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LOW TEMPERATURE APPLICATIONS OF A PHASE CHANGE THERMAL STORAGE SYSTEM PERFORMANCE

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Abstract. *This work presents a simplified numerical model for predicting the transient behaviour of a shell-and-tube thermal storage unit with the pcm on the shell unit and the heat transfer fluid (HTF) circulating inside the tubes. The solution of the system consists of adopting a velocity field of a complete developed flow coupled with the energy equations of the heat transfer fluid (HTF), the pipe wall and the phase change material (pcm). The axial temperature increase experienced by the coolant gives rise to two dimensional freezing around the tubes. The control volume finite difference approach is used to solve the two dimensional equations describing the phase change thermal storage system. The influence of time, Reynolds and Stefan numbers, tube wall materials on the heat storage system thermal performance is presented in terms of temperature and front position distributions.*

Key words: *Ice bank, Thermal performance, Solidification front*

1. INTRODUCTION

Understanding the solid-liquid phase change (melting/solidification) heat transfer characteristics is of fundamental importance in a wide range of naturally occurring processes and engineering applications such as a formation of ice on a lake or river, the freezing of water pipes, metal processing, freezing of food stuffs, thermal control of space craft , and thermal energy storage system.

Solidification numerical techniques modelling can be categorised into two groups; front tracking methods are often restricted to one dimensional problems or those where the solidification front of relatively simple form. Fixed domain methods tend to be more versatile and easier to implement, and therefore have found wider use in solidification modelling. Four fixed domain methods are commonly employed to solve solidification problems (Bounds et al., 1996); these are so-called enthalpy (Voller and Cross (1980, Tamma and Namaburu (1990)), capacitance (Runnels and Carey (1991)), fictitious heat flow (Rolph and Bathe (1984)), and temperature recovery techniques (Chem and Lee (1991). Good review techniques are given by Voller et al. (1990) and Dalhuijsen et al.(1986).

In most of the published works the heat transfer between the phase change material and the heat transfer fluid was calculated using empirical correlations instead of solving the whole problem as one domain. The phase change problems are by nature a transient ones and, for this reason, the heat transfer fluid boundary conditions change with interface progress. Therefore, the temperature field of the heat transfer fluid would never establish steady state regime. For short cylinders and low velocities, the entrance laminar region can dominate the flow along the cylinder length.

In this paper, a transient two dimensional phase change thermal storage system is numerically modelled. For simplicity the HTF is considered to be fully developed and the energy equations of the HTF, pipe wall and pcm are coupled and solved as one domain. First the model is validated by comparison with other results obtained from the literature then the influence of time period, Reynolds and Stefan numbers and wall materials on the system thermal performance is investigated. The results are expressed terms of the temperature and solidification front position distribution.

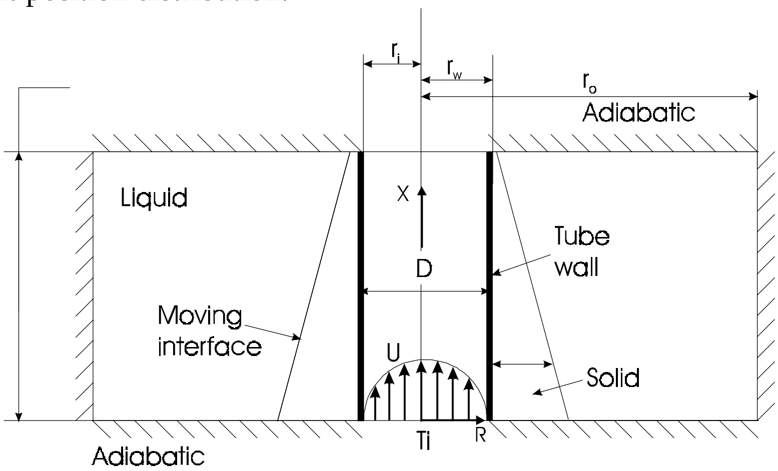

Figure 1- Schematic diagram of the PCM thermal storage system

2. MATHEMATICAL MODELLING

To study the parameters involved in the model, the following transformations are adopted:

$$\theta = \frac{T - T_m}{T_{in} - T_m},\ R = \frac{r}{D},\ X = \frac{x}{D},\ U = \frac{u}{U_m},\ \mathrm{Re}_e = \frac{U_m D}{\nu_f},\ \tau = \frac{U_m}{D} t,$$

