in lower evaporator temperatures than that for the uncovered collector, with a consequent reduction in COP. Since for the majority of the winter there is little or no solar radiation (i.e., night and cloudy periods) and higher than average space heating loads occur at these times, it is not surprising that the covered collector heat pump system has poor performance.

The overall performance difference between the conventional solar system and the heat pump system was significantly greater in Albuquerque than in Madison. The fact that Albuquerque receives more solar radiation greatly benefitted the performance of the conventional solar system but does not contribute as much to the performance of the collector heat pump systems.

Effects of Convection Coefficients

There is uncertainty in estimating the collector loss coefficient. The results obtained in Fig. 5 for Madison were generated with the parameters of Table 1 with the exception that U_L for the uncovered collector were taken to be 10, 20, and 30 W/m^{2}C, reflecting variations of local average windspeed and structural effects. For the covered collector, differences in the "tightness" of the covers and insulation were estimated using loss coefficient values of 3.0 and 1.0 W/m^{2}C. Reducing the covered collector loss coefficient decreases the yearly space heating performance of the system considerably. This indicates that convection is the primary energy source, and that large collector loss coefficients are necessary for good performance.

Effect of Maximum COP

In order to take full advantage of high sol-air temperatures during operation, an analysis of system performance was made with a maximum COP of 20.0 instead of 4.0 and with all other parameters the same as in the base case. The performance curves of Fig. 2 were extended to allow continuous performance improvement up to a COP of 20 as shown in Fig. 6. There is little overall performance improvement for the covered systems and none for the uncovered systems. The fraction of hours for which COP's greater than 4.0 were achieved were almost none for the uncovered system and quite small for the covered system.

Effects of Heat Pump Size

The annual nonpurchased fraction as a function of nominal heat pump size for Madison is shown in Fig. 7. The results for Albuquerque are similar. The collector area for the uncovered collector system is increased in proportion to the nominal heat pump size, at a rate of 12 m^2/ton (3.4 m^2/kW). The design heating load for Madison of 10.1 kW is also shown. For both the uncovered collector and conventional heat pump systems at a nominal heat pump size equal to the design space heating load, the nonpurchased fraction is about 5 percent less than that obtained with a 5-ton (17.6-kW) unit. The heat pump size selected for a residential space heating application need not be much greater than the design building heating load. For an uncovered collector heat pump system, there is no advantage to using a smaller nominal heat pump size than a conventional heat pump. The performance curves for both types of heat pumps tend to become level at the same value of nominal heat pump size.

Collector Control Strategy

The potential for performance improvement by combining the convective heat exchange properties of the coverless collector and the insulative properties of the covered collector into one system was investigated. The system COP can be maximized by selecting at each instant of time which of the

