ON TEMPERATURE DISTRIBUTION AND HEAT TRANSFER CHARACTERISTICS OF A PHASE CHANGE MATERIAL IN AN ANNULUS

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الملخص

تقدم هذه الورقة نموذجا رياضيا لانتقال الحرارة العابر ثنائي الأبعاد لمادة متغيرة الطور. تتكون المنظومة من أنبوب نحاسي محاط بغلاف محوري مكونا فراغا حلقيا.تم تعبئة الفراغ الحلقي بالمادة المتغيرة الطور التي تتبادل الحرارة مع المائع الثنائي الطور المتدفق في الأنبوب. استخدمت طريقة الحجوم المحدودة لنمذجة المسألة عدديا.تمتاز هذه الطريقة بإمكانية الأخذ في الاعتبار الحرارة الكامنة عن طريق تعريف الإنثالبي الكلية، بينما تمت نمذجة التدفق ثنائي الطور من خلال معادلة الطاقة ومعادلة تجريبية لمعامل انتقال الحرارة. أجريت دراسة بارامترية لمعرفة تأثير مجموعة من موائع التبريد المختلفة الخواص، ودرجة حرارة التشبع، وزمن التفريغ، ومعدل التدفق، على التوزيع الحراري داخل المادة المتغيرة الطور وأيضا على كمية الطاقة الحرارية المنتقلة. تم اختيار ثلاثة أنواع من موائع التبريد وهي، ثاني أكسيد الكربون (R744) والفريون 134a والفريون 407C. بيّنت النتائج أن استخدام ثاني أكسيد الكربون صاحبه أفضلية في خصائص انتقال الحرارة وقصر في زمن التجميد مقارنة ببقية الموائع المختارة.

ABSTRACT

This paper presents a numerical model of a two–dimensional axi-symmetrical transient controlled heat conduction of Phase Change Material (PCM). The system consists of a copper tube surrounded by a coaxial cylinder which forms an annular gap around the tube. The annular is filled with PCM while the two-phase fluid undergoes heat exchange through a tube wall. The finite volume approach for numerical modeling of phase change problem is employed. The essential feature of this approach is that the evolution of latent heat is accounted for by the definition of total enthalpy (H), while the two-phase fluid flows is modeled via a single energy equation with empirical correlation for the heat transfer coefficient. A parametric study is carried out to find out influence of different kinds of refrigerants, refrigerant saturation temperature, discharging time, and refrigerant mass flow on the temperature distribution inside the PCM, and the amount of heat released by PCM. Three refrigerants with different properties are selected for the present study; those are, carbon dioxide (R744), Freon 134a (R134a) and Freon 407C (R407C). The results show that the refrigerant R744 is characterized with the best heat transfer characteristics, and the lowest solidification time of PCM.

KEYWORDS: Phase Change Material; Enthalpy; Finite Volume; Saturation temperature; Refrigerants.

INTRODUCTION

There are many practical heat transfer problems involve a change of phase of a material due to energy absorption or release (i.e., melting or freezing of the phase change material). Examples would include solar energy storage for night heating, and the formation of ice for space cooling [1]. Several researchers have investigated the behavior of phase change materials analytically and numerically, the analytical solutions are limited by their mathematical complexity. The solution of a phase change problem consists of a transient heat conduction problem, in up to three space dimensions, coupled with a convection problem, or in the case of the present work, coupled with two phase flow. Springer [2] numerically solved the problem of freezing or melting of cylinders for homogeneous phase change materials for a given temperature distribution along the inner wall. Cao and Faghri [3] performed a numerical analysis of freezing in a phase change material with a square cross-section, the analysis included the effect of natural convection but assumed constant, uniform wall temperature. Charach, Keizrnan, and Sokolov [4] studied the problem of axisymmetric freezing around a coolant-carrying tube which provided a uniform, constant heat transfer coefficient. Hasan [5] also employed a constant heat transfer coefficient from the coolant when computing the speed of the radial phase transition front around a vertical and horizontal tube.

The present work is oriented to model the two-dimensional, time- dependent, freezing of PCM (water initially at an arbitrary temperature of 280 K). The PCM is in an annulus that surrounds a metal tube through which a two phase fluid is flowing. The problem is modeled as a conjugate problem of conduction through the PCM with forced convection at one of the boundaries.

SYSTEM DESCRIPTION

The Thermal Energy unit being investigated consists of a copper tube surrounded by a coaxial cylinder which forms an annular gap around the tube as shown in Figure (1). The gap is filled with Phase Change Material (water). The fluid is flowing through the copper tube also undergoes a phase change. In the present study the PCM is initially in the liquid phase (at 280 K and atmospheric pressure), and the refrigerant enters the tube as a saturated liquid, and changes to a saturated vapor as it absorbs energy from the PCM through the copper wall.

