Modified RC Thermal Circuit Model Applied to Cold Storage System with Multi-Loop Heat Pipes

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ABSTRACT

This paper proposes a theoretical model to investigate the thermal performance of a cold storage system with multi-loop wickless heat pipes. The cold storage system utilizes the superior heat transfer characteristics of heat pipe and eliminates drawbacks found in the conventional thermal storage tank. A modified RC circuit model to determine the thermal characteristics of the cold storage system has been developed. Experimental investigations are then conducted to study the cold storage thermal performance in an experimental system with the ratio of distance between heat pipes to outer diameter of heat pipe W/D = 2. Different heat transfer mechanisms, including nucleate boiling, geyser boiling, and natural convection, are identified in different experimental systems with various liquid fills. This paper probes the effect of the fill level on cold storage rate and cumulative cold storage quantity. Comparisons of this theory with experimental data show good agreements in the nucleate boiling stage of cold storage process.

INTRODUCTION

A number of thermal energy storage systems (Yimer and Madami 1997; Hasnain 1998; Dincer and Rosen 2001; Dincer 2002) have been considered and developed in recent years. Most of the energy storage systems utilize an active control method to store or release thermal energy. That is, in the system design of thermal storage, a pump is included to transfer thermal energy from a high-temperature heat source to the thermal storage tank via flowing working fluid. To utilize the stored thermal energy, an electromagnetic valve is used under control to change the flow path of the working fluid so that energy stored in the storage tank is released to and used by a low-temperature heat sink. There are two drawbacks found in such a type of thermal storage system. First, the need for an operation cost and the power consumption of the pump. The thermal storage shall be unworkable in case of a system failure. Second, change of the charge and discharge ability of the conventional storage systems basically relies on the system piping design; therefore, only two functions, i.e., energy storage and energy release, are available in its operating modes. It is impossible for both the heat supply side and the heat utilization side of the thermal storage to operate at the same time during the energy utilization. A new cold storage system is proposed in this paper, which utilizes the superior heat transfer characteristics of heat pipe and eliminates drawbacks found in the conventional thermal storage tank.

The cold storage system with multi-loop heat pipes, as shown in Figure 1, consists of an energy storage tank and two heat pipe loops. The energy storage tank is filled with phase-change medium (PCM) so that thermal energy can be stored or released via melting or freezing of the phase-change medium between solid and liquid states. Insulating material is provided to cover the outside of the energy storage tank to prevent heat loss. A top cover is provided at the top of the tank for replenishing the phase-change medium into the chamber, and a drain hole is provided at the bottom of the tank for draining the phase-change medium. The heat pipe loops include three parts, namely, a group of parallel wickless heat pipes vertically disposed inside the energy storage tank and vertical high-temperature heat exchange and vertical low-temperature heat exchanger separately located outside of the tank. The parallel heat pipes combine the high-temperature heat exchanger to form a two-phase closed-loop thermosyphon for cold storage. The cold release loop is constructed by connecting the parallel heat pipes and the low-temperature heat exchanger. The paral-