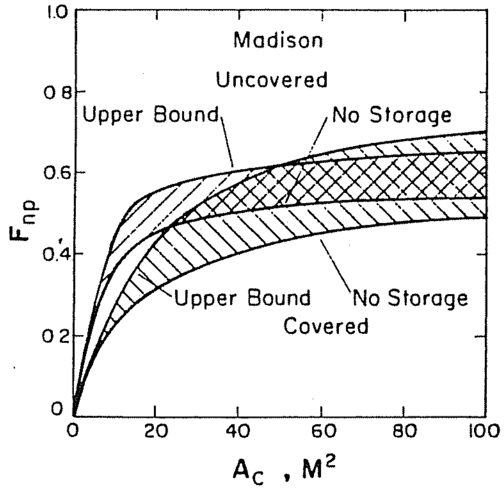

Fig. 9

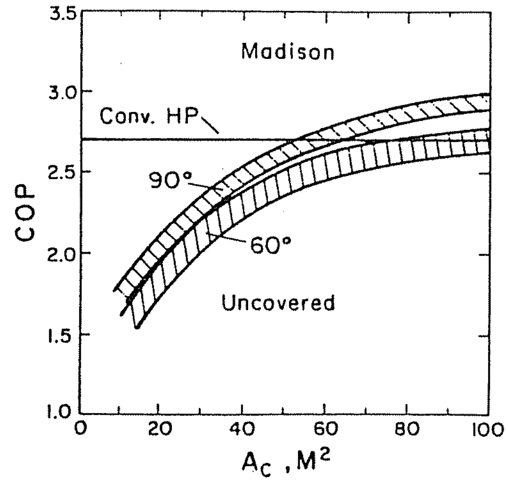

Fig. 10

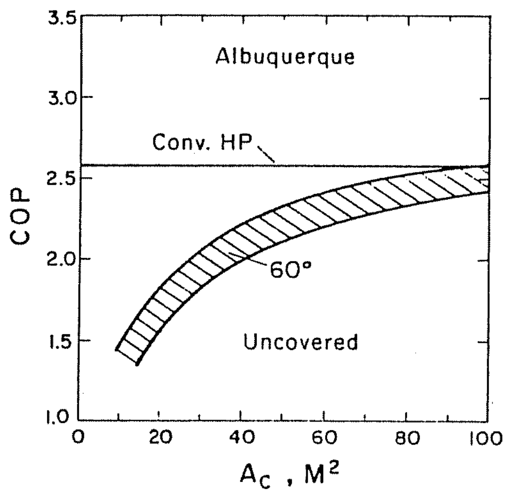

Fig. 11

Table 1

	$F_R U_L$ (W/m^{2}C)	$F_R \tau\alpha$
Covered collector	3.0	0.72
Uncovered collector	2.0	0.80
Conventional solar	4.2	0.70
Collector slope	60 C	
Heat pump size	7 kW	
Building loss coefficient	231 W/mC	

two modes—covered or uncovered—will be the most efficient. In actual operation, a covered collector would be utilized for both modes. Convective heat transfer could be obtained by blowing outdoor air between the covers and collector plate.

The results are shown in Fig. 8 for the space heating load in Madison using the parameters of Table 1. The results for Albuquerque are similar. The performance improvement over the uncovered collector system is only slight. Since the covered mode is only used when the evaporator temperature exceeds the ambient temperature, its use is limited to a small fraction of hours, which is less than 20 percent for collector areas less than 40 m^2. In addition, space heating loads are typically small during periods of radiation, which leads to the conclusion that only a small fraction of the load (less than 20 percent) can see a performance improvement with this type of control.

Performance Upper Bound With Storage

The potential performance improvements using storage with refrigeration-filled collector heat pumps is greater than that for conventional heat pumps because of the ability to take advantage of higher daytime COP. Energy delivered during daytime operation can be stored for use at night. The upper bound on system performance can be determined using the sol-air temperature bin method. The bin array is "scanned" first at the highest sol-air temperature. The heat pump capacity and electrical input contributions for these bins are totaled. The scanning continues at the next lowest bin and continues descending in sol-air temperature until the total energy delivered equals the total monthly load. Thus the method has the effect of utilizing the highest heat pump performance available during the month to satisfy the total monthly load requirement. A storage system that would store any amount of energy without losses is assumed implicitly with this method. It is also assumed that a "controller" knows when the sol-air temperature is high enough to operate the heat pump.

Upper bound performance results are plotted for the yearly Madison space heating load in Fig. 9, along with the corresponding performance curves for the nonstorage simulations described earlier. The results show a potential for substantial improvement with storage, especially for covered collector heat pump systems. The upper bound performance of the covered collector system exceeds that of the uncovered collector system in this example above 60 m^2 collector area. Some form of storage will improve the overall performance of both collector heat pump systems, especially the covered system. In many cases, however, installed systems are the uncovered collector type with no storage.

Effect of Building Capacitance

In practice, the house itself may provide some thermal storage. In order to explore this, simulations were performed for systems thermally coupled to a building with thermal capacitance. TRNSYS [8] components for walls, roof, and room were used in conjunction with the model for a refrigerant-filled collector heat pump to achieve seasonal performance results. The average overall building loss coefficient $(UA)_h$ is the same in these simulations as for the bin method. The storage of energy in the building thermal mass is achieved by allowing the room temperature to vary over a range; both 5 and 10°C were employed. The control strategy permits continuous system operation in daytime until the maximum room temperature is reached. During the night, this energy is released and then the heat pump system and the necessary auxiliary energy are used to maintain the minimum room temperature. The minimum room temperatures were 18 and 16°C, respectively, for the 5 and 10°C range. The performance of the uncovered and covered collector systems with building capacitance was compared with the zero capacitance results for Madison. With energy storage in the building, the system performance can be improved significantly. An increase in nonpurchased fraction over the no-storage case of as much as 11.5 percent of the load is achieved with the covered collector system of 48-m^2 collector area and a 10°C temperature range. The zero capacitance design method provides a lower estimate of performance, and reflects those systems which do not allow significant house heating.

Space Cooling Applications

An advantage of a heat pump lies in its ability to switch from heating to cooling upon demand. The cooling performance of a refrigerant-filled collector heat pump might be expected to be reduced due to high condenser temperatures. The potential for using the refrigerant-filled collector heat pump in the cooling mode without altering the heating mode design or orientation of the collector will be evaluted in this section. Only the uncovered collector system is considered, since its heating performance is superior to that of the covered collector system. The cooling operation is modeled similar to that for heating. The collector now becomes the condenser, and the solar energy absorbed adds to the heat that must be rejected. It was assumed that the refrigerant and plate temperatures are either equal or differ by an amount derived from manufacturer data. This sets bounds on performance. The remaining system parameters were derived from manufacturers catalog data for heat pumps in the cooling mode. The building cooling load included solar heat gain and internal heat gain effects.