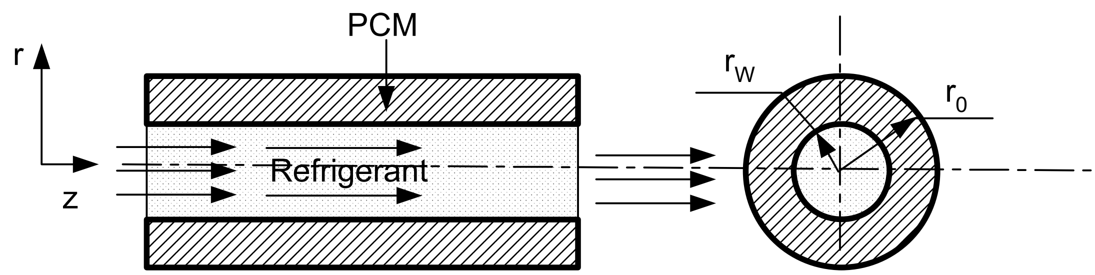

Figure 1: Thermal energy unit and its cross sectional view

MATHEMATICAL MODEL

For the present analysis, the following assumptions are made:

- The annular is insulated at the outer radius r_0, and at both ends.

- The PCM is homogeneous and isotropic water.
- The problem is a two-dimensional, transient heat conduction problem in (r,z,t) domain, with conjugate forced convection of a two-phase fluid at its inner boundary.
- The thermo-physical properties of the PCM are independent of temperature but can be different in the solid and liquid phases.
- Due to its complexity, the effect of natural convection in the liquid PCM is not taken into consideration in the present work.

The governing differential equation, initial condition, and boundary conditions are:

Differential equation

$$\rho_P \frac{\partial H}{\partial t} = k_P \left[\frac{\partial^2 T_P}{\partial r^2} + \frac{1}{r}\frac{\partial T_P}{\partial r} + \frac{\partial^2 T_P}{\partial z^2} \right] \text{in } r_w \le r \le r_0 \text{ and } 0 \le z \le L,\ t > 0 \tag{1}$$

Where:

ρ_P is the PCM density.

H is the enthalpy.

T_P is the PCM temperature.

t is the time.

k_P is the PCM thermal conductivity.

r and z denote the radial and axial coordinates respectively.

r_w and r_0 refer to the inner wall and outer wall radii respectively .

L is the axial length of the thermal unit.

Boundary conditions

$$\frac{\partial T_P}{\partial r} = 0 \text{ at } r = r_0, t > 0 \tag{2}$$

$$\frac{\partial T_P}{\partial z} = 0 \text{ at } z = 0 \text{ and } z = L, t > 0 \tag{3}$$

$$h(T_f - T_W) = -k_P \frac{\partial T_P}{\partial r} \text{ at } r = r_W, t > 0 \tag{4}$$

Where T_f and T_W are the refrigerant and wall temperatures, respectively.

Initial condition

$$T_P(r,z) = T_0 \text{ in } r_W \le r \le r_0 \text{ and } 0 \le z \le L, t = 0 \tag{5}$$

In the liquid phase: $T_P(r,z) > T_m$

$$k_P = k_L, \rho_P = \rho_L, \text{and } H(r,z) = C_{PL}[T_P(r,z) - T_m] \tag{6-a}$$

In the solid phase: $T_P(r,z) < T_m$

$$k_P = k_S, \rho_P = \rho_S, \text{and } H(r,z) = C_{PS}[T_P(r,z) - T_m] - LH \tag{6-b}$$

In the mixture phase (solid and liquid): $T_P(r,z) = T_m$

$$k_P = k_L - \frac{H}{LH}(K_s - k_L), \text{and } \rho_P = \rho_L - \frac{H}{LH}(\rho_s - \rho_L) \tag{6-c}$$

Where the subscripts P, S, L, and m refer to phase change, solid phase, liquid phase, and melting respectively. While C and LH refer to specific heat and latent heat respectively. The heat transfer coefficient h for the refrigerant side is based on Gungor and Winterton's correlation [6]:

$$h = h_l\left[1 + 3000B_0^{0.86} + 1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_l}{\rho_v}\right)^{0.41}\right] \quad (7)$$

Where the subscripts l and v refer to the liquid and vapor phase in the refrigerant side respectively, while x is the quality.
The heat transfer coefficient for the single-phase, liquid heat transfer coefficient, h_l is computed from the well-known Dittus-Boelter equation [6]:

$$h_l = 0.023\,Re_l^{0.8}\,Pr_l^{m}\,\frac{k_l}{D} \quad (8)$$

Here:
m equals to 0.4 for heating and 0.3 for cooling

$$Re_l = G(1-x)D/\mu_l \quad (9)$$

$$Pr_l = \frac{\mu_l \times CP_l}{k_l} \quad (10)$$

$$B_0 = \frac{q}{h_{fg}G} \quad (11)$$

Where:
Re_l, Pr_l are the liquid Reynolds number and Prandtl number respectively.
k_l, and μ_l are the thermal conductivity, and viscosity of the liquid respectively .
D is the inner diameter of the copper tube.
G is the mass flux through the tube.
B_0 is the dimensionless boiling number.
q is the heat flux and h_{fg} is the heat of vaporization.
The heat transfer Q, between the refrigerant and the PCM is determined from the energy balance, where:

$$Q = \frac{T_p - T_f}{R_{in} + R_c + R_{ext}} \quad (12)$$

Here:
R_{in} is the thermal resistance of the refrigerant due to heat convection.
R_c is the thermal resistance due to heat conduction through the tube thickness.
R_{ext} is the thermal resistance due to heat conduction through the PCM

DISCRITIZATION EQUATIONS

A finite-volume, enthalpy method is applied to the heat conduction problem [4]. The PCM in the annular gap is divided into equal lengths (dz) and layers of equal thickness (dr). In the finite-volume method, the temperature at the center (dz/2, dr/2) of each lump is considered to be uniform, at each instant of time [5]. The principle of conservation of energy is applied at each boundary of each finite annular volume. A finite volume approximation is applied to a control volume of the PCM, as shown in Figure (2).