$$C = \frac{C^o}{c_l},\ K = \frac{k}{k_l},\ St = \frac{c_l\left(T_{in} - T_m\right)}{\lambda},\ \varepsilon = \frac{\delta T}{\left(T_{in} - T_m\right)}$$

where T_m, T_{in}, and θ are phase change, inlet and non-dimensional temperatures, respectively. r and D are radius and diameter of the tube. u, U_m are local and maximum velocities, respectively. Re, ν_f, t, $\mathrm{C^o}$, c_l, k, k_l, St, λ and $2\delta T$, are Reynolds number, working fluid kinematic viscosity, time, thermal capacity, thermal capacity of liquid phase, thermal conductivity and thermal conductivity of liquid phase, Stefan number, latent heat, phase change interface, respectively.

The velocity field of the working fluid is assumed to be completely developed flow and it is given by the equation:

$$U = U_m\left(1 - R^2\right) \tag{1}$$

The dimensionless general energy equation which presents the energy equations of the HTF, the pipe wall and the pcm is

$$\left(\frac{\partial C\theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}\right)=\xi\left[\frac{1}{R}\frac{\partial}{\partial R}\left(KR\frac{\partial \theta}{\partial R}\right)+\frac{\partial}{\partial X}\left(K\frac{\partial \theta}{\partial X}\right)\right]-\frac{\partial S}{\partial \tau} \tag{2}$$

where $U = 0$ *and* $C = K = 1$ for the heat transfer fluid and the tube wall. The term S is given by;

$$S(\theta)=\begin{cases} C_{sl}\varepsilon, & \theta < -\varepsilon \\ C_{sl}\varepsilon+\dfrac{1}{St}, & \theta > \varepsilon \\ \dfrac{1}{2St}+\dfrac{\varepsilon(1+C_{sl})}{2}, & -\varepsilon \le \theta \le \varepsilon \end{cases} \tag{5}$$

and;

$$\xi=\begin{cases} \dfrac{1}{\mathrm{Re}_f\ \mathrm{Pr}_f}, & \text{For the HTF} \\ \dfrac{1}{\mathrm{Re}_f\ \mathrm{Pr}_f}\dfrac{\alpha_w}{\alpha_f}, & \text{For the tube wall} \\ \dfrac{1}{\mathrm{Re}_f\ \mathrm{Pr}_f}\dfrac{\alpha_l}{\alpha_f}, & \text{For the PCM} \end{cases} \tag{6}$$

The thermal capacity of the phase change material is non-dimensionalized in the following form:

$$C(\theta)=\begin{cases} 1, & \theta < -\varepsilon \\ C_{sl}, & \theta > \varepsilon \\ \dfrac{1}{2St\varepsilon}+\dfrac{1+C_{sl}}{2}, & -\varepsilon \le \theta \le \varepsilon \end{cases} \tag{7}$$

In the same manner the thermal conductivity is

$$K(\theta)=\begin{cases} K_{sl}, & \theta < -\varepsilon \\ K_{sl}+(1-\mathrm{K}_{sl})(\theta+\varepsilon)/2\varepsilon, & \\ \text{for } -\varepsilon \le \theta \le \varepsilon & \\ 1, & \theta > \varepsilon \end{cases} \tag{8}$$

Where $K_{sl} = \frac{k_s}{k_l}$, and $C_{sl} = \frac{c_s}{c_l}$, k_s, c_s are thermal conductivity and thermal capacity of solid phase.

The non-dimensional form of the initial and boundary conditions is

initial conditions: $\tau = 0$

the entire domain: $0 \le X \le L/D$; $0 < R < R_o$ $\theta = \varepsilon$

boundary conditions: $\tau > 0$

entrance conditions: $X = 0$; $0 < R < R_i$: $\theta = 1$

$R_i < R < R_o$: $\frac{\partial \theta}{\partial X} = 0$

exit conditions: $X = L/D$; $0 < R < R_o$: $\frac{\partial \theta}{\partial \mathrm{X}} = 0$

outer radius: $0 < X < L/D$, $\mathrm{R} = \mathrm{R}_o$; $\left.\frac{\partial \theta}{\partial R}\right|_{R=R_o} = 0$

fluid-wall interface: $k_w \left.\frac{\partial \theta}{\partial R}\right|_{R=R_i^+} = k_f \left.\frac{\partial \theta}{\partial R}\right|_{R=R_i^-}$

wall-PCM interface: $k_p \left.\frac{\partial \theta}{\partial R}\right|_{R=R_w^+} = k_w \left.\frac{\partial \theta}{\partial R}\right|_{R=R_w^-}$

where L is tube length, The subscripts *i, p, f* and *w* are internal, pcm, fluid and tube wall.