The model development of this system for cooling is similar to the model for heating operation. The basic collector equation (1) is used as a starting point to account for radiation and convection heat transfer. In the cooling mode, the collector plate is used as the refrigerant condenser. Refrigerant leaving the compressor is in a superheated state and a significant amount of the condenser heat exchanger is needed to desuperheat the refrigerant before condensation at constant temperature occurs. A lower bound for plate temperature was taken as the condensing temperature, and an upper bound was derived from manufacturers data.

Figures 10 and 11 show the range of cooling COP as a function of collector area for Madison and Albuquerque during the summer (June-August). The performance is shown as a band due to the two assumptions regarding refrigerant temperature. The performance of a conventional cooling unit during the same period is also shown. The performance of collector/condenser systems is poorer at small collector areas. In Madison, the collector system uses about 24 and 14 percent more electrical energy than the conventional heat pump at an area of 30 m^2 with 60 and 90 deg collector slope, respectively. In Albuquerque, the same system at 60 deg tilt requires at least 27 percent more electrical input. The cooling performance for uncovered collector heat pumps is substantially poorer than that of conventional heat pumps, which subtracts from benefits in heating performance.

Conclusions

The major conclusions from this study are:

1 A sol-air temperature bin method for predicting the seasonal performance of refrigerant-filled collector heat pumps has been developed. This method is useful for design and evaluation of these systems.

2 Uncovered collector heat pump systems have better performance than conventional heat pumps over a wide range of collector areas and have consistently superior performance over covered collector heat pump systems.

3 The maximum attainable COP does not place a limit on the seasonal performance of the system.

4 The installed size of uncovered collector heat pump and conventional heat pumps is essentially equal to the design heating load of the residence.

5 The control of collector operation to utilize either the covered or uncovered cosystems have better performance than conventional heat pumps over a wide range of collector areas and have consistently superior performance over covered collector heat pump systems.

6 The performance of collector heat pump systems can be improved considerably with energy storage. The building itself may act as an energy storage device and contribute to increased performance.

7 Compressor cycling and start-up transients downgrade overall performance of the heat pump system, but does not change the relative performance differences between different heat pump types.

8 The cooling performance of refrigerant-filled collector heat pumps is low compared to conventional heat pumps in the cooling mode.

Acknowledgments

The assistance of Drs. S. A. Klein and J. A. Duffie is greatly appreciated. This work was supported in part by the Solar Heating and Cooling Research and Development Branch, Office of Conservation and Solar Applications, DOE, and by the Wisconsin State Energy Office.

References

1 Chaturvedi, K., Chiang, Y. F., and Roberts, A. S., "Analysis of Two-Phase Flow Solar Collectors with Application to Heat Pumps," ASME Paper 80-WA/Sol-32, ASME Winter Annual Meeting, 1980.

2 Kush, E. A., "Performance of Heat Pumps at Elevated Evaporating Temperatures—With Application to Solar Input," JOURNAL OF SOLAR ENERGY ENGINEERING, Vol. 102, Aug. 1980.

3 Krakow, K. I., and Lin, S., "A Solar Source Heat Pump with Refrigerant-Cooled Solar Collectors for Cold Climates," *ASHRAE Transactions*, Vol. 88.

4 Freeman, T. L., Mitchell, J. W., and Audit, T. E., "Performance of Combined Solar Heat Pump Systems," *Solar Energy*, Vol. 22, Nov. 1979.

5 Charters, W. W. S., "Combined Solar Heat Pump Systems," University of Melbourne, Department of Mechanical Engineering Report, Feb. 1979.

6 Dixon, C. W. S., "A Practical Study of Solar Boosted Heat Pump Applied to Space Heating and Cooling," M.S. thesis, University of Melbourne, Jan. 1979.

7 Duffie, J. A., and Beckman, W. A., *Solar Engineering of Thermal Processes*, Wiley, New York, 1980.

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

9 "SOLMET Typical Meteorological Year," Tape Deck 934, National Oceanic and Atmospheric Administration, Environmental Data Science, National Climatic Center, Asheville, N.C.

10 "F-CHART 4.0, A Design Program for Solar Heating Systems," Engineering Experiment Station Report #50, Solar Energy Laboratory, University of Wisconsin-Madison.

11 "Test Procedures for Central Air Conditioners, Including Heat Pumps," *Federal Register*, Pt. III, Department of Energy, Office of Conservation and Solar Energy, Dec. 1979.