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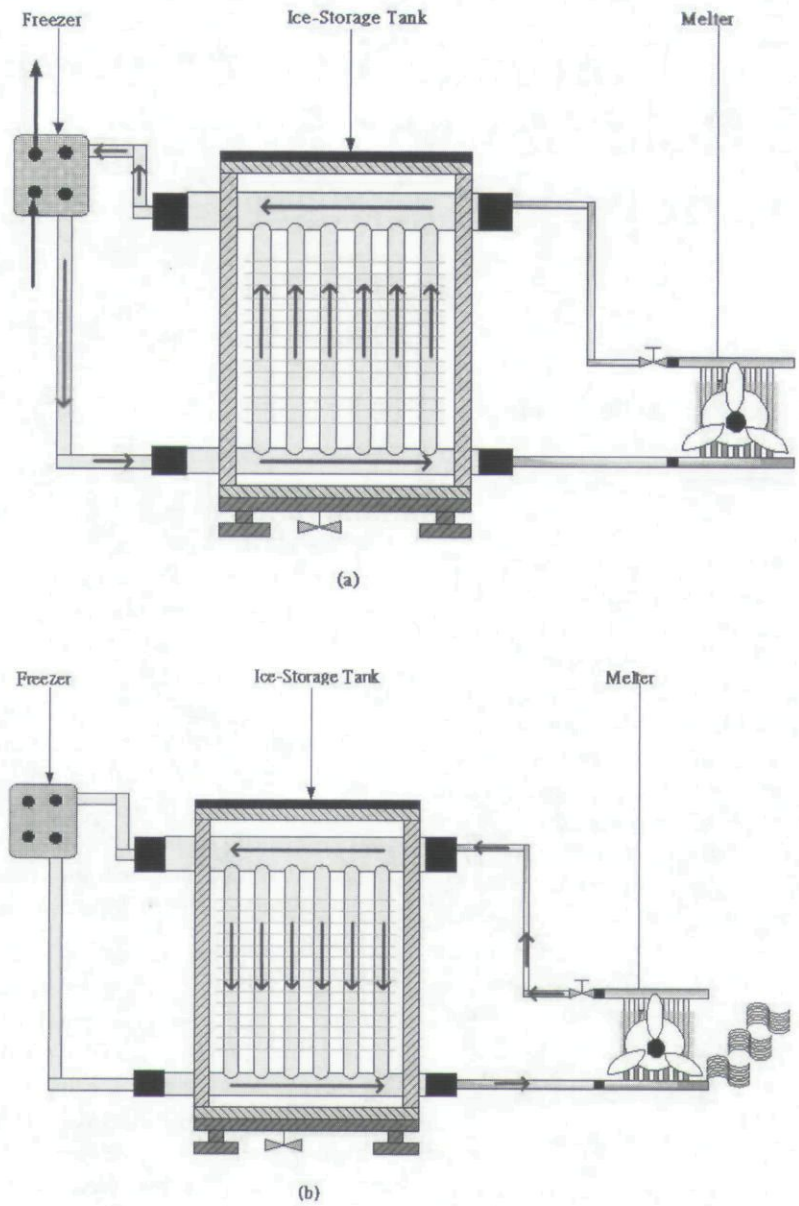

Figure 1 *Cold storage system with multi-loop heat pipes: (a) charge mode (b) discharge mode.*

lel heat pipes have external short fins densely provided around their outer surfaces to increase thermal conductive contact areas thereof. These short fins also divide inner space of the energy storage tank into multiple energy storage cells. The phase-change medium becomes molten or frozen in these energy storage cells to store or release cold energy. The vertical high/low-temperature heat exchangers outside the energy storage tank are used to exchange heat with high-temperature and low-temperature flowing fluid, respectively. An adequate amount of working fluid is filled in the heat pipe loops.

Figure 1 shows the manner in which the cold storage system with multi-loop heat pipes operates to store cold energy. When an amount of low-temperature flowing fluid flows into the low-temperature heat exchanger in a direction as shown by the arrows in Figure 1a, it absorbs heat in the vapor working fluid inside the vertical low-temperature heat pipes and is heated to have increased enthalpy value. The vapor working fluid inside the heat pipes condenses into a liquid working fluid that forms a thin layer of condensate along the inner wall surface of the vertical low-temperature heat pipe, then flows downward under gravity into the vertically parallel heat pipes. At this point, the liquid working fluid in the vertically parallel heat pipes absorbs energy stored in the liquid phase-change medium in the cells outside the heat pipes and comes to a boil to produce vapor working fluid that flows upward due to its buoyancy into the vertical low-temperature heat pipe to complete one cycle. Energy stored in the phase-change medium is transferred to the low-temperature working fluid flowing through the vertically parallel heat to boil and evaporate the working fluid and thereby freeze the liquid phase-change medium into a solid state. Figure 1b shows the function in which the cold storage system with multi-loop heat pipes operates to discharge cold energy.

The RC thermal circuit model, which is analogous to the RC electrical circuit model, is proposed to investigate the cold storage system with multi-loop heat pipes. The analogous electric circuit is shown in Figure 2. It comprises thermal resistance of each heat transfer device (R), thermal capacity of the energy storage tank (C), heat transfer rate in the cold storage system (q), and each temperature (T). Those parameters in the RC thermal circuit model can be analogous to the electric resistance, electric capacity, current, and electric potential, respectively, in the RC electrical model.

The thermal RC concept has been successfully applied to temperature control engineering owing to its simplicity for complex heat transfer problems. Engeler and Garfinkel (1965) uses the thermal RC concept to analyze the temperature rise at the junction of a GaAs laser for a variety of pulse currents and base temperatures. It is shown that the thermal behavior of the diode may be calculated analytically at room temperature because of the constancy of the thermal parameters. Min et al. (1990) studied the transient thermal property of semiconductor devices with the RC concept and the method of images. The software developed is useful for a variety of device structures under pulsed conditions with high peak power or switching conditions. Wu (2000) utilized a simple RC thermal circuit similar to an electrical circuit to analyze the controlled system composed of an electrical heater and oven fast. It is shown that RC thermal circuit offers good assistance for temperature control of a system.

