LA-UR-85-2248

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CONF-850604--10

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LA-UR--85-2248

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SUBMITTED TO INTERSOL '85
Solar Energy Congress of the
International Solar Energy Society
June 23-29, 1985
Montreal, Quebec, CANADA

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A SELF-PUMPING VAPOR SYSTEM FOR HYBRID SPACE HEATING*

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ABSTRACT

This paper reports investigations of a passive system that transports heat from a solar collector downward. Refrigerant is evaporated in a collector and the vapor transports heat to one or more condensers located at an arbitrary elevation below the collector. The vapor pressure lifts the condensed liquid to an accumulator located near the top of the collector. The results of annual performance calculations are presented for a space heating application in which one or more condensers are located within passive thermal storage units that release heat directly to one or more zones of the building. Simple sizing rules for optimal energy yield are given.

KEYWORDS

Self-pumping, vapor transport, passive solar, hybrid solar, refrigerant.

DESCRIPTION OF THE SELF-PUMPING SYSTEM

Background

Most hybrid systems utilize active collection with pumps and controllers. We are investigating a passive system that transports heat across arbitrary distances at low temperature difference, that requires no external pumps or controls, and that does not freeze, corrode, or lose heat at night. Several papers [1] discuss self-pumping schemes. Workers at Ispra [2] have experimented with an arrangement similar to the first system that we studied [3,4], which had a single accumulator. We showed that this system requires precise external cooling of the accumulator for optimal operation, and we therefore conceived a two-accumulator system that provides internal cooling.

Two-accumulator System

Figure 1 shows one version of the two-accumulator system, which utilizes

*Work supported by US Department of Energy, Office of Solar Heat Technologies.

three check valves. All plumbing and both accumulators are insulated, so that almost all heat transfer occurs only at the collector or at the condenser, which is located in thermal storage. The pressure and temperature of the upper accumulator are kept close to those of the condenser by the uninterrupted vapor link between these components. When the float valve is open, vapor from the collector passes to the condenser, and condensed liquid drains back into the lower accumulator. During this phase of the cycle, the collector, condenser, and upper accumulator are at nearly equal pressures, and liquid continuously drains from the upper accumulator, keeping the collector nearly filled. When the lower accumulator is full, the float valve closes causing the pressure and temperature of the collector and lower accumulator to rise, lifting liquid into the upper accumulator. When the lower accumulator is nearly empty, the float valve opens. The lower accumulator rapidly cools to the temperature of the condenser due to the renewed vapor connection, and the cycle begins again.

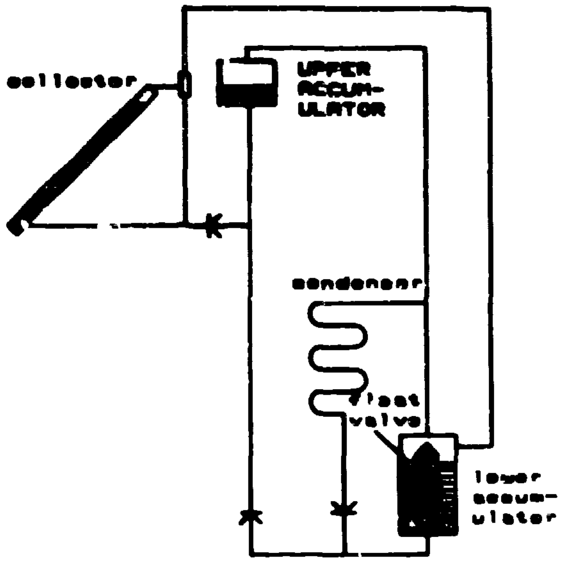

Fig. 1 Two-accumulator vapor system.

The two-accumulator system has several advantages when compared with the single-accumulator system. a) During the condensing phase of the cycle, the collector is continuously fed with liquid. b) During the condensing phase, the collector temperature is close to the storage temperature. In the single-accumulator system, the collector temperature is always elevated by at least the amount required for liquid lift, and may be elevated much more due to imperfect cooling of the accumulator. c) After the pumping phase, nearly perfect cooling of the accumulator occurs as its excess thermal energy is transferred to the condenser as useful heat.

TESTS OF THE TWO-ACCUMULATOR SYSTEM

We constructed an indoor mockup of the two-accumulator system in which the solar collector is simulated by an electrically-heated copper evaporator of approximately the same mass and volume as the absorber of a fin-tube collector with 2 m^2 area. The working fluid is R-11 (trichloromonofluoromethene). The condenser is a 15 m coil of 16 mm OD copper tubing immersed in an uninsulated tank containing 185 l water. Liquid lift is approximately 5 m. For this experimental mockup, the time-average elevation of collector temperature above storage temperature is given by

$$\Delta T = \Delta T_0 + \frac{1}{U_H} Q, \qquad (1)$$

in which Q is the simulated delivered power per unit collector area and U_H is similar to a heat transfer coefficient for a heat exchanger. Although the elevation of average collector temperature can presumably be reduced, we have used this empirical equation in our numerical simulations of the self-pumping space heating system.

