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PHASE CHANGE AROUND A FINNED TUBE

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ABSTRACT

This study presents the heat transfer enhancement in the thermal energy storage system by using radially finned tube. The solution of the system consists of the solving the equations of the heat transfer fluid (HTF), the pipe wall and fin, and the phase change material (PCM) as one domain. The control volume finite difference approach and the semi implicit solver (SIS) are used to solve the equations. Fully developed velocity distribution is taken in the HTF. Flow parameters (Re number and inlet temperature of coolant) and fin parameters (the number of fins, fin length, fin thickness) are found to influence solidification fronts and the total stored energy.

Key Words : Thermal energy storage, Finned tube

KANATLI BİR BORU ETRAFINDAKİ FAZ DEĞİŞİMİ

ÖZET

Bu çalışmada, bir ısıl enerji depolama sisteminde radyal kanatlı boru kullanılmasıyla ısı transferinde yaratılan artış incelenmiştir. Sistemin çözümü, ısı transfer akışkanı (ITA), boru duvarı ve kanat ve faz değişim malzemesi (FDM) için yazılan denklemlerin bir bütün olarak çözümünü içerir. Bu denklemlerin çözümü için kontrol hacmi sonlu fark yaklaşımı ile yarı kapalı çözücü (SIS) kullanılmıştır. Isı transfer akışkanı içinde tam gelişmiş hız dağılımı alınmıştır. Akış parametreleri (Re sayısı ve soğutkanın giriş sıcaklığı) ve kanat parametrelerinin (kanat sayısı, kanat uzunluğu ve kanat kalınlığı) katılaşma eğrisi ve toplam depolanan enerjiye etkisi bulunmuştur.

Anahtar Kelimeler : Isıl enerji depolama, Kanatlı boru

1. INTRODUCTION

In designing a latent heat storage unit, the melting or solidification periods of a certain phase material has to be known. In addition, to predict the heat transfer coefficients during the phase change process, one must know the operating conditions and the storage configuration. Referring to literature, two types of configurations have been essentially studied. One of them is the shell-and-tube type heat exchanger with the phase change material placed in the shell, and the heat transfer fluid flowing through the tubes. Studies related to this configuration are done by Ismail and Alves (1986), Cao and Faghri (1991a), Cao and Faghri (1991b), Zhang and Faghri (1996), Bellecci and Conti (1993), and Lacroix (1993). The second configuration is a rigid capsule in which the phase change material has been placed, and the heat transfer material flows through a tube surrounding the capsule. It has been determined that the shell-and-tube type heat exchanger is the most promising device as a latent heat storage system that requires high efficiency for a minimum volume. As described previously, in such an energy storing unit, the phase change material (PCM) fills the shell, and the heat transfer fluid (HTF) flowing through the tubes, serves to convey the stored energy to and from the unit. Recently, a theoretical model of the shell-and-tube type unit for storing energy has been reported by Ismail and Alves (1986). In addition, Cao and Faghri (1991b, 1992) also modelled a similar problem at which both the heat charging and the recovery processes were performed by the circulating fluid. For both models, the shell wall of the unit was assumed adiabatic. Using the enthalpy model, the problem of storing energy in a shell-and-tube type unit was also solved by Bellecci and Conti (1993). Cao and Faghri (1991a) studied the latent heat energy storage systems for both annular and countercurrent flows and numerically determined that the storage system with the countercurrent flow was an efficient way to absorb heat energy. Lacroix (1993a) presented a computational methodology for solving the melting and resolidification of a phase change material around two horizontal and cylindrical sources or sinks spaced vertically.