This paper presents theoretical analysis and experimental investigations of the charge characteristics of the cold storage system with multi-loop heat pipes. A modified RC thermal circuit model is proposed to study the thermal characteristics of the cold storage system. Experimental investigations are performed in an experimental system with a ratio of distance between heat pipes and outer diameter of the heat pipe W/D = 2. Flow patterns are identified in different experimental systems with various liquid fills. This paper also probes the effect of the fill level in multi-loop heat pipes on the cold storage rate and cumulative cold storage quantity. The parameters, including thermal resistance, refrigerant temperature, and cumulative storage quantity, are discussed and compared with experimental measurements.

MODIFIED RC THERMAL CIRCUIT MODEL

The modified RC thermal circuit model is shown in Figure 2. It consists of source temperature potential ΔT_o, thermal resistances of R_{t1} and R_{t2}, thermal capacity C_t, and environmental temperature. Source temperature potential represents the temperature difference in low temperature of the flowing fluid in the low-temperature heat exchanger and environmental temperature. The thermal resistance of R_{t1} includes the convective thermal resistance of the low-temperature flowing fluid, the conductive thermal resistance of the low-temperature heat exchanger, and the convective thermal resistance of the refrigerant. The thermal resistance of R_{t2} combines the convective thermal resistance of the refrigerant, the conductive thermal resistance of the heat pipes, and the convective and conductive thermal resistance of PCM. The thermal capacity C_t represents the thermal capacity for cold storage.

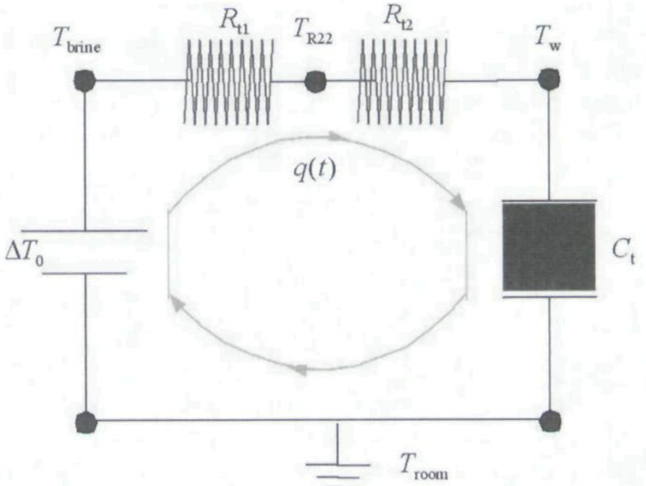

Figure 2 *Modified RC thermal circuit model applied to cold storage system under charge mode.*

A classical RC thermal circuit model is limited to one-phase material with constant thermal resistance and capacity during heat transfer and is not suitable for phase-change material (PCM) with latent heat storage in energy engineering, for example, ice storage air-conditioning. In order to apply the classical RC model to the cold storage system, the thermal capacity needs to be modified as the summation of sensible and latent heat capacities. It becomes larger with the growth of ice layer thickness. The thermal capacity C_t for cold storage is defined as the summation of sensible heat capacity and latent heat capacity, which is

$$C_t \equiv M_l C_{pl} u(T - T_{ls}) + M_{ls} h_{ls} \delta(T - T_{ls}) + M_s C_{ps} u(T_{ls} - T) . \tag{1}$$

In Equation 1 M_l is the mass of water, M_{ls} is the solidification rate, and M_s is the mass of ice. The solidification rate should be equal to the time rate of the mass difference between water and ice. The first and third terms on the RHS of Equation 1 are the thermal capacity of water and ice in terms of sensible heat storage. A step function multiplied by specific heat is utilized to define the thermal capacity for PCM temperature above or below the phase-change temperature T_{ls}. The second term represents the thermal capacity of latent heat storage. Delta function multiplied by latent heat is utilized to define the thermal capacity for PCM temperature equal to the phase-change temperature.

The temperature potential in the modified RC thermal circuit similar to Kirchhoff's voltage law is shown as

$$\Delta T_0 = qR_{t1} + qR_{t2} + \frac{\int q dt}{C_t}, \tag{2}$$

where q is the cold storage rate. In the above equation, the first term represents the temperature difference between the low-temperature flowing fluid and refrigerant, the second term is the temperature difference of the refrigerant and PCM, and the third indicates the temperature potential between PCM and environment temperature.