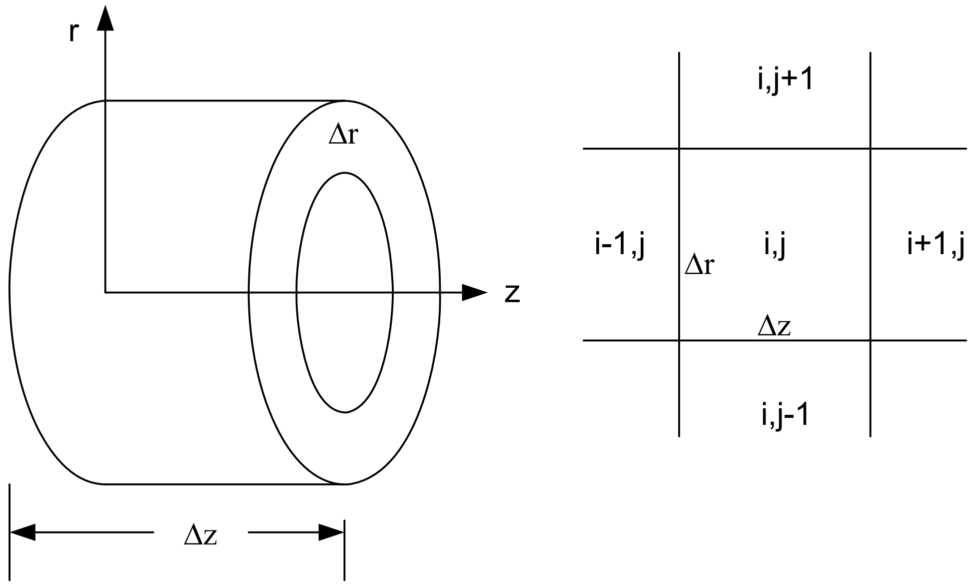

Figure 2: Finit volume

The Enthalpy Method is applied to the governing equation (1).

$$\rho_P V_{i,j} \frac{\partial H_{i,j}}{\partial t} = k_P A_j \left.\frac{\partial T}{\partial r}\right|_j - k_P A_{j-1} \left.\frac{\partial T}{\partial r}\right|_{j-1} - k_P A_{i-1} \frac{\partial T}{\partial z_{i-1}} + k_P A_i \left.\frac{\partial T}{\partial z}\right|_i \tag{13-a}$$

where,

$$A_j = 2\pi r_j \Delta z \tag{13-b}$$

$$A_{j-1} = 2\pi r_{j-1} \Delta z \tag{13-c}$$

$$A_i = A_{i-1} = \pi\left(r_j^2 - r_{j-1}^2\right) \tag{13-d}$$

$$V_{i,j} = \pi\left(r_j^2 - r_{j-1}^2\right)\Delta z \tag{13-e}$$

A and V refer to the surface area and the volume of the finite annular volume respectively. The separate contributions of energy applied to the finite volume approximation are:

Energy through the left boundary (q_{left}):

$$q_{left} = -\frac{\left(k_{i-1,j} + k_{i,j}\right)}{2} A_{i-1} \frac{\left(T_{i-1,j} - T_{i,j}\right)}{\Delta z} \tag{14}$$

Energy through the right boundary (q_{right}):

$$q_{right} = -\frac{\left(k_{i,j} + k_{i+1,j}\right)}{2} A_i \frac{\left(T_{i,j} - T_{i+1,j}\right)}{\Delta z} \tag{15}$$

Energy through the bottom (q_{bottom}):

$$q_{bottom} = -\frac{\left(k_{i,j-1}V_{i,j-1} + k_{i,j}V_{i,j}\right)}{\left(V_{i,j-1}+V_{i,j}\right)} A_{j-1} \frac{\left(T_{i,j-1} - T_{i,j}\right)}{\Delta r} \tag{16}$$

Energy through the top, (q_{top}):

$$q_{top} = -\frac{\left(k_{i,j}V_{i,j} + k_{i,j+1}V_{i,j+1}\right)}{\left(V_{i,j} + V_{i,j+1}\right)} A_{j} \frac{\left(T_{i,j} - T_{i,j+1}\right)}{\Delta r} \tag{17}$$

Substituting equations (14-17) into the governing energy equation and by performing the required differencing to the energy stored term results in:

$$H_{i,j}^{n+1} = H_{i,j}^{n} + \frac{\Delta t}{\rho_{i,j}V_{i,j}} \left(q_{left} + q_{bottom} - q_{right} - q_{top}\right)^{n} \tag{18}$$