3. RESULTS AND DISSCUTION

Before presenting the numerical results for the phase change thermal storage system, the two dimensional freezing model is checked against other numerical results. They are compared with others obtained by Cao and Faghri (1991) who solved a phase change thermal storage system conjugated a forced convection. In their work they defended the importance of the solution of the momentum equations to avoid uncertainties due to the use of empirical relations. Belleci and Conti (1992) published an article showing that the use of empirical relations would not have significant effects on the results. Recently Jesus (1998) solved the same problem using temperature immobilisation method getting satisfactory results. All of the results obtained from the literature together with the results obtained by this work are presented in Fig. (2). As can be seen from the figure, the results obtained by this work are satisfactory. The parameters characterising the problem used for the comparison and solved first by Cao and Faghri in Tab. (1).

Table 1. Parameters used for model validation

$$\mathrm{Re} = 2200, \quad St = 0.5, \quad \mathrm{Pr} = 0.065, \quad \varepsilon = 0.01,$$
$$C_{sl} = K_{sl} = 1, \quad \alpha_L/\alpha_F = 0.02, \quad \alpha_W/\alpha_F ,$$
$$k_F/k_W = 1.42, \quad k_L/k_W = 1.42, \quad \mathrm{r_o}/D = 1.325$$
$$r_W/D = 0.575, \quad L/D = 12$$

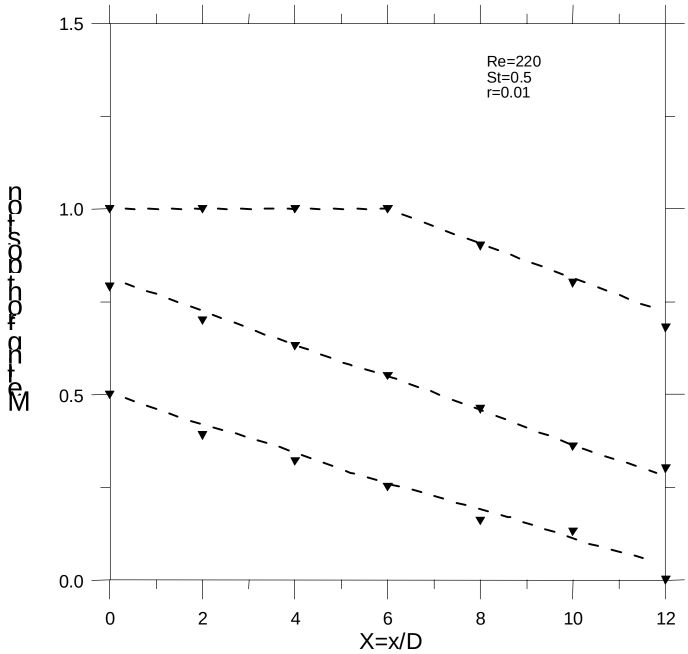

Figure 2- Solidification front position.

After checking the validity of the model, the numerical calculation for the pcm storage system was then conducted to investigate the thermal performance of a vertical phase change thermal storage system of low temperature applications. The system consists of water as the phase change material and the working fluid consists of a mixture of 70% water and 30% ethyl glycol. This type of material is the most used in ice banks (Ismail 1998) The physical and the system geometrical parameters are given in tables 2-5.

Table 2. Physical properties of water

	ρ *(kg/m³)*	c_p *(J/kg.K*	*k (W/m.K)*	λ *(J/kg)*
Solid	920.0	4217.0	2.31	333000.0
Liquid	1000.0	4225	0.57	

Table 3. Physical properties of the working fluid

ρ *(kg/m³)*	c_p *(J/kg.K)*	*k (W/m.K)*	ν *(m²/s)*	Pr
1046.0	*3640.0*	*0.49*	5.736×10^{-6}	*44.57*

Table 4. Physical properties of tube wall materials.