The cold storage rate is calculated as the solution of the differential formula:

$$q = q_0 e^{\frac{-t}{R_t C_t}} \tag{3}$$

In Equation 3, the characteristic time $r_t = R_t C_t$ indicates the decay degree of the cold storage rate. The cold storage rate declines faster as the characteristic time decreases. The refrigerant and PCM temperatures are calculated from Equation 2, and they are

$$T_{R22} = T_{room} - \Delta T_0 + q \times R_{t1} \tag{4}$$

$$T_w = T_{room} - \Delta T_0 + q \times R_t \tag{5}$$

In Equation 4, the thermal resistance R_{t1} is assumed to be a constant. It represents only the conductive thermal resistance of the low-temperature heat exchanger, which is far larger than other thermal resistance. The overall thermal resistance R_t is dominated by conductive thermal resistance R_c, which can be calculated from

$$R_c(t) = \frac{\ln(r_o(t)/r_i)}{2\pi k_c L N} \tag{6}$$

$$r_o = r_i \times \sqrt{1 + \frac{M_s}{N \rho_c \pi r_i^2 L}} \tag{7}$$

Cumulative cold storage Q_{ss} similar to cumulative electrical charge is calculated with the summation of cumulative cold storage during a short sampling time Q_{si}:

$$Q_{si} = \int_0^{\Delta t} q dt = q_0 R_t C_t \left[1 - e^{\frac{-\Delta t}{R_t C_t}}\right] \tag{8}$$

$$Q_{ss} = \sum Q_{si} \tag{9}$$

Coupling the modified RC thermal circuit model of Equations 1, 3, 5, and 8-9 and the conductive thermal resistance of Equations 6-7, all parameters are numerically solved with iteration during sample time interval.

EXPERIMENTAL INVESTIGATIONS

An experimental cold storage system with multi-loop heat pipes was designed according to the size of energy storage tank, 30 × 12 × 65 cm, with the ratio of distance between heat pipes and outer diameter of heat pipe W/D = 2, as shown in Figure 3. A liftable panel was arranged at the top of the tank for charging the PCM. At the front of the tank, tempered glass replaces the stainless steel material to allow the observation of ice layer formation. The entire storage tank was wrapped in 20 mm thickness of low thermal conductivity material to reduce heat loss. The multi-loop heat pipes were made of ASTM B88 L-type copper tubes, where the tubes in the storage tank were three rows of alternately arranged tubes with different W/D. The low-tempera-

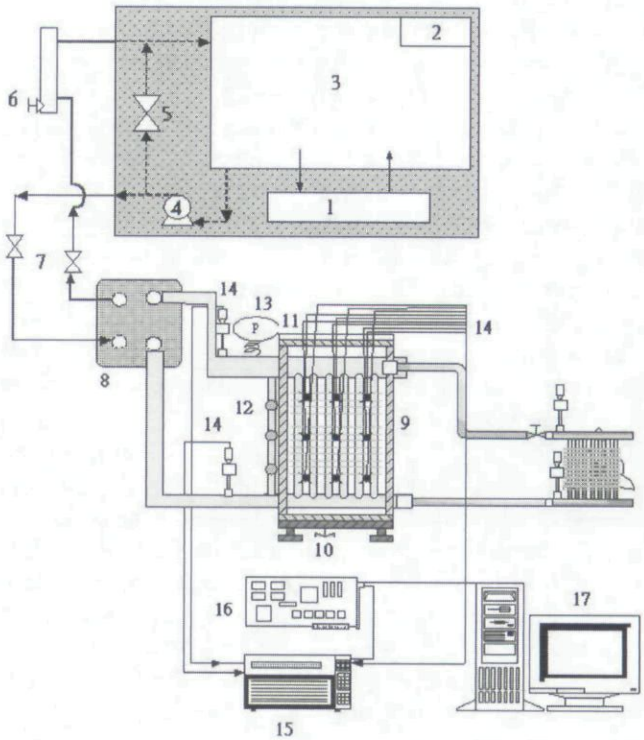

Figure 3 *Experimental systems, equipment, and measurement systems: 1 chiller, 2 resistance heater, 3 thermostatic tank, 4 brine cycle pump, 5 bypass valve, 6 purgmeter, 7 valve, 8 heat exchanger, 9 energy storage tank, 10 drain hole, 11 upper cover, 12 liquid level indicator, 13 pressure gauge, 14 thermocouples, 15 data recorder, 16 GPIB card, 17 monitor.*

ture heat exchanger employed was the high-efficiency plate type. The PCM in the storage tank was pure water and R-22 was used as the working fluid in the multi-loop heat pipes.