Where superscript n+1 is the value at the new time step and superscript n refers to the value at the previous time step. The first row of finite volumes immediately adjacent to the refrigerant tube is treated differently. The copper wall is treated as the j-1 node and q_{bottom} is the conjugate boundary condition between the refrigerant and the PCM. The explicit solution technique is adopted for the present work. The stability requirement is [7]:

$$\frac{k\Delta t}{\rho C(\Delta r)^2} + \frac{k\Delta t}{\rho C(\Delta z)^2} \leq \frac{1}{2} \tag{19}$$

Let $k = k_S$, $\rho = \rho_S$, and $C = C_{PS}$. Simplifying equation (19), we have the following for the time step in our numerical model.

$$\Delta t = \frac{\rho_S C_{PS}}{2k_S} \left(\frac{\Delta r^2 \Delta z^2}{\Delta r^2 + \Delta z^2} \right) \tag{20}$$

RESULTS AND DISCUSSIONS

The present work aims to find the temperature distribution and the heat transfer characteristics for PCM during the process of discharging. The PCM is initially at a uniform temperature of 280 K, and the refrigerant mass flow rate is 15.5 g/s. Different kinds of refrigerants are tested.

Since the temperature distribution and the rate of heat transfer between the PCM and the refrigerant depend on the heat transfer coefficient and are calculated from equations (7) and (8), effect of axial distance on local heat transfer coefficient at discharging time equal to 1270 s, and refrigerant saturation temperature equal to 244 K, is shown in Figure (3). It is found that the refrigerant R744 has the largest heat transfer coefficient, while the refrigerant R134a has the lowest one. The two main parameters that make the difference are Prandtl number and Reynolds number, for instance at the middle of the axial distance, Table (1) shows the values of those numbers. The R744 has the lowest Prandtl number, and also the largest Reynolds number, and hence the largest coefficients of heat transfer.

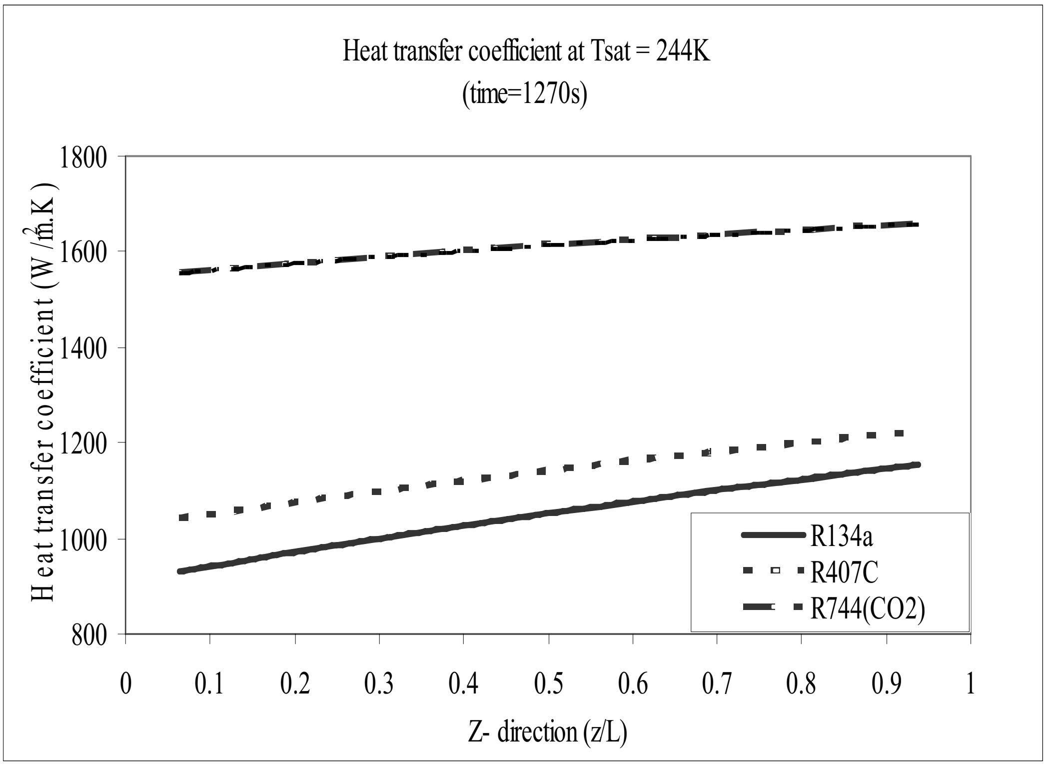

Figure 3: Variation of heat transfer coefficient in the axial direction

Table 1: heat transfer coefficient, Reynolds and Prandtl numbers

	(Z/L)=0.5		
	Pr_l	Re_l	h ($W/m^2.K$)
R134a	4.97	5708.05	1469.6
R407c	3.47	7847.56	1598.7
R744	2.35	14562.3	2275.8

Effect of refrigerant saturation temperature on solidification time is shown in Figure (4). The solidification time here is the minimum time required to solidify all PCM inside the shell. As expected, the shortest solidification time is obtained when using R744 as a refrigerant, and the longer one is obtained when using R134a, this is mainly due to the differences in the heat transfer coefficient values. Since the rate of heat transfer between the PCM and the refrigerant decreases with decreasing in temperature difference, the time of solidification increases with the increase in the saturation temperature, this is also shown in Figure (4).

