EXPERIMENTAL STUDY OF A SELF-PUMPING BOILING COLLECTOR SOLAR HOT WATER SYSTEM

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ABSTRACT

An experimental study of a pilot scale passive solar hot water heating system utilizing a boiling collector is presented. The self-pumping system alternates between two modes of operation. During the run cycle, vapor pressure drives the evaporated refrigerant downward from the collector to the condenser. Once a preset quantity of refrigerant is condensed, vapor pressure is again used to force the return of the condensate to the collector during the pump cycle. The pilot system is optionally configured to operate with a mechanical pump. Experimental results include measurements of temperatures and pressures within the system as well as thermal input and output energies. The loss in thermal efficiency due to the use of vapor pressure to return liquid refrigerant to the collector is assessed by comparing self-pumping and mechanical-pumping operation. Performance of the self-pumping system is strongly influenced by transient thermal losses. Refinement of the system design, including individual components, is required to reduce losses and improve performance.

INTRODUCTION

A boiling collector solar system has several advantages in comparison to more traditional sensible heat collector systems. Improved efficiencies are possible due to higher heat transfer coefficients [1,2]. The use of a refrigerant as the working fluid also eliminates problems associated with freezing, scaling and corrosion. But perhaps the primary advantage of the boiling collector system is that it can be configured to operate as a self-controlling and self-pumping passive vapor downward heat transport system. In such a system, the vapor pressure not only drives the vapor from the elevated collector to the condenser but also forces the return of the condensed liquid to the collector. The capital, operating and maintenance costs of a mechanical pump are eliminated, and since the system operates only when sufficient solar energy is available, the controller used in conventional systems is not necessary. Moreover, the system may be designed to operate without electricity.

The major drawback of a boiling collector self-pumping system is a reduction in thermal efficiency caused by the elevated collector pressure and temperature needed to lift the condensed refrigerant to the collector. Since the collector temperature rise needed to pump the liquid is a function of the lifting height, the reduction in efficiency imposed by self-pumping increases as the height difference between the condenser and collector is increased.

This paper presents the results of an experimental study of the pilot scale self-pumping solar hot water system shown in Fig. 1. The basic system consists of a single solar collector panel, a separator, two refrigerant accumulators, a small refrigerant reservoir, a tube-in-shell heat exchanger, water storage tanks and associated piping and valves. The system operates in cycles, alternating between a run cycle and a pumping cycle. During the run cycle, shown in Fig. 1a, liquid refrigerant (R-113) is gravity fed to the roof-mounted collector from the upper accumulator tank and a small auxiliary reservoir until the liquid level in the upper accumulator reaches a lower float switch. Any liquid exiting the collector is recirculated through the separator. Vapor exiting the collector travels through the vapor space in the upper accumulator to the shell side of the condenser. The condensed refrigerant collects in the lower accumulator. During this mode of operation, solenoid valve #2 is closed to prevent vapor condensation in the piping connecting the collector and lower accumulator and also to maintain the lower accumulator at a low pressure so that the condenser is not flooded.

Once the lower float switch is tripped, the system switches to the pump cycle, as shown in Fig. 1b. Condensed refrigerant is returned to the upper accumulator by utilizing the pressure differential created by vapor pressure build-up in the collector. Solenoid valve #1 closes the line to the condenser and solenoid valve #2 opens the vapor line from the collector to the lower accumulator. Refrigerant in the small reservoir continues to drain into the collector to avoid dry out. Vapor produced in the collector is prevented from condensing, and as a result the temperature and pressure increase in the collector and lower accumulator. The piping connecting the upper...
accumulator to the condenser permits the pressure and temperature in these two components to remain low. When the pressure in the lower accumulator exceeds the pressure in the upper accumulator by an amount equivalent to the head difference (approximately 6.4 m), the condensate is forced from the lower accumulator to the upper accumulator. Once the upper accumulator is filled, a high level float switch is tripped, and the vapor line connecting the collector and upper accumulator is opened. The collector and lower accumulator then equilibrate to the pressure and temperature of the condenser, and the run cycle is repeated.