As shown in Figure 3, the experimental equipment was set up mainly for constant production of brine through the low-temperature chiller in a large thermostatic tank. The brine tank was 71 × 130 × 66 cm with maximum capacity of 600 L. A 10 HP chiller was equipped with a 2 HP pump to circulate the brine to the low-temperature heat exchanger. Another 1/2 HP pump was installed inside the thermostatic tank to recirculate the brine and achieve a uniform temperature. Brine water in the thermostatic tank was 28% ethene glycol solution. The temperature range of the thermostatic tank was −20~80°C with accuracy of ±0.3°C.

The measurement system consists of T-type thermocouples, flow meter, pressure gauge, electronic scale, liquid level indicator, digital camera, and height indicator. The physical quantities measured include the brine inlet/outlet temperature of the heat exchanger, brine cycle flow rate, temperature and pressure of refrigerant in and out of the ice storage tank, variation of water temperature in the ice storage tank, liquid refrigerant level in the ice storage tank, change of water level inside the energy storage chamber, and the surrounding temperature.

The thermal energy rate supplied to the thermal battery $\dot{Q}_i$ is computed from the temperature difference at the inlet and outlet of the brine flowing through the low-temperature heat exchanger:

$$\dot{Q}_i = \dot{m}C_{pb}[T_{out}(t) - T_{in}(t)] \tag{10}$$

The cold storage rate $\dot{Q}_o$ is the rate of water latent and sensible heat storage in the energy storage chamber during a period of time. The former may be calculated from the thickness of the ice layer by observing the variation of water level in the energy storage chamber; the latter may be obtained based on measurements of temperatures of water in the energy storage chamber. The water temperature at various points of the energy storing chamber is T_w, and the average temperature of measurement points is $\Sigma T_w/m$, where m is the number of measurement points in the energy storage chamber. Thus, the cold storage rate of the cold storage system, $\dot{Q}_o$, can be expressed as

$$\begin{aligned} Q_o = {} & M_w C_{pw}[\sum T_w(t)/m - \sum T_w(t-\Delta t)/m] \\ & /\Delta t + \rho_w \rho_c/(\rho_w - \rho_c) H A_{tank} h_{ls}/\Delta t\,. \end{aligned} \tag{11}$$

Therefore, the cumulative cold storage quantity in these experiments Q_s is the summation of cold storage rates for each time interval in the cold storage system with multi-loop heat pipes:

$$Q_s = \int_0^t \dot{Q}_o dt = \sum_0^{t/\Delta t} \dot{Q}_o \Delta t \tag{12}$$

RESULTS AND DISCUSSION

To examine the effect of the fill level of liquid refrigerant in the energy storage tank on the cold storage phenomenon of the cold storage system with multi-loop heat pipes, system performance was analyzed at the liquid levels of 30 cm, 40 cm, and 50 cm, respectively. The effects of fill level on liquid temperature at the lower header and vapor temperatures at the upper location are quantitatively demonstrated in Figures 4-6. It is seen that under all operating conditions, the experimental and predicted temperature of both liquid and vapor drops steeply at the beginning, but subsequent experimental and predicted temperature profiles of liquid and vapor variation differ with different fill levels. When the fill level is at 50 cm, as shown in Figure 4, both the experimental and predicted liquid temperatures rise from subcooling to operation temperature at the time of 2,100 seconds, while the vapor temperature always keeps saturated. Since the cold storage rate at the high fill level for the entire cold storage process is large enough for nucleate boiling and since the heat transfer mechanism during the entire cold storage process is dominated by nucleate boiling, predicted temperature profiles of liquid and vapor show good agreement with experimental measurements.

The vapor and liquid temperature histories for fill levels at 30 and 40 cm are shown in Figures 5 and 6. Experimental data of vapor temperature at the time of 800 and 1,300 seconds for the fill level of 30 and 40 cm rises from saturation to superheated vapor. Oscillation of experimental vapor temperature occurs. In the case of 30 cm fill level, it can be observed that the experimental vapor temperature becomes more violent than that of 40 cm fill level. However, both experimental liquid temperatures in Figures 5 and 6 always keep saturated. Both predicted vapor and liquid temperature profiles for the fill level of 30 and 40 cm still keep constant. It is obvious that the heat transfer mechanisms at the 30 and 40 cm fill levels change from nucleate boiling to geyser boiling. Under geyser boiling conditions, liquid is periodically propelled from the evaporator to the condenser section with a significant velocity. This liquid with fast movement results in oscillating heat transfer and produces a strange sound in the thermosiphon. In an extreme case, it may damage the container wall.