Tube material	ρ-(kg/m³)	c_p-(J/kg.K)	k-(W/m.K)
Copper	8954.	384.	398.
Stainless Steel	8055.	480.	15.1
PVC	1380.	960.	0.15

Table 5. Operational and geometrical parameters

Tube internal diameter (m)	0.0224
Tube wall thickness (m)	0.00155
Tube external diameter (m)	0.028
Tube length (m)	1.50
Working fluid velocity (m/s)	0.50

Initially the system was considered as liquid at its melting temperature, T_m. The heat transfer fluid inters the tubes with a lower temperature than the melting temperature of the phase change material. The energy of the phase change material is stored as both latent and sensible heat. The grid size used is 60 (axial) × 72 (radial), the last consists of 20 (HTF), 3 (tube wall) and 47 (pcm). A dimensionless length of 60 and time step of $\Delta\tau$, between 20 and 1000 is used. The dimensionless range of the mushy phase, 2ε, is taken to be 0.002 and the system initial temperature is assumed to be equal to $-\varepsilon$.

Phase change system thermal performance is a function of many parameters among others such as the time period, Reynolds and Stefan numbers, the material of tube wall and the system length. These parameters will be used in this work to evaluate the thermal performance of the system under study.

Figure 3 presents the radial temperature distribution at the middle of the pipe (X=30) for different time periods. The three regions of the domain namely the heat transfer fluid, the pipe wall and the phase change material are illustrated by the vertical lines. The solidification interfaces of the different time periods are indicated by the intersection of the line passing through θ=1.0 and the corresponding temperature curves. As can be seen from the figure the temperature curve moves downward and the corresponding solidification interface progresses to the right indicating a greater energy storage.

Figure 4 shows the solidification front position along the axial direction at different time periods. It can be seen that at a dimensionless time equal to $8.0\text{x}10^4$ the solidification interface has reached the outer radius of the system for dimensionless length (X) approximately less than 33, while some of the pcm remains liquid for X>33. The reason for that is the better thermal exchange at the entrance of the system and consequently decreasing the heat exchange up stream due to the decrease in the temperature gradient between the HTF and the pcm.

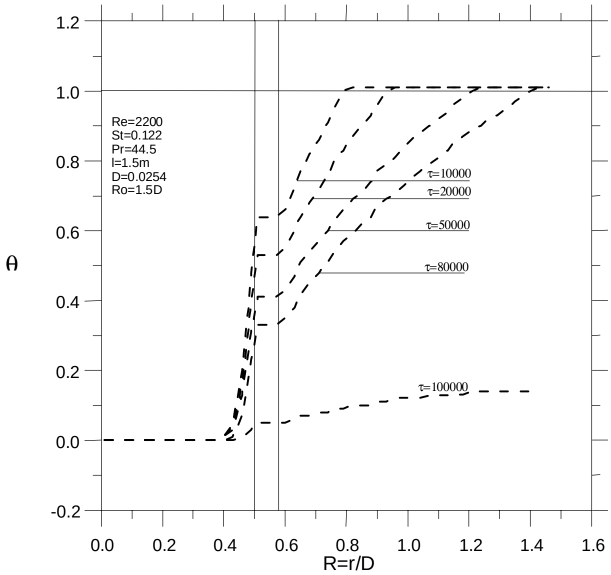

Figure 3- Radial temperature distribution at the middle of the pipe for different time periods.

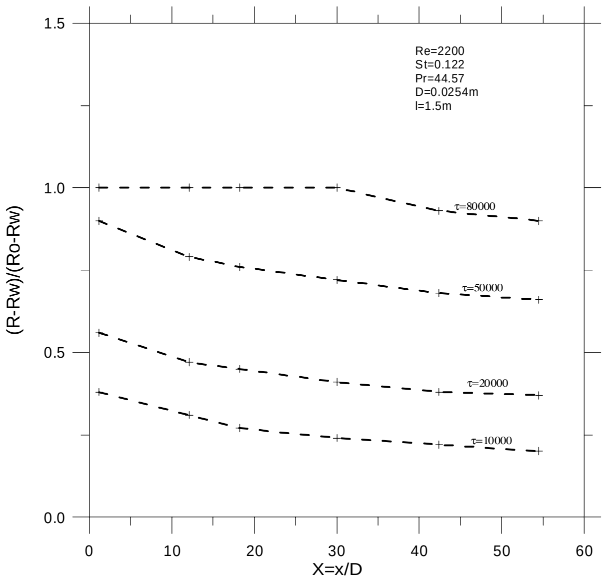

Figure 4- Location of solidification front in the axial direction

Figure 5, shows the solidification front position as a function of the dimensionless axial length for different Reynolds numbers. It can be seen that for Reynolds number greater than or equal to 2200 the solidification front has reached the outer radius for X<20. It can also be concluded that the Reynolds number is a significant parameter on pcm system thermal performance and it has to be considered when designing such systems. The parameter can supplied by nature such reveres or water falls.