The system can also be operated with a mechanical pump. Mechanical-pumping operation is identical to that described for self-pumping except that during the pumping cycle, the condensate is quickly returned to the upper accumulator by the pump rather than by pressure build-up in the collector.

**BACKGROUND**

Many different designs for passive downward heat transport have been proposed. A comprehensive review of numerous early self-pumping schemes is presented by both Wachtell [3] and Roberts [4]. Most of these early systems are unnecessarily complex, but their design principle, based on the pressure-temperature cycles inherent to the evaporation/condensation process, is the basis of more recent system designs. Much progress has been made in the last decade towards understanding and optimizing downward vapor transport systems intended specifically for solar applications. Several patents propose self-pumping systems incorporating a single refrigerant reservoir or accumulator [5–7], and Knecht [8] obtained a patent in 1982 for a two-accumulator system similar in concept to the system presented in this paper. In addition to the experimental work at the Solar Energy Applications Laboratory (SEAL), significant experimental tests of solar self-pumping systems are reported by Hedstrom and Neeper at Los Alamos [10–15], and De Bini and Italian co-workers [16,17].

Hedstrom and Neeper experimented with several vapor transport solar systems including an early percolation system [10] which proved unreliable and two self-pumping systems. Both self-pumping systems used the refrigerant R-11 and had a maximum lifting height of 5 m. Their first self-pumping system operated in cycles much like ours, but was configured with a single accumulator located slightly above the collector and a condenser submerged in a large water storage tank [11–13]. The single accumulator design has three major flaws. Most importantly, the collector temperature is always higher than the storage temperature by at least the amount required for lifting the liquid and, in most cases, is elevated even more. The collector temperature can only be lowered by externally cooling the accumulator. Thus optimizing system thermal efficiency requires a tradeoff in the thermal losses due to accumulator cooling and the losses due to high collector temperatures. Second, the condenser is used ineffectively. The pressure differential necessary for pumping results from flooding the condenser during the run cycle and thereby decreasing the heat transfer area until the reduction in heat transfer causes the necessary rise in collector temperature and pressure to lift the liquid from the condenser to accumulator. And third, since no provision is made for liquid flow to the collector during pumping, the collector may dry out.

Independently, Hedstrom and Neeper conceived the two-accumulator system patented earlier by Knecht and similar to our system. This design solves the problems experienced with the earlier single accumulator system. The first tests of this design were conducted indoors with an electrically heated evaporator located approximately 5 m above the condenser [13,14]. Later tests were conducted with a single solar panel under conditions more realistic of residential architectures.
where solar collectors are typically mounted on the roof. Daily thermal efficiencies as high as 45 percent are reported.

The first system tested by De Beni et al. [16] had a single accumulator located within the storage water and pressurized with an inert gas. Tests of the system using a boiler as the evaporator were promising; however, the requirement that the evaporator dry out prior to pumping is unsatisfactory for solar applications. A later bench scale boiler powered self-pumping system using R-114 tested by this group [17] is practically identical in principle to the single accumulator solar system of the Los Alamos group.

The self-pumping system at SEAL was first operated during the 1986-87 winter [9]. Results for this early system confirmed the viability of the design concept. Measured system efficiencies and thermal output were however below those of many glycol and water circulating systems. During the 1987-88 year, the system was modified to reduce condensation losses in the connecting piping and to permit mechanical as well as self-pumping. The results reported in this paper were obtained from November, 1987 through April, 1988.

EXPERIMENTAL FACILITY AND PROCEDURES

The Self-Pumping System

The pilot scale self-pumping boiling collector system is located in Solar House III, one of five residential style solar heated and cooled buildings at SEAL. Details of the building structure are not included since the boiling collector system is not part of either the space heating or water heating systems.

A detailed system schematic indicating relative component elevations and instrumentation is shown in Fig. 2. The collector is located on the uninsulated outer roof at a 45 degree angle with respect to the horizontal. The separator, reservoir, upper accumulator, and associated piping and valves are located between the outer roof and a weather-tight inner house roof. The three sections of piping that connect the collector to the condenser and lower accumulator run along the outside of the building from the roof area to an enclosed ground level room where the condenser, lower accumulator, and two 150 liter hot water storage tanks are located.