Description of the Numerical Model

Our computer program represents an active collection system that delivers solar heat to a water tank, which passively delivers heat in an uncontrolled manner to the room. For simulation of the vapor system, the flow rate and heat transfer quantities were set to appropriate values, as determined from previous validation with test cell data [5]. Figure 2 shows diagrams of the computer model of the self-pumping vapor system and of the pumped liquid drainback hybrid system (circulating water) with which the vapor system is compared. (Although the vapor system is strictly passive, we regard it as a hybrid system because of its similarity with the pumped hybrid system.) Auxiliary heat is supplied if necessary to maintain the lower limit of room temperature at 18.3 C. The upper limit of room temperature is 23.9 C, above which energy is presumed to be vented. The calculated solar savings fraction (SSF) is based on the energy required to maintain the building at 18.3 C. In our calculations, ALL SYSTEM PARAMETERS ARE NORMALIZED TO COLLECTOR AREA--for example, the load/collector ratio (LCR) is the building load coefficient in watts per degree Centigrade, divided by the collector area. Calculations for the pumped liquid hybrid system and for the vapor system show nearly identical results. Hence, all results shown in this paper will be for the vapor system, with the understanding that the same results apply for the pumped liquid system.

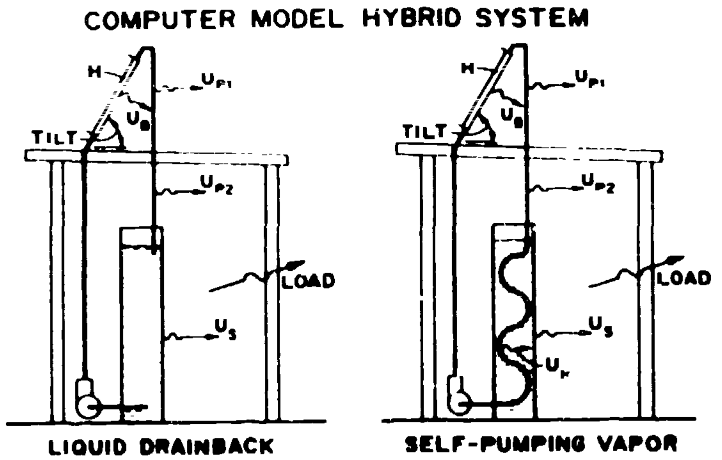

Fig. 2 Diagram of the computer models.

Comparison with Water Walls

If the hybrid system were idealized so that there were no losses from the back of the collector or from the pipes ($U_B = U_{P1} = U_{P2} = 0$), then it should behave very much like a water wall with selective surface and night insulation. Figure 3 shows the annual SSF for this idealized case with the collector tilted at 90° to match the tilt of a water wall. The curves labeled WWC1 and WWC3 represent computer-generated correlations for single-glazed, selective-surfaced water walls without and with night insulation, respectively [6]. Calculated points are also shown for the base case system with the parameters as given in Table 1. The water walls and the vapor system of Fig. 3 all have the same thermal storage (955 kJ/C per square meter of collector), and the same storage-to-room heat transfer coefficient (U_S = 8.5 W/C per square meter of collector). Albuquerque and Buffalo represent extremes of sunny and cloudy climates. Figure 3 shows that the calculated performance of the idealized system does indeed approximate that of the night-insulated water wall, and that the thermal losses of the more realistic base case cause a significant penalty. We suspect that in some instances the hybrid system provides less energy than predicted by the WWC3 correlations because of differences in the way the two computer programs treat ground-reflected and diffuse radiation, and because the correlations are not necessarily an exact representation of the original water wall calculations.

We conclude that the vapor (or the pumped liquid) hybrid system behaves approximately as a night-insulated water wall, but with practical advantages of allowing an elevated, tilted collector and heat delivery to locations remote from the collector.

Table 1 Parameters of Idealized and Base Case Systems

	Base	Ideal	
W	283	283	W/m_c^2C
U_B	1.7	0.05	"
H	113	113	"
U_{P1}	0.28	0.	"
U_{P2}	0.28	0.	"
U_H	80	80	"
ΔT_0	1.5	1.5	C

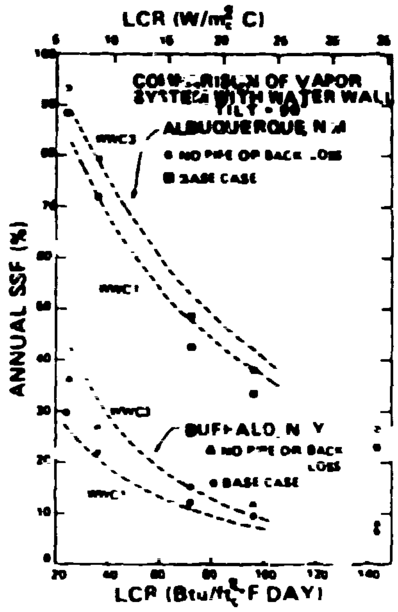

Fig. 3 Comparison of vapor system performance with correlations for waterwalls.