The reason for these discrepancies between the experimental measurements and the present model is that the application of the present theoretical model is suitable only for the nucleate boiling stage. We use the nucleate boiling correlation to determine the convective thermal resistances of refrigerant among R_{t1} and R_{t2} in Figure 2. Nucleate boiling with an excellent heat transfer coefficient makes the convective thermal resistances of refrigerant far smaller than the conductive thermal resistance of PCM. Thus, the thermal resistances of nucleate boiling can be neglected. The overall thermal resistance in the cold storage system is determined only by conductive thermal resistance. With the lapse of time during the charge process, the heat transfer mechanism is changed from nucleate boiling to geyser boiling. The geyser boiling with smaller heat

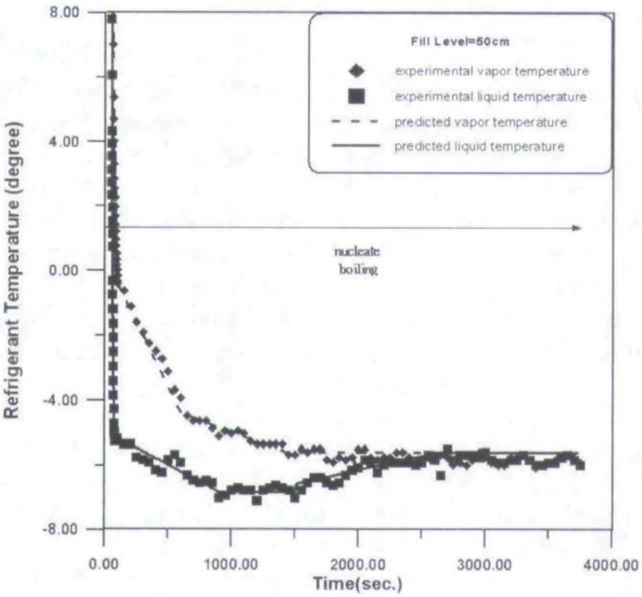

Figure 4 Liquid and vapor temperature histories for high fill level, 50 cm.

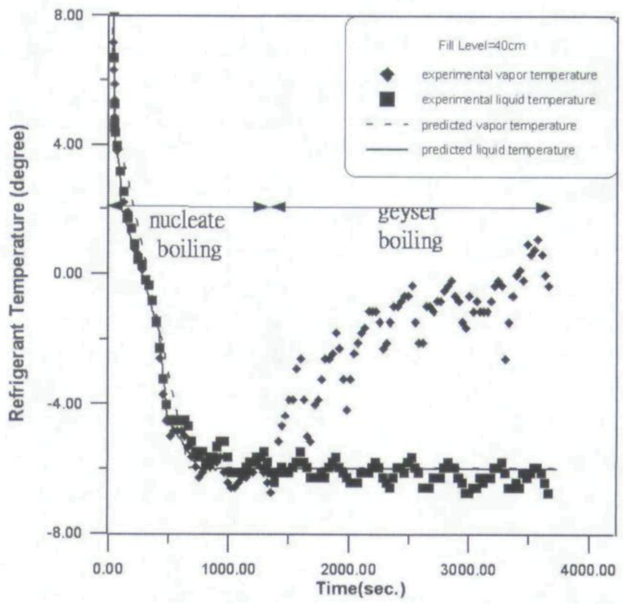

Figure 5 Liquid and vapor temperature histories for medium fill level, 40 cm.

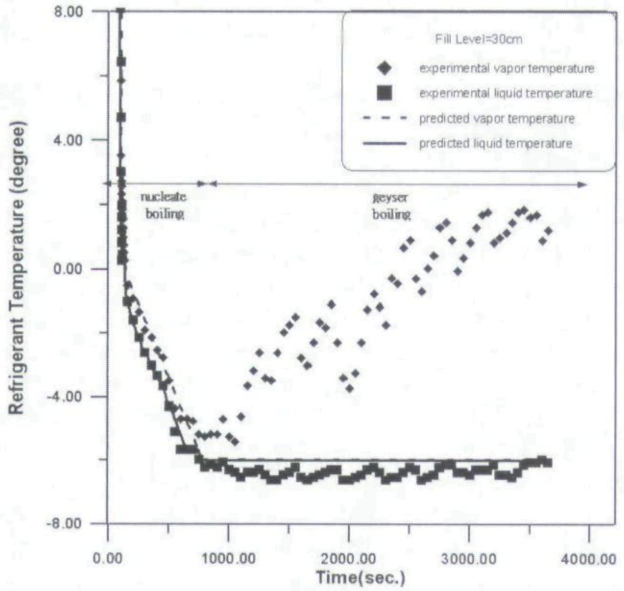

Figure 6 Liquid and vapor temperature histories for low fill level, 30 cm.