Figure 6 shows the radial temperature distribution at the middle of the pipe for different Stefan numbers. As can be seen from the figure, as Stefan number increases the temperature profile moves upward indicating better heat exchange between the PCM and the HTF. This phenomenon is, by definition, due the increase of the difference between the heat transfer fluid temperature and the PCM temperature. The influence of the Stefan number on the system performance can be better illustrated in Fig. 7 which shows the radial location of the melting front as a function of the system length. The melting front position increases significantly with the increase of Stefan number.

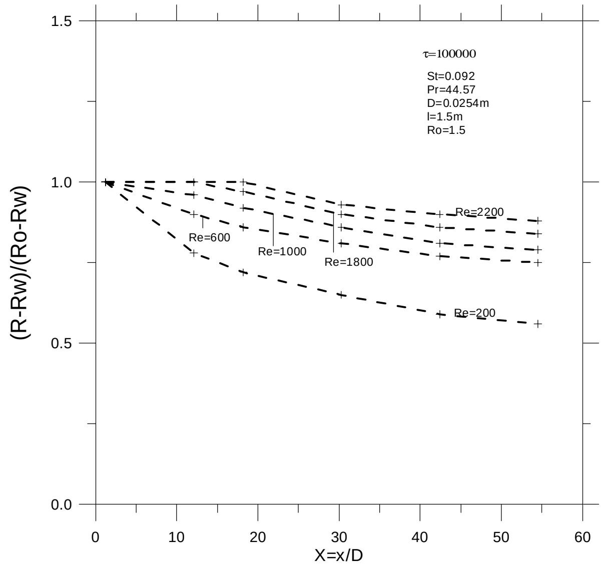

Figure 5- Position of the solidification front for different Reynolds numbers.

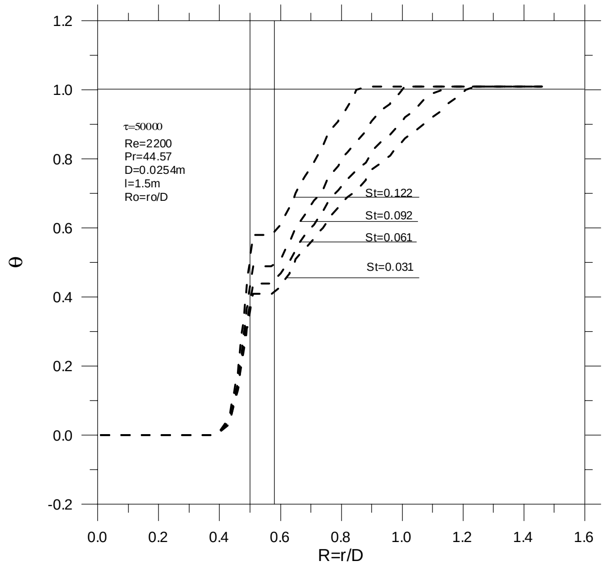

Figure 6 - Radial temperature distribution for different Stefan numbers

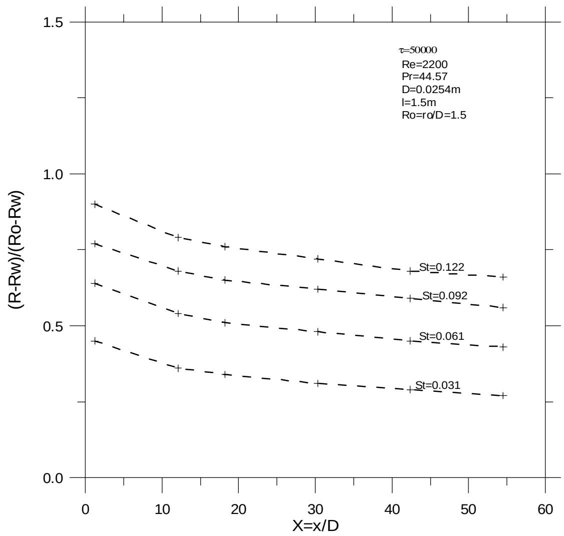

Figure 7- Position of the solidification front for different tube wall material.