One of the methods used for increasing the rate of energy storage is to increase the heat transfer surface area by employing finned surfaces. A review of heat transfer analysis of energy storage systems equipped with finned surfaces is provided by Humhries and Griggs (1977). Moreover, experimental studies presented by Abhat (1978; 1980) consider an application of a heat-pipe with finned surfaces as a heat transfer element of an energy storage system. To investigate the effect of fins with rectangular cross-section on the rate of melting and solidification, numerous studies both experimental and theoretical have been published. Bathelt and Viskanta (1981) studied the solidification problem around a horizontal finned tube with four different fin spacings were presented by Sasaguchi et al., (1988). Situating the finned tube vertically, Sparrow et al., (1981) experimentally investigated the shapes of the frozen layer on tubes with four fins. However, Kalhori and Ramadyani (1985) studied a similar problem with six fins on the tube. In all these studies, the rate of phase change is compared with the rate obtained when a bare tube is used under the same operating conditions. The phase change phenomena in a tank that had a rectangular cross-section with lower surface isolated and the upper surface furnished with fins was studied. Studies related to the horizontal tank were done by Henze and Humprey (1981) and the studies for the vertical tank carried out by Ho and Viskanta (1984). Solidification within two concentric cylinders having longitudinal fins was theoretically studied by Padmanabhan and Khrishna (1989) and a correlation relating the percent of solidification to the fin thickness and length, the number of fin, the Stefan and the Fourier number of the problem was stated. On the other hand, Sasaguchi and Sakamato (1989) theoretically studied the melting phenomena on the same geometry by considering strong influence of natural convection on melting. A theoretical model for predicting the transient behaviour of a shell-and-tube storage unit with the PCM on the shell side and HTF circulating through the finned tube or bare tube was presented by Lacroix (1993b). Finally, Zhang and Faghri (1996) indicated that heat transfer in the latent heat thermal energy storing system might be enhanced by using internally finned tubes.

In this study, two-dimensional phase change phenomena around a horizontal radially finned tube have been investigated numerically. Determining the temperature distribution through the storage tank, the amount of stored energy has been estimated accurately. Moreover, the effects of fin diameter and spacing on solidification are also displayed.

2. MATHEMATICAL MODELLING

The physical model of the problem is shown in Figure 1. The phase change material fills the shell space which is out of finned tube while heat transfer fluid flows inside the tube of radius r_i. The PCM is assumed adiabatic at the outer radius of thermal energy storage tank, r_{inf}. In addition, the following assumptions are made:

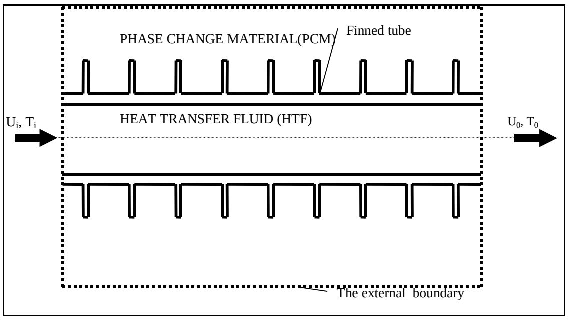

Figure 1. Schematic of the thermal energy storage system with externally finned tube

- The thermophysical properties of the PCM, finned tube, and heat transfer fluid are independent of temperature. But, the properties of PCM can be different in solid and liquid phases.
- The PCM is homogeneous and isotropic.
- System is well insulated.
- The transfer fluid flow inside the tube is laminar with fully development inlet velocity.
- As the initial temperature of the system is considered to be the same or closed to the phase change temperature, the natural convection effect around the tube and the fins can be neglected.

The dimensionless energy equation governing two-dimensional transient incompressible laminar flows with no viscous dissipation is

$$\frac{\partial T}{\partial \tau} + V\frac{\partial T}{\partial R} + U\frac{\partial T}{\partial X} = \frac{1}{\mathbf{Re}_f\,.\mathbf{Pr}_f}\left(\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial T}{\partial R}\right) + \frac{\partial^2 T}{\partial X^2}\right) \quad (1)$$

The heat conduction in the PCM is described by a temperature transforming model using a fixed grid numerical model (Cao and Faghri, 1990). This model assumes that the solidification process occurs over a range of phase change temperature from $(T_m^0 - \delta T^0)$ to $(T_m^0 + \delta T^0)$, but it can also be successfully used to simulate the solidification process occurring at a single temperature by taking a small range of phase change temperature, $2\delta T^0$. This model has the advantage of eliminating the time step and grid size limitations that are normally encountered in other fixed grid methods.

The dimensionless energy equation for the PCM is written as,

$$\frac{\partial (CT)}{\partial \tau} = \frac{1}{\mathbf{Re}_f\,.\mathbf{Pr}_f}\frac{\alpha_l}{\alpha_f}\left(\frac{1}{R}\frac{\partial}{\partial R}\left(KR\frac{\partial T}{\partial R}\right) + \frac{\partial}{\partial X}\left(K\frac{\partial T}{\partial X}\right)\right) - \frac{\partial S}{\partial \tau} \quad (2)$$