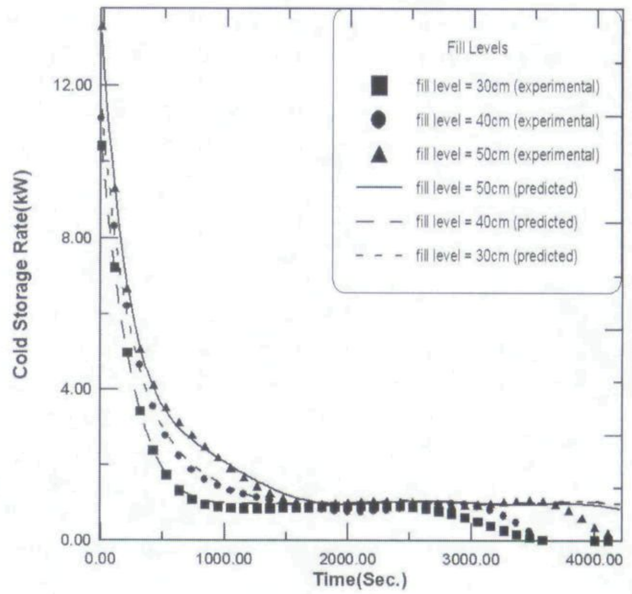

Figure 7 Cold storage rate for different fill levels.

transfer coefficient means the convective thermal resistances of refrigerant cannot be neglected in comparison with the conductive thermal resistance of PCM. In the present model we neglect the thermal resistance of geyser boiling, which will cause a large discrepancy with experimental data.

Figure 7 depicts the experimental and predicted cold storage rates at the fill levels of 50, 40, and 30 cm, respectively. It is observed that with the lapse of time, the experimental and predicted cold storage rates of different fill levels decrease rapidly, then gradually become flat. But subsequent experimental and predicted cold storage rate profiles differ. As the ice layer thickness increases to a certain value (approximate 2.0 cm), the cold storage rate becomes stable. The turning time, which is defined as the time when the ice layer reaches a certain value (approximately 2.0 cm), occurs at 800, 1,300, and 1,700 seconds for the fill levels of 30, 40, and 50 cm, respectively. It also can be seen from this figure that the higher the fill level, the faster the cold storage rate. Experimental cold storage rates decrease to zero earlier with the lower fill level, but the predicted cold storage rates at different fill levels in the last stage nearly keep constant. This is because the present model considers that nucleate boiling is the only heat transfer mechanism during the cold storage process. As seen in Equation 6, the thermal resistance of nucleate boiling is smaller than that of the ice layer. The assumption in the theoretical cold storage model ignores the thermal resistance of working fluid for nucleate boiling. However, the geyser boiling occurred after nucleate boiling during the cold storage process at medium and low fill levels. The thermal resistance of the working fluid for geyser boiling has the same order of magnitude as that of ice layer conduction and cannot be neglected. In the present model we apply the heat transfer mechanism of nucleate boiling instead of the heat transfer mechanism of geyser boiling. It will result in underprediction of the total thermal resistance, which leads to overestimation of the cold storage rate in the cold storage stage of geyser boiling.

Figure 8 illustrates experimental and predicted cumulative cold storage quantity versus time under different fill levels. It is seen from the figure that at any fill level, there exists an initial stage with steeply curved slope and a secondary stage flattened gradually, both experimentally and numerically. The turning points occur at 800, 1,300, and 1,700 seconds for the fill levels of 30, 40, and 50 cm, respectively. When cold storage ends at 3,680 seconds, the final experimental cold storage quantities for the fill level of 30, 40, and 50 cm are 4,102.34 KJ, 4,935.47 KJ, and 6,409.31 KJ, respectively, and numerical predictions are 5,165.71 KJ, 5,744.61 KJ, and 6,693.33 KJ. The cumulative cold storage quantity and turning point time are larger for the case of higher fill level. The turning point times at all fill levels are the same as those of the cold storage rate. Besides, it is found that the predicted cumulative cold storage quantity of all fill levels is larger than the experimental results, and error percentage of predicted result is larger for the case of lower fill level. This is because the larger period of geyser boiling occurred during the cold storage process as the fill level is lower, and the predicted cold storage rate is larger than the experimental cold storage rate for the geyser boiling stage. As a result, the final predicted cumulative cold storage quantity is larger with lower liquid fill level due to the longer geyser boiling period of the cold storage process.