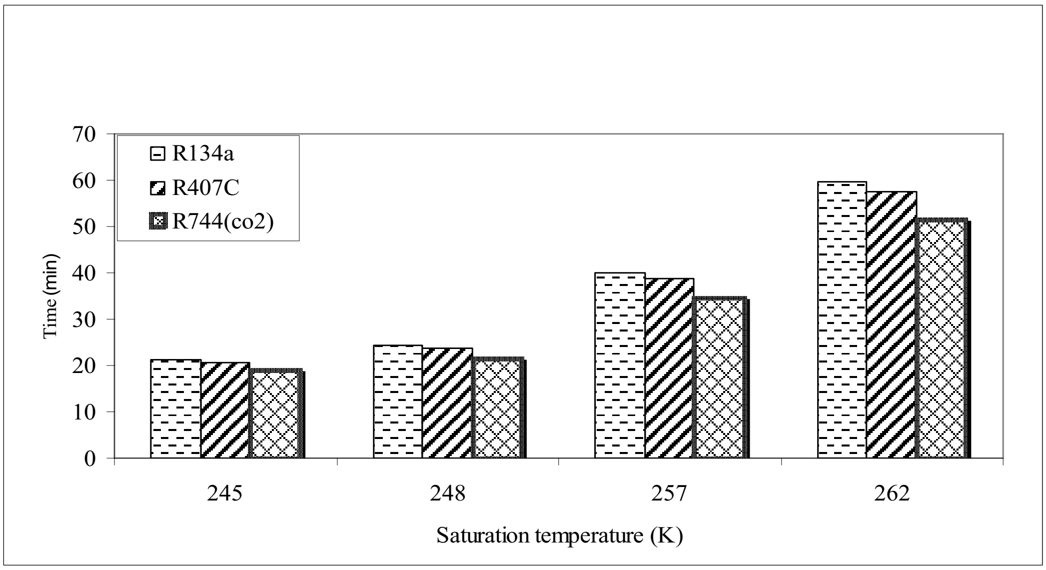

Figure 4: Effect saturation temperature on the time solidification of PCM

Effect of the saturation temperature on the radial temperature distribution inside PCM (solid phase) is shown in Figures (5), (6), and (7) at the middle of the unit. Since the annular pipe is insulated at both ends, the temperature in the axial direction is almost identical. Three saturation temperatures are selected; those are 244 K, 248 K, and 257 K. Since the rate of heat transfer is the lowest when using R134a, the time of discharging is selected as the minimum time required to solidify the PCM when using R134a as a refrigerant, those times insure also the solidification of the PCM when using the other two refrigerants, the selected discharging times are then 1270 s, 1450 s, and 2400 s. The results show that when using R744, the PCM temperature approaching the refrigerant temperature faster than when using the other two refrigerants, this behavior is expected since using R744 implies the highest heat transfer coefficient.

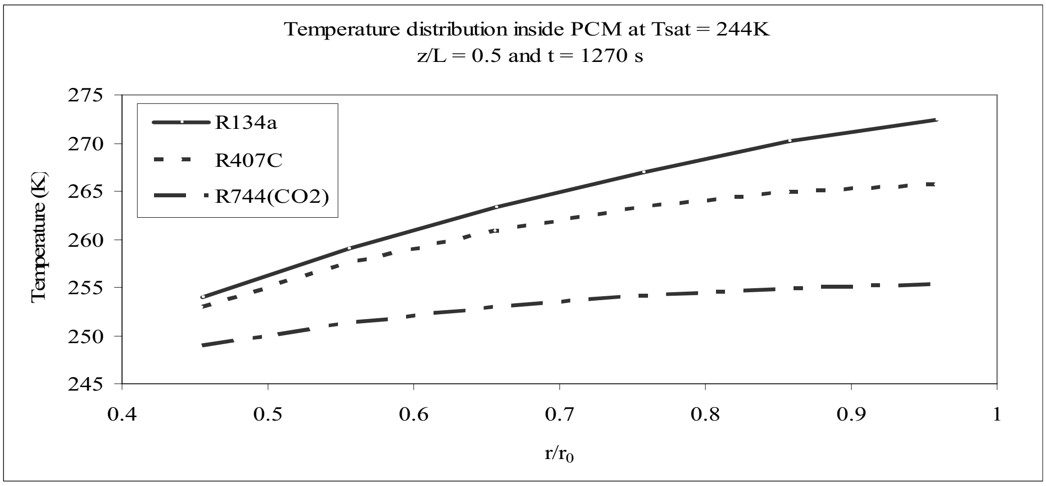

Figure 5: Effect of saturation temperature of Refrigerants flow on the temperature distribution inside the phase change material (Tsat=244K)