where,

$$C = C(T) = \begin{cases} \rho_{sl}C_{sl} & T < -\delta T^* \quad \text{solid phase} \\ \frac{1}{2}(1+\rho_{sl})*\left(\frac{1}{2}(1+C_{sl}) + \frac{1}{2.Ste.\delta T^*}\right) & -\delta T^* \le T \le \delta T^* \quad \text{mushy phase} \\ 1 & T > \delta T^* \quad \text{liquid phase} \end{cases} \quad (3)$$

$$S = S(T) = \begin{cases} \rho_{sl}C_{sl}\delta T^* & T^* < -\delta T^* \quad \text{solid phase} \\ \frac{1}{2}(1+\rho_{sl})*\left(\frac{1}{2}\delta T^*(1+C_{sl}) + \frac{1}{2Ste}\right) & -\delta T^* \le T \le \delta T^* \quad \text{mushy phase} \\ C_{sl}\delta T^* + \frac{1}{Ste} & T > \delta T^* \quad \text{liquid phase} \end{cases} \quad (4)$$

$$K = K(T) = \begin{cases} K_{sl} & T < -\delta T^* \quad \text{solid phase} \\ K_{sl} + (1-K_{sl})(T+\delta T^*)/2\delta T^* & -\delta T^* \le T \le \delta T^* \quad \text{mushy phase} \\ 1 & T > \delta T^* \quad \text{liquid phase} \end{cases} \quad (5)$$

The dimensionless energy equation for the tube wall is

$$\frac{\partial T}{\partial \tau} = \frac{1}{Re_f\,.Pr_f}\frac{\alpha_w}{\alpha_f}\frac{k_l}{k_w}\left[\frac{1}{R}\frac{\partial}{\partial R}\left(KR\frac{\partial T}{\partial R}\right) + \frac{\partial}{\partial X}\left(K\frac{\partial T}{\partial X}\right)\right] \quad (6)$$

In this study, following non-dimensional variables are used.

$$R = \frac{r}{D},\quad X = \frac{x}{D},\quad U = \frac{u}{u_0},\quad V = \frac{v}{u_0},\quad Re_f = \frac{u_0 D}{\gamma_f}$$

$$T = \frac{T^0 - T_m^0}{T_m^0 - T_{in}^0},\quad \tau = \frac{u_0 t}{D},\quad P = \frac{p - p_0}{\rho_f u_o^2}$$

$$C = \frac{C^0}{\rho_l c_l},\quad K = \frac{k}{k_l},\quad S = \frac{S^0}{\rho_l c_l.(T_m^0 - T_{in}^0)},$$

$$Ste = \frac{c_l.(T_m^0 - T_{in}^0)}{\Delta H},\ \delta T = \frac{\delta T^0}{(T_m^0 - T_{in}^0)}$$

3. THE NUMERICAL PROCEDURE

Temperature distribution inside the solution region can be calculated by solving the energy equation, which includes the components of the velocity. The solution procedure used for solving this energy equation which has been defined by equation (1) to (6) is the control volume approach described by Patankar (Patankar, 1980). In this problem, fully developed velocity profile is used and only the energy equation is solved. At the PCM / wall, the wall / fluid interfaces, and the fin / PCM, the harmonic mean of thermal conductivity is used. Semi implicit solver (SIS) is used for solving the energy equation (Lee, 1989). Since the problem is two-dimensional, this algorithm is chosen. During each time step, iterations are needed. When a definite iteration is made, which satisfies predetermined convergence criteria, it is passed to next time level and the procedure is continued till a given time is reached. Different grid sizes and time steps for the same problem have been tested and it has been proven that both PCM model and the numerical scheme are essentially independent of grid sizes for the numerical results.

4. NUMERICAL RESULTS AND DISCUSSION

Before presenting the numerical results for this problem, the results of melting around a bare tube obtained by the use of SIMPLE algorithm (Cao and Faghri, 1991b) and by the use of analytical velocity distribution (Govier and Aziz, 1972) are given in the same figure to make comparison possible. Figure 2 shows this comparison. In can be seen that the present model agrees well with the results of the related study.

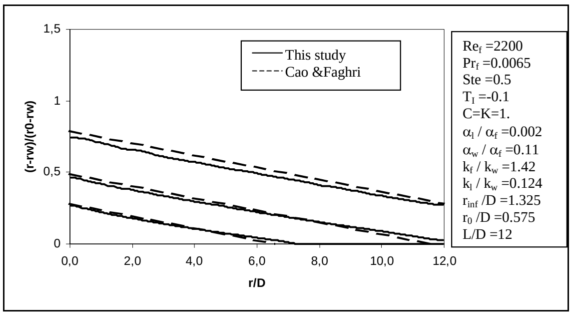

Figure 2. Melting fronts along axial direction for different time (τ=50,150,400)

The numerical calculations for the latent energy storage system were then conducted with configuration as shown in Figure 1. The system is initially at a temperature T_i=3.5°C. This value is initial temperature used in experimental study and greater than the melting temperature. The heat transfer fluid enter into pipe at a temperature T_{in}=-10°C. In the numerical calculations, Re numbers of HTF are taken as 500 and 1000.