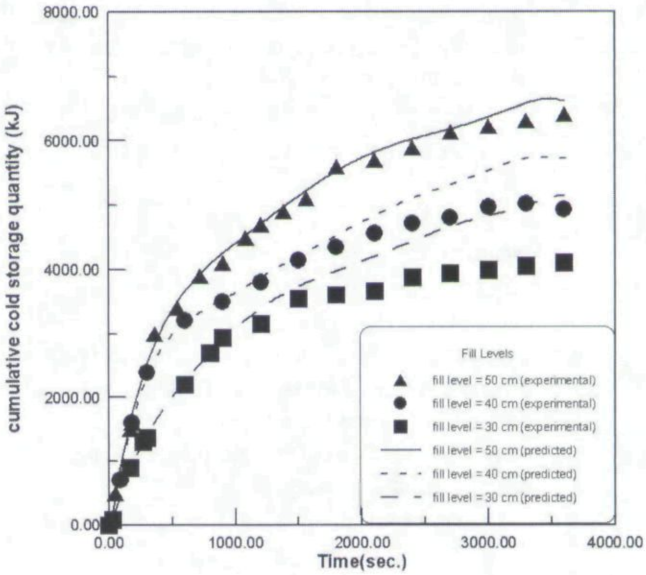

Figure 8 *Cumulative cold storage quantity for different fill levels.*

CONCLUSIONS

Regarding the effect of fill level on the cold storage system with multi-loop heat pipes, if the fill level is higher, the cold storage rate is larger. The time period for the heat transfer mechanism of nucleate boiling is longer, and the cumulative cold storage quantity is larger. Since the thermal resistance assumption of refrigerant convection in the modified RC thermal circuit model is only appropriate for the nucleate boiling stage, not for the geyser boiling, the prediction of cold storage quantity from the present model is overestimated for case of geyser boiling that exists during the cold storage process.

Future investigations will examine the effect of other parameters, such as different W/D ratios, different heat pipe tilt angles, and other phase-change materials, in the cold storage system with multi-loop heat pipes. Theoretical investigation might also modify the present model to account for geyser boiling.

NOMENCLATURE

A	=	area (m)
C_t	=	thermal capacity (J/K)
C_{pl}	=	specific heat of liquid PCM (J/kg·K)
C_{ps}	=	specific heat of solid PCM (J/kg·K)
C_{pb}	=	specific heat of brine (J/kg·K)
C_{pw}	=	specific heat of water (J/kg·K)
h_{ls}	=	solidification latent heat of water (KJ/kg)
H	=	water height difference during the sample interval (m)
k	=	conductivity (J·L/K)
L	=	length of heat pipe (m)
M	=	PCM weight (kg)
M_w	=	weight of water in the cold storage tank (kg)
M_i	=	solidification weight in one cell of the cold storage tank (kg)

M_{total} = total weight in one cell of the cold storage tank (kg)
m = numbers of measurement points inside the cold storage tank (dimensionless)
$\dot{m}$ = mass flow rate of the brine (kg/s)
N = number of cells inside the cold storage tank (dimensionless)
T = temperature (K)
t = measurement time (s)
u = step function (dimensionless)
$\dot{Q}_i$ = experimental thermal energy rate supplied to cold storage system with multi-loop heat pipes (w)
$\dot{Q}_o$ = experimental cold storage rate of cold storage system with multi-loop heat pipes (w)
Q_{si} = cumulative cold storage quantity of prediction during the sample interval (kJ)
Q_{ss} = cumulative cold storage quantity of prediction during the sample interval (kJ)
q = predicted cold storage rate during the sample interval (w)
q_0 = predicted cold storage rate at the beginning of the sample interval (w)
R_1 = thermal resistance composed of convection of low-temperature flowing fluid, conduction of low-temperature heat exchanger, and convective thermal resistance of refrigerant (K/J)
Rt_2 = thermal resistance composed of convection of refrigerant, conduction of heat pipes, and convection and conduction of PCM (K/J)
R_t = overall thermal resistance (K/J)
r_o = outer radius of ice layer (m)
r_i = inner radius of ice layer (m)
ΔT_0 = source temperature potential (K)
Δt = sample time interval (s)

Greek Symbols

ρ = density (kg/m^3)
δ = delta function (dimensionless)

Subscripts

b = brine
c = ice
in = inlet brine
l = liquid state
ls = phase change from liquid state to solid state
out = outlet brine
s = solid state
$tank$ = tank
w = water

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