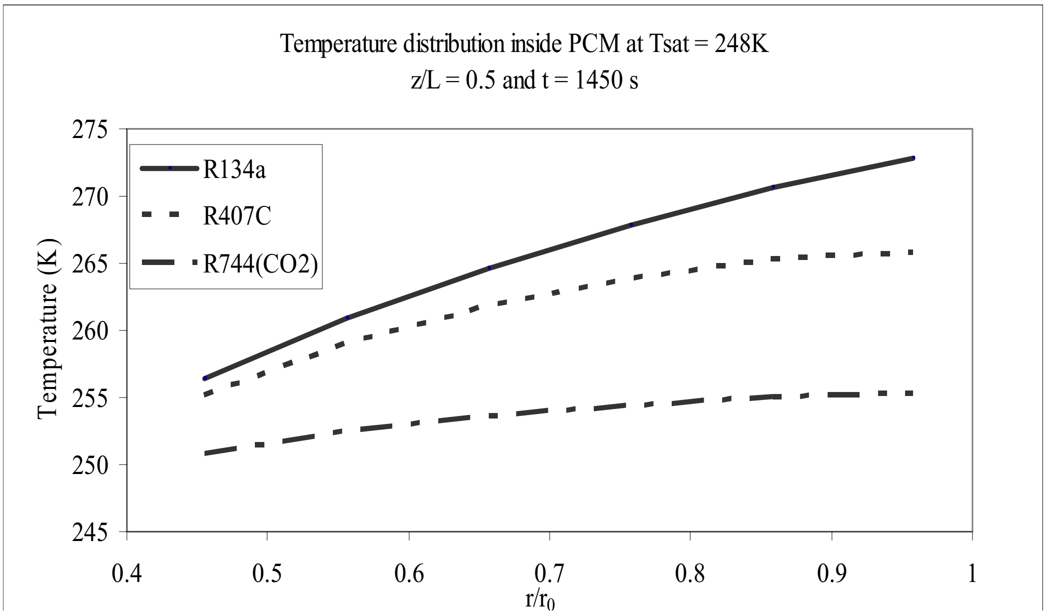

Figure 6: Effect of saturation temperature of Refrigerants flow on the temperature distribution inside the phase change material (Tsat = 248K)

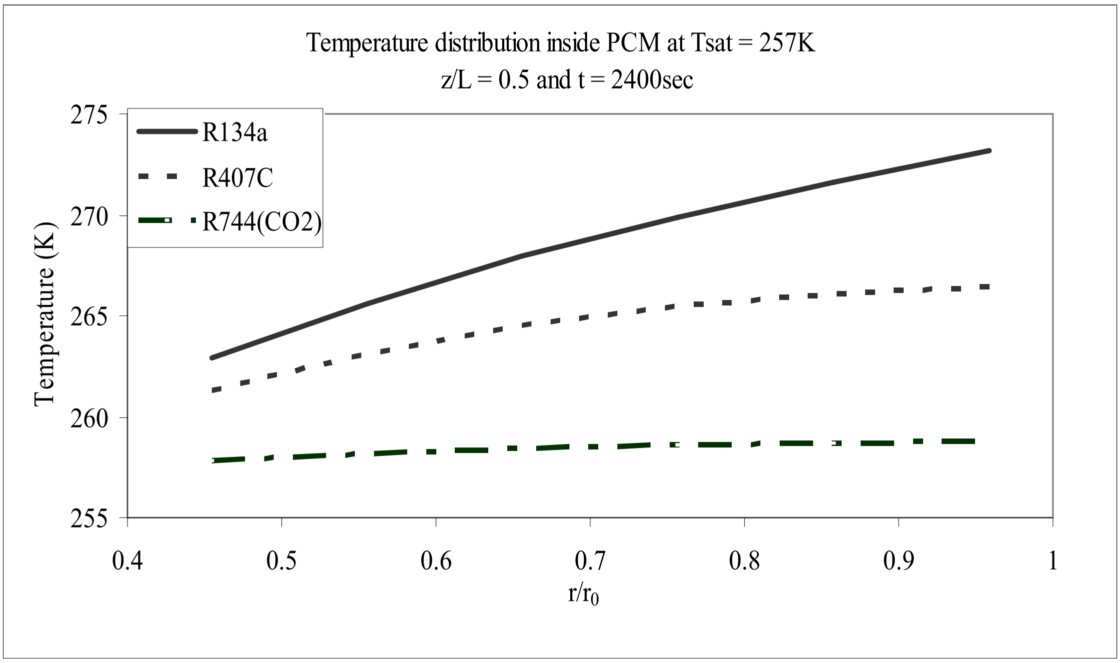

Figure 7: Effect of saturation temperature of Refrigerants flow on the temperature distribution inside the phase change material (Tsat = 257K)

Influence of discharging time on the rate of heat transfer is shown in Figure (8). The benefit of high heat transfer coefficient associated with the use of R744, is offset by the fast decrease in temperature difference between PCM and refrigerant, the result is a small deviation in the rate of heat transfer.