The grid size for used the numerical calculation were 264 (axial) x 160 (radial) for n=11 and 360 (axial) x 160 (radial) for n=17. In order to assess the effect of fin spacing X_L, the fin radius R_f, and Re number, a series of 8 numerical experiments were performed. The numerical experimental conditions are summarised in Table 1.

For the certain fin-to-tube radius ratio, fin spacing and Re number, solidification fronts are given in Figure 3 through Figure 8 for the different time, which are τ=10000, 20000, 30000, 40000, 50000 respectively.

Table 1 Summary Of Numerical Experiment

Case	Re	R_f	$X_0/X_L/n$
1	500	1.25	1.4125/1.2/17
2	500	1.25	1.5375/2.0/11
3	500	1.25	1.4125/1.2/17
4	500	1.25	1.5375/2.0/11
5	1000	1.75	1.4125/1.2/17
6	1000	1.75	1.5375/2.0/11
7	1000	1.75	1.4125/1.2/17
8	1000	1.75	1.5375/2.0/11
R_i=0.5, R_o=0.75, R_{inf}=15, X_f=0.175			

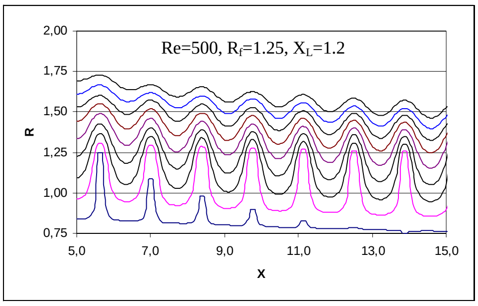

Figure 3. Solidification fronts with respect to axial direction for different time (τ = 10000, 20000,….., 90000)

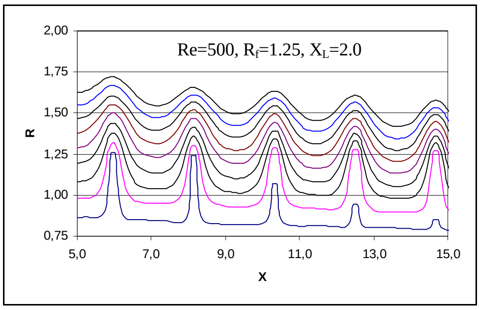

Figure 4. Solidification fronts with respect to axial direction for different time (τ = 10000,20000,….., 90000)

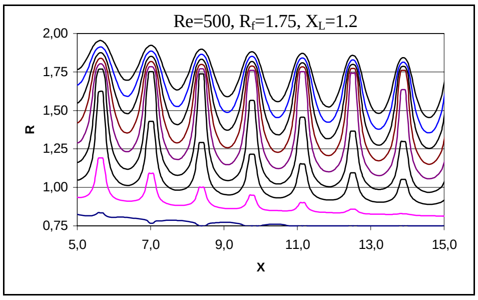

Figure 5. Solidification fronts with respect to axial direction for different time (τ = 10000, 20000,..….., 90000)

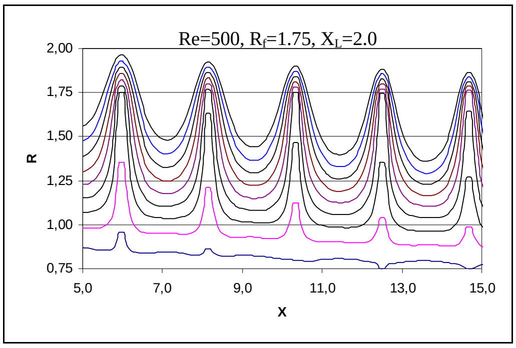

Figure 6. Solidification fronts with respect to axial direction for different time (τ = 10000, 20000,….., 90000)