The Effects of System Parameters

Figures 4-6 illustrate the dependence of annual SSF on U_S, storage capacity, and LCR. Figure 4 shows that, for storage greater than 600 kJ/C per m^2 of collector area, the energy savings has a broad maximum near U_S = 12 W/C per m^2 of collector area. Figure 5 shows the the dependence of SSF on storage, with each data point taken near the value of U_S that maximizes SSF at the particular value of storage. Figure 5 shows that the annual SSF of vapor or pumped liquid hybrid systems is insensitive to the amount of storage between 600 and 1200 kJ/m_c^2C. Figure 6 shows the dependence of SSF on LCR for a nearly optimal system with U_S = 11.3W/m_c^2C, and storage of either 612 or 1225 kJ/m_c^2C. From these figures we propose some rules of thumb for design: LCR is chosen for the desired solar fraction, U_S should be in the range 10-15 W/m_c^2C, and storage should be in the range 600-1000 kJ/m_c^2C. Swisher [7] suggests that storage should be at least 800 kJ/m_c^2C.

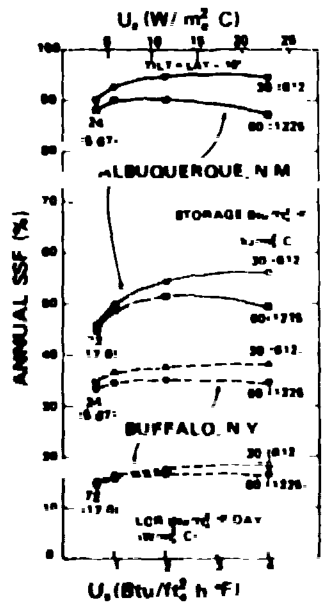

Fig. 4 SSF as a function of U_S

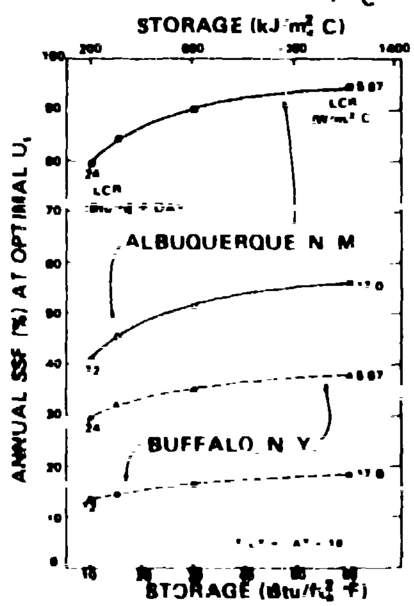

Fig. 5 SSF as a function of thermal storage capacity.

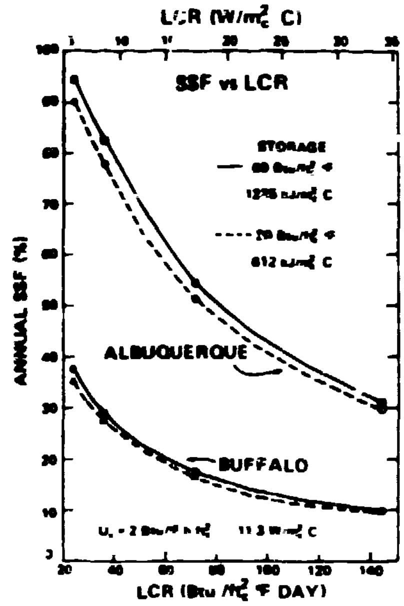

Fig. 6 SSF vs LCR.

In the sunny climate of Albuquerque, the daily swings of storage temperature are larger than in less sunny climates. With 612 kJ/m_c^2C of storage capacity, the average daily extremes of storage temperature during winter are approximately 25 and 38 C in Albuquerque. Because the daily maximum storage temperature is well above the desired room temperature, the effects of mean radiant temperature may be significant as Swisher suggests [7]. Additional energy savings might therefore result from a reduction of thermostat setpoint, or by use of a manually operated curtain that is drawn across the exposed surface of storage when excess heating occurs.

NOMENCLATURE

ΔT	Elevation of collector temperture above storage temperature.
ΔT_O	Simulated heat exchanger temperature offset.
U_B	Back loss from plate heat transfer coefficient (per collector area).
U_H	Heat exchanger heat transfer coefficient (per collector area).
U_{P1}	Outdoor pipe loss heat transfer coefficient (per collector area).
U_{P2}	Indoor pipe loss heat transfer coefficient (per collector area).
U_S	Storage-to-room heat transfer coefficient (per collector area).
H	Plate-to-fluid heat transfer coefficient (per collector area).
W	Simulated thermal capacity of fluid flow rate (per collector area).

REFERENCES

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