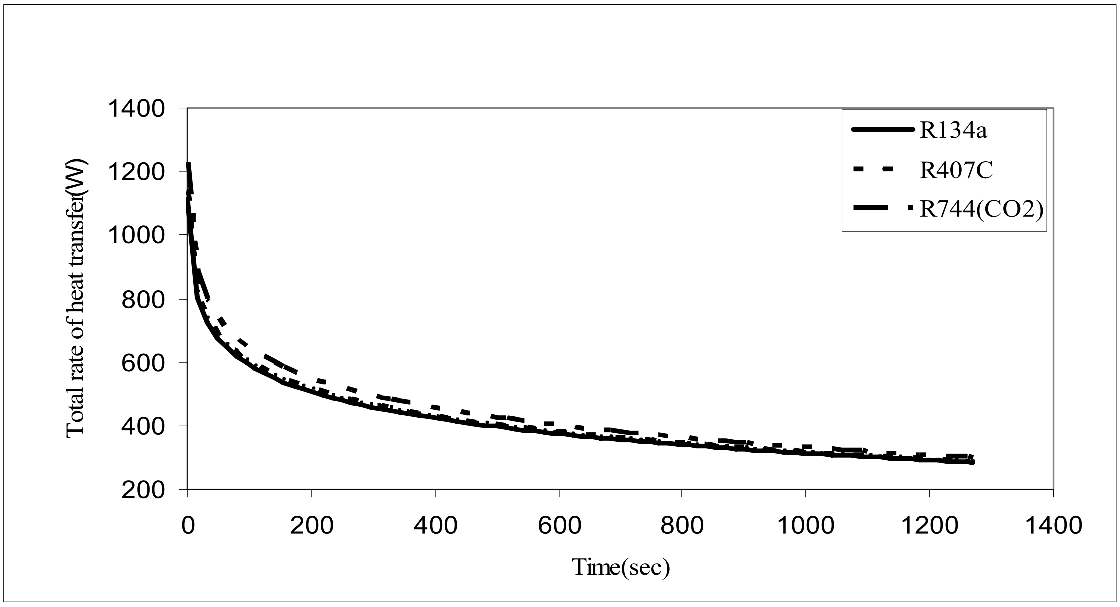

Figure 8: Time effect on the total heat rate absorbed by the refrigerant

Figure (9) shows effect of discharging time on the total amount of energy released from the PCM and absorbed by the refrigerant. A saturation temperature of 244 K, and a mass flow rate of 15.5 g/s are selected for the next discussion. The maximum discharging time is 1270 K, this the minimum time required to solidify the PCM when using any of the three refrigerants. Although, there are no substantial differences in total amount of energy absorbed by different kinds of refrigerants which is in a good agreement with aforementioned reasons, as expected more energy is liberated from the PCM and absorbed by the refrigerant when using R744 as a refrigerant.

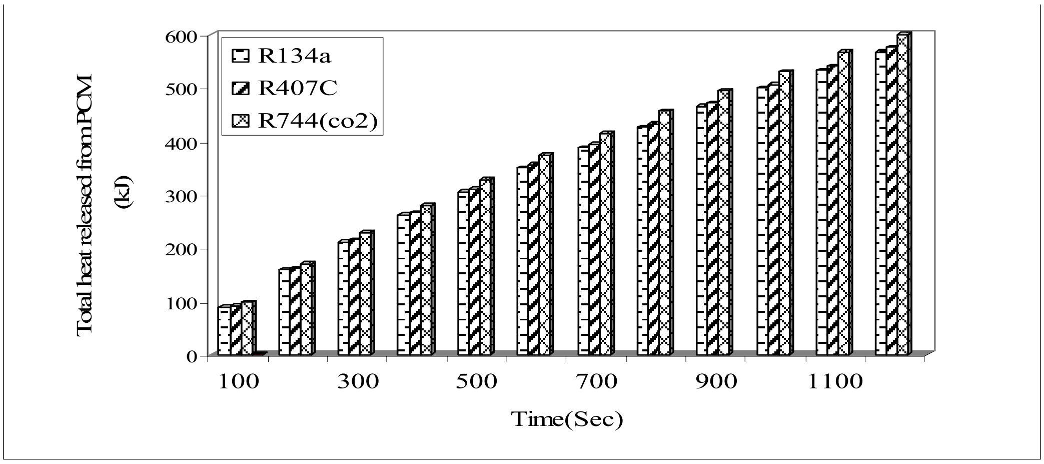

Figure 9: Time effect on the total heat released from PCM

Influence of the saturation temperature on the total amount of energy released by PCM is shown in Figure (10). The amount of energy released by PCM decreases with the increase of the saturation temperature of the refrigerant. This result is expected, since the rate of heat transfer, and hence, the total amount of energy released by PCM depend on the temperature difference between the refrigerant and the PCM. Although there are no substantial differences, still, the R744 posses a higher heat transfer coefficient, and then associated with the higher total amount of energy released by PCM.