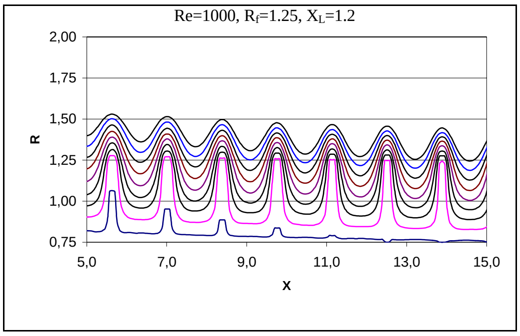

Figure 7. Solidification fronts with respect to axial direction for different time

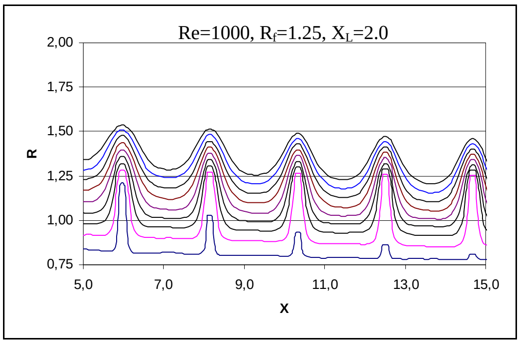

Figure 8. Solidification fronts with respect to axial direction for different time (τ = 10000, 20000,….., 90000)

Because of the increment in Re number, the heat transfer and storage energy are expected to rise. But, in our solutions, higher energy storage is obtained for low Re number (500) than for high Re number (1000). Because, while Re number is increasing for constant tube diameter D and thermal properties of heat transfer fluid, the mean fluid velocity and dimensionless time increases. Hence, to make a comparison, the value of dimensionless time for Re number (1000) should be taken half of the value of dimensionless time for Re number (500).

4. COMPARISON AND CONCLUSIONS

In this study, the heat transfer enhancement in the energy storage system by using radially finned tube is studied numerically. The computational code of this model is checked by using experimental data (Erek, 1999). Erek (1999) shows the comparison of the fin tip temperature and the total energy storage by the present numerical method and experimental study. The effect on solidification fronts for different fin parameters and flow parameters is shown in Figure 3 - 8. The results show the solidification fronts can be significantly increased if the fin height is increased. These results give some knowledge for the design of the thermal energy storage system. To obtain more precision results, the studies are repeated for different parameters and different cases that have large scale. The influence of these parameters will be analysed future studies.

5. NOMENCLATURE

c	:	specific heat [J kg^{-1} K^{-1}]
c_m	:	specific heat of mushy phase, $(c_s+c_l)/2$ [J kg^{-1} K^{-1}]
C^0	:	coefficient in equation (3)
C	:	$C^0/(c_l.\rho_l)$
C_{sl}	:	c_s/c_l
D	:	inside diameter of the circular pipe [m]
k	:	thermal conductivity [W m^{-1} K^{-1}]
K	:	dimensionless thermal conductivity, k/k_l
K_{sl}	:	k_s/k_l
L	:	length of pipe [m]
M, N	:	total grid size in radial and axial direction
Pr_f	:	fluid Prandtl number, ν_f/α_f
Re_f	:	fluid Reynolds number, u_0 D/ν_f
S^0	:	term in equation (4)
S	:	source term, $S^0/\rho_l c_l (T_m^0 - T_{in}^0)$
s	:	interface position [m]
Ste	:	Stefan number
T^0	:	temperature [K]
T_m^0	:	melting temperature [K]
T	:	dimensionless temperature, $(T^0 - T_m^0)/(T_m^0 - T_{in}^0)$
t	:	time [s]
u_0	:	inlet velocity [m s^{-1}]
U,V	:	dimensionless velocities, u/u_0, v/u_0
u,v	:	velocities [m s^{-1}]
X,R	:	dimensionless coordinate direction, x/D, r/D
x,r	:	coordinate direction.

Greek Symbols

α : thermal diffusivity [m² s^{-1}]
$2\delta T^0$: phase-change temperature range, or mushy phase range [K]
δT^* : $\delta T^0 /(T_{in}^0 - T_m^0)$ or $\delta T^0 /(T_m^0 - T_{in}^0)$
ν : kinematic viscosity [m² s^{-1}]
μ : dynamic viscosity [N s m^{-2}]
ρ : density [kg m^{-3}]
τ : dimensionless time, $u_0.t/D$

Subscripts

f : transfer fluid or fin
i : initial condition, or inside radius of the pipe
in : inlet
inf : outside radius of the thermal storage tank
L : liquid PCM
m : mushy phase
o : outside radius of the pipe
p : PCM
s : solid PCM
w : container wall

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