The condenser is a manifolded tube flat plate collector with a gross surface area of 1.7 m². The glazings are tempered glass and the absorber coating is black chrome. The tube-in-ribbon absorber plate is constructed of 0.081 cm copper and contains six 0.64 cm nominal diameter tubes with 1.27 cm headers.

The condenser is a single pass tube-in-shell heat exchanger. The refrigerant condenses in the shell while storage water circulates through the tubes. The condenser is oversized for this application and thus the difference in condensing and storage temperatures is minimized.

The lower accumulator and reservoir are commercial refrigerant tanks. The 50 pound lower accumulator tank has a 19 liter capacity and the 30 pound reservoir tank holds approximately 11 liters of refrigerant. The 26 liter upper accumulator is constructed of a thicker wall air compressor tank and is equipped with two magnetic float switches which control the two solenoid valves. The length of the run cycle is varied by adjusting the levels at which the float switches are triggered. All the components and piping are insulated.

RESULTS

Energy output is the product of the insolation and the gross area of the solar collector. Energy output is the heat across the condenser, \( Q_{\text{cond}} \) equal to \( mc_{p}\Delta T \) where \( m \) is the water mass flow rate, \( c_{p} \) is the water specific heat and \( \Delta T \) is the water temperature rise measured across the condenser.

Self-pumping system. Typical operating characteristics of the self-pumping system are illustrated by data obtained on April 6, 1988. Average
ambient temperature and pressure were 22.6 deg C and 84.5 kPa. Figure 3 is a plot of the collector inlet and outlet temperatures and refrigerant and water condenser inlet temperatures throughout the day. Figure 4 shows pressures at the condenser inlet and in the upper and lower accumulators. Figures 6 and 7 show these same temperatures and pressures on an expanded scale for one mid-day pump and run cycle. Figure 5 is a plot of the instantaneous energy across the condenser and insolation throughout the day. Mode of operation is indicated by the dotted cycle status line. The system is in the run cycle when the status line is high. Significant points are identified with a number designation. Daily operation is characterized by three distinct periods: morning transient, mid-day operation alternating between the run and pump cycles, and evening transient. On this particular day, the morning transient period begins at 5:38 am at sunrise with the system in the run cycle (point 1). (The first cycle of each day is determined by the status of the system at the end of the previous day.) This period of operation is characterized by slow heating of the system as it recovers from overnight cooling of the refrigerant and hardware. The onset of liquid thermosiphon recirculation within the collector loop is indicated in Fig. 3 at point 2 by a change in the slope of the rise in collector inlet temperature from 2.5 deg C/hr to 16.0 deg C/hr. Liquid circulation has a stabilizing effect on collector loop pressures. Refrigerant boiling begins at 7:30 am (point 3). Although not shown on the temperature plot, six thermocouples along the flow direction of the collector show a change from a monotonically increasing temperature profile, characteristic of a sensible heat collector, to a profile characteristic of a boiling collector in which the outlet temperature is lower than the temperature at the liquid/vapor interface. As shown in Fig. 3, collector outlet temperature drops at point 3 from 25.1 deg C to 18.6 deg C. A corresponding sudden increase in condenser inlet temperature from 16.3 deg C to 18.6 deg C indicates that vapor is being introduced into the condenser. The condenser inlet temperature is, however, less than that of the water (39.5 deg C). Thus, as shown in Fig. 5, heat transferred to storage is negative. It is not until three hours after sunrise (point 4) that the vapor temperature in the condenser reaches the water temperature (39.6 deg C) and useful energy is recovered. At this point, Fig. 5 indicates a sharp increase in the heat across the condenser from 0.01 kW to 0.31 kW. At the end of the morning transient period, condenser temperatures remain close to the water storage temperature and pressures throughout the system stabilize during subsequent run cycles. The length of run and pump cycles vary during the day due to changing ambient conditions. However, characteristics of the temperature and pressure cycles are similar during the mid-day period and the single pump/run cycle shown in Figs. 6 and 7 is selected for discussion. At the beginning of the 11:40 am pump cycle (point 5), temperatures and pressures within the collector loop and lower accumulator begin to increase. The duration of the pumping cycle depends on the time necessary to increase the pressure in the lower accumulator sufficiently above that of the upper accumulator to force the return of the condensed refrigerant to the upper accumulator. The pressure differential needed for the 6.4 m height difference in the two accumulators is approximately 96 kPa at a water storage temperature of 40 deg C. The temperature and pressure levels required in the lower accumulator for pumping increase throughout the day (from 65.1 deg C and 74.2 kPa for the first cycle to 70.7 deg C and 105.5 kPa for the last cycle) as the storage water temperature (from 39.9 deg C to 48.9 deg C) and thus upper accumulator pressure (from -21.9 to 14.6 kPa) increase. As shown in Fig. 7, this particular pump cycle lasts approximately 20 minutes. Thirteen minutes after the initiation of the pump cycle,
the pressure differential is sufficient to lift the collected liquid. This point is indicated by the knee in the collector outlet pressure curve at point 6. As the liquid is forced upward, the collector and lower accumulator pressures must continue to increase (although at a slower rate) since the height of the liquid column which must be raised increases as the upper accumulator fills.