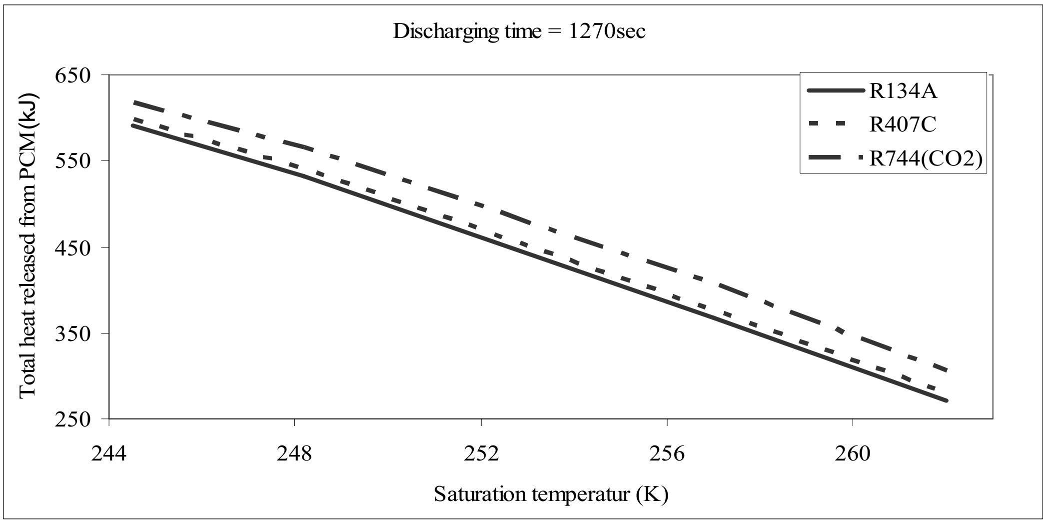

Figure 10: Effect of the saturation temperature of fluid on the total heat transfer released from PCM

The rate of heat transfer and the total amount of energy released by PCM can be enhanced by increasing the mass flow rate of the refrigerant. This is due to the increase of Reynolds and hence the heat transfer coefficient with the increasing mass flow rate. A saturation temperature of 244 K and discharging time of 1270 s, which is the minimum time required to solidify the PCM when using R134a for the selected range of flow rates, are selected for the Figure (11).

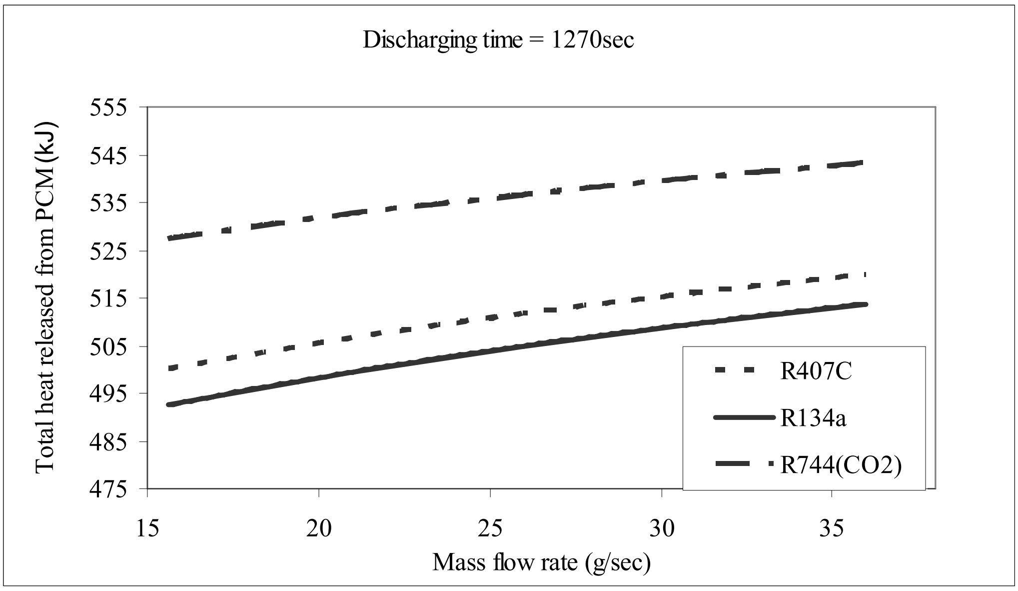

Figure 11: Effect of mass flow rate on total amount of heat released

CONCLUSIONS

- A numerical model has been presented to simulate the transient behavior of two-dimensional (axial and radial), cylindrical, solidification of phase change material (PCM) interacting with a two-phase fluid.
- The phase change process was modeled using the enthalpy method along with the finite volume approach.
- The convective heat transfer of the refrigerant fluid was modeled by existing empirical correlations.
- A series of numerical calculations is employed to find out effect of three types of refrigerants, those are R744, R134a, and R407C.
- Among the three refrigerants, it is found that R744 has the highest heat transfer coefficient, and R134a has the lowest one.
- The minimum time for PCM solidification is associated with the use of R744.
- The radial temperature distribution inside PCM is presented, for different saturation temperatures and different refrigerants.
- The results show that, there are no substantial differences between the rate of heat transfer and the total energy released by PCM associated with the use of the three refrigerants.
- Effect of saturation temperature on total heat released is presented. It is found that the total heat released decreases with the increase of the saturation temperature.
- The total heat released can be enhanced by increasing the refrigerant mass flow rate.

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