The material of the tube wall is an important parameter in the prediction and design of phase change storage systems, due to its effect on the thermal performance and the total cost of the equipment. In the selection of this material the following aspects must be considered: it should be an inert material (does not react neither with the working fluid nor with the phase change material), low cost, easy maintenance and high thermal conductivity.

Figure 8 and 9 shows the influence of the tube wall material on the radial temperature distribution and solidification front position. As can be seen from the figure, systems with PVC as the wall material would give lower performance than steel or copper. The identical results of the steel and copper is due the low thermal conductivity of the phase change material and in this way both would have enough capacity to discharge the heat passed through the phase change material.

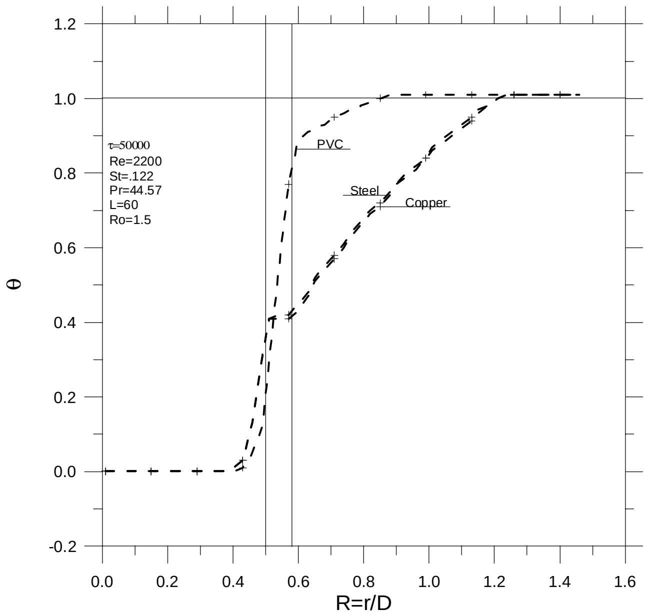

Figure 8- Radial temperature distribution for different tube wall materials.

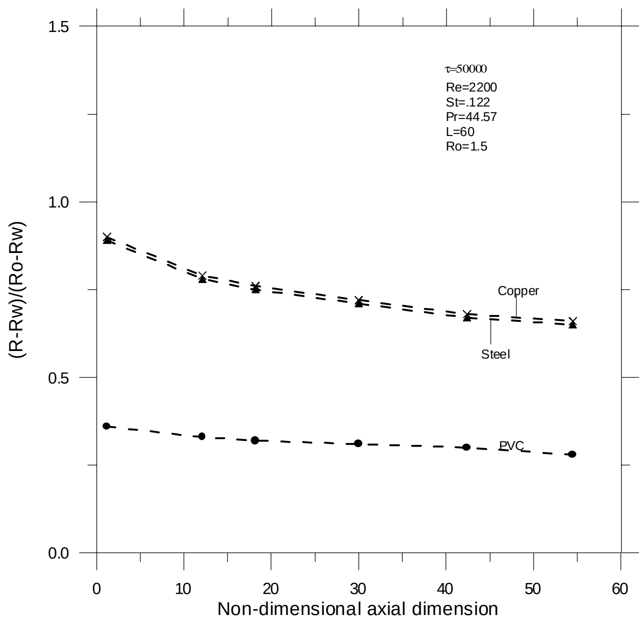

Figure 9- Solidification front position as function of tube wall materials.

4. CONCLUSION

A low temperature applications of a phase change thermal storage system is investigated using a simplified numerical model where the latent heat was included in a source term and the working fluid flow was considered to be fully developed. The influence of various parameters on the thermal performance of the system is considered. The Reynolds and Stefan numbers and the tube wall materials are shown to have a significant effect on the pcm thermal system performance. The design of any system is concerned with minimum cost. It may desirable to design a system with an efficiency lower than technologically possible if the cost is significantly reduced. Thus, if the velocity of the heat transfer fluid was not provided by nature the Reynolds number should not exceed 1500 because it has little influence on the thermal performance of the system. The same can be concluded in the case of the tube material, if the thermal performance is the only parameter of evaluation any material with thermal conductivity between the thermal conductivities of stainless steel and copper can be used.

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