At point 7 in Figs. 6 and 7, the mid-day pumping cycle ends once the upper accumulator is filled, and the run cycle begins. Temperatures and pressures in the collector loop and lower accumulator suddenly decrease as the solenoid valve #1 opens to allow the hot vapor to expand into the upper accumulator and condenser. As a result, temperature and pressure in the condenser increase and the heat across the condenser increases sharply as shown by the conspicuous spike in Fig. 5. Approximately 10 minutes into the run cycle (point 8), the collector inlet temperature drops more than 20 deg C (from 61.5 deg C to 40.0 deg C) as subcooled liquid flows from the upper accumulator to replace fluid depleted from the collector and reservoir during the pump cycle. This temperature drop is responsible for the depression in heat across the condenser indicated in Fig. 5.

During the remaining 12 minutes of the run cycle (from point 9 to point 10 on the figures), the refrigerant at the collector inlet equilibrates to a slightly higher temperature (47 deg C) due to mixing of the subcooled liquid from the upper accumulator (37.2 deg C) and liquid recirculated from the collector outlet via the separator and reservoir (47.4 deg C). Based on an energy balance at the tee where the refrigerant lines join upstream of the collector inlet, it is estimated that as much as 90% of the fluid exiting the collector is returned to the inlet as liquid with only 10% passing on to the condenser as vapor. The difference between collector outlet and water temperature is 4.6 deg C. Heat across the condenser rebounds to a value of 0.9 kW during this period.

The system continues to alternate between run and pump cycles during the middle of the day. However, high incident angles and atmospheric attenuation typical of the late afternoon reduce the available solar energy input to a level too low for pumping (point 11). As shown in Fig. 4, the pressure in the lower accumulator increases during the last pump cycle but never achieves the level necessary for pumping. All the energy collected during the last two hours of daylight is lost as the system cools to the ambient.

Daily efficiency of the system on this day is 29 percent. A maximum run/pump cycle efficiency of 38 percent is achieved mid-day between 1:00 and 2:00 pm.

**Mechanical-pumping system.** On April 11, the system was operated with the mechanical pump. The average ambient temperature and pressure were 18.8 deg C and 88.3 kPa. Contrasted to the self-pumping system characteristics, mechanical-pumping during the day temperatures and pressures are nearly uniform throughout the day. Figure 8 is a plot of selected temperatures and pressures as well as insolation and heat across the condenser. Temperatures shown include: collector inlet and outlet, refrigerant and water at the condenser inlet, and ambient. Pressures plotted are those at the condenser inlet and the lower accumulator.

The system begins in the run mode at sunrise (5:33 am). Unlike the self-pumping system, the mechanically pumped system always begins and ends the day in the run mode. An increase in condenser inlet temperature indicates boiling in the collector. At this point early in the day, the refrigerant temperature in the condenser is less than the water temperature and the measured heat across the condenser is 0.16 kW. The morning transient or warm-up period, during which no useful energy is produced, lasts nearly three hours until 8:26 am (point 2).

A dramatic drop in temperature at the top of the collector (from 55.4 deg C to 38.1 deg C) at point 2 in Fig. 8 indicates that boiling has begun lower in the collector. This is confirmed by an increase in
of this system parameter reflects the dependency of system performance on thermal losses. Measured self-pumping daily efficiencies range from 1 to 31 percent. The average daily efficiency is 17 percent. Mechanical-pumping thermal efficiencies range from 5 to 36 percent with an average value of 26 percent. The energy required for mechanical pumping is not included in the efficiency calculation.

**Seasonal Performance**

Daily thermal efficiencies, \( \eta \), of both systems are plotted in Fig. 9 as a function of the system parameter \((T_s - T_a)/I\) where \(T_s\) is the average water storage temperature in deg C, \(T_a\) is the average ambient temperature in deg C and \(I\) is the average insolation in W/m². The decrease in efficiency with increasing values of this system parameter reflects the dependency of system performance on thermal losses. Measured self-pumping daily efficiencies range from 1 to 31 percent. The average daily efficiency is 17 percent. Mechanical-pumping thermal efficiencies range from 5 to 36 percent with an average value of 26 percent. The energy required for mechanical pumping is not included in the efficiency calculation.

**Fig. 8.** Temperatures (collector inlet and outlet, refrigerant and water at the condenser inlet, and ambient), pressures (refrigerant at the condenser inlet and lower accumulator), and energy input (insolation) and output (heat across the condenser, \(Q_{\text{cond}}\)) of mechanical pumping system plotted every minute, April 11, 1988.

**Fig. 9.** Daily thermal efficiency as a function of the inlet parameter \((T_s - T_a)/I\) where \(T_s\) is the average water storage temperature, \(T_a\) is the average ambient temperature and \(I\) is the average insolation.

A linear regression technique is used to correlate the efficiency data for both systems. Daily efficiencies are best expressed as:

\[ \eta = 0.28 - 2.1(T_s - T_a)/I \]  
(1)

for the self-pumping system and

\[ \eta = 0.46 - 4.7(T_s - T_a)/I \]  
(2)

for the mechanical pumping system. A low correlation coefficient of 0.7 for both of these expressions reflects the scatter in the data and indicates that factors other than the chosen system parameter influence system performance. The gap between the mechanical-pumping curve and the self-pumping curve represents the reduction in thermal efficiency due to using solar energy to return the condensed refrigerant to the elevated collector loop. Approximately 20 percent of the total collected energy is used during the self-pumping cycle.

Figure 10 is a plot of average daily energy output of the system, based on heat across the condenser \((Q_{\text{cond}}, \text{MJ})\), as a function of solar energy input to the system \((Q_{\text{solar}}, \text{MJ})\). Linear correlations for this data are:

\[ Q_{\text{cond}} = 0.26(Q_{\text{solar}}) - 1.8 \]  
(3)

with a correlation coefficient of 0.8 for the self-pumping system, and

\[ Q_{\text{cond}} = 0.37(Q_{\text{solar}}) - 2.8 \]  
(4)

with a correlation coefficient of 0.9 for the mechanical-pumping system. Both curves indicate that the systems are inoperative if the total available solar...
The length of the pumping cycle is crucial to the design of any self-pumping system. It is during this mode of operation that collector temperatures are elevated and thus thermal losses are greatest. The length of the pumping cycle depends on both the lifting height and the quantity of refrigerant in the reservoir that must by elevated in temperature and pressure. By reducing the size of the reservoir, the time to reach the necessary pressure to lift the condensate can be reduced. Once again a tradeoff is necessary. If the reservoir is too small, the collector may dry out during the pumping cycle and the system will not be able to pump.

An optimization study is underway to determine the effects of various system parameters and design alternatives on overall system efficiency. Future evaluation of the system will also explore the use of alternate refrigerants to reduce the environmental hazard of long term accumulation of chlorinated fluorocarbons (CFCs) in the atmosphere. On the basis of this work, a newly designed system will be evaluated from both an economic and operational standpoint.

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REFERENCES


