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Heat transfer enhancement in latent heat thermal energy storage system by using the internally finned tube

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Abstract—The heat transfer enhancement in the latent heat thermal energy storage system by using an internally finned tube is presented in this paper. The phase change material fills the annular shell space around the tube, while the transfer fluid flows within the internally finned tube. The melting of the phase change material is described by a temperature transforming model coupled to the heat transfer from the transfer fluid. The heat conduction in the internal fins is an unsteady two-dimensional heat conduction problem and is solved by a finite difference method. The results showed that adding internal fins is an efficient way to enhance the heat transfer in thermal energy storage systems when a fluid with a low thermal conductivity is used as the transfer fluid. Copyright © 1996 Elsevier Science Ltd.

INTRODUCTION

The thermal energy storage system is very important to solar energy utilization systems because of the periodic feature of solar energy. The latent heat thermal energy storage system using a phase change material (PCM) is an efficient way of storing or releasing a large amount of heat during melting or solidification. An effective latent heat storage system is the shell-and-tube heat exchanger. In this exchanger, the PCM fills the annular shell space around the tube, while the transfer fluid flows within the tube. This type of latent heat storage system has been studied by a number of investigators employing numerical and analytical methods [1–5]. Since the thermal conductivity of the PCM is relatively low, some researchers have studied the heat transfer enhancement obtained by adding fins to the PCM side of the thermal energy storage system [6,7]. Recently, Lacroix [8] studied the heat transfer behavior of a latent heat thermal energy storage unit using an externally finned tube surrounded by PCM. The phase change problem of the PCM (*n*-Octadecane) around an externally finned tube was studied by using an enthalpy model and linked to the heat transfer from the transfer fluid (water) inside the tube. The present authors [9] studied the same problem by using a temperature transforming model [10] for the melting of PCM and an analytical method [5] for the transfer fluid inside the tube. The results of refs. [8,9] showed that adding fins to the PCM side of the thermal energy storage system is an efficient way to enhance the heat transfer in the latent heat thermal energy storage system if a high thermal conductive fluid is used as the transfer fluid inside the tube.

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When air is used as the transfer fluid [4,11], the forced convective heat transfer coefficient inside the tube is very low due to the very low thermal conductivity of air. Therefore, the main thermal resistance of the thermal energy storage system occurs in the transfer fluid instead of in the PCM. In this case, the addition of fins to the PCM side of the tube is not an efficient way to enhance the heat transfer in the latent heat thermal energy storage system. It is therefore necessary to enhance the forced convective heat transfer inside the tube. Although there are many different ways to enhance the turbulent heat transfer inside the tube, using internal fins is a simple and efficient way to enhance the internal convective heat transfer [12]. Edwards and Jensen [13] summarized different experimental and numerical results of convective heat transfer in internally finned tubes. They then developed a method of predicting the pressure drop and heat transfer rates of turbulent flow in internally finned tubes by determining a characteristic length for the flow. The correlation proposed by Edwards and Jensen [13] provides a simple and accurate method for heat exchanger designers to evaluate the performance of finned tubes in heat exchanger applications.

In this paper, the heat transfer enhancement of the latent heat thermal energy storage system by using an internally finned tube will be studied. The melting process of the PCM in the annular shell space outside the tube will be solved by the temperature transforming model that was proposed by Cao and Faghri [10]. This temperature transforming model is linked to the heat transfer from the transfer fluid inside the internally finned tube. The forced convective heat transfer coefficient inside the tube is quoted from ref. [13]. The heat conduction in the internal fins is an unsteady two-dimensional heat conduction problem

NOMENCLATURE

C^0 heat capacity [J (m^3 K)$^{-1}$]
C dimensionless heat capacity, C^0/C^0_ℓ
D internal diameter of the tube, $2r_i$
e height of longitudinal fins [m]
E dimensionless height of longitudinal fins, e/r_i
Fo Fourier number, $\alpha_\ell t/r_i^2$
h local heat transfer coefficient [W (m^2 K)$^{-1}$]
H latent heat of melting [J kg^{-1}]
k thermal conductivity [W (m K)$^{-1}$]
K dimensionless thermal conductivity, k/k_ℓ
l length of the thermal energy storage system [m]
L dimensionless length of the thermal energy storage system, l/r_i
n number of longitudinal fins
Nu local Nusselt number, hD/k_f
Q_{fin} dimensionless heat flux conducting by fin
r radial coordinate [m]
R dimensionless radial coordinate, r/r_i
Re Reynolds number, $U_m D/\nu$
r_i internal radius of the tube [m]
r_m radius of the melting front [m]
R_m dimensionless radius of the melting front, r_m/r_i
r_o radius of the shell [m]
R_o dimensionless radius of the shell, r_o/r_i
S^0 term in equation (1) [J kg^{-1}]
S $S^0/C^0_\ell(T^0_{in}-T^0_m)$
Ste Stefan number, $C^0_\ell(T^0_{in}-T^0_m)/(\rho H)$
T^0 temperature, [K]
T dimensionless temperature, $(T^0-T^0_m)/(T^0_{in}-T^0_m)$
U_m average velocity of the transfer fluid inside the tube [m s^{-1}]
w thickness of the longitudinal fins [m]
W dimensionless thickness of the longitudinal fins, w/r_i
x axial coordinate [m]
X dimensionless axial coordinate, x/r_i
y coordinate along height direction of the fin [m]
Y dimensionless coordinate along height direction of the fin, y/r_i.

Greek symbols
α thermal diffusivity [m^2 s^{-1}]
δT $\delta T^0/(T^0_{in}-T^0_m)$
$2\delta T^0$ phase-change temperature range [K]
η heat transfer enhancement ratio, equation (23)
ρ density.

Subscripts
f transfer fluid
fin longitudinal fins
i initial condition, or internal radius of the tube
in inlet
ℓ liquid phase
m melting point
o shell
s solid phase.

and will be solved by a finite difference method. The effects of the geometric structure of the internal longitudinal fins on the melting process will also be investigated.

PHYSICAL MODEL OF THE MELTING PROBLEM

The physical model of the problem is shown in Fig. 1. The PCM fills the annular shell space of inner radius r_i and outer radius r_o, while the transfer fluid flows inside the internally finned tube in the longitudinal direction, with an outer radius of r_i. The geometric structure of the internal longitudinal finned tube is also shown in Fig. 1. The PCM is assumed adiabatic at the outer radius, r_o. In order to solve this problem, the following assumptions are necessary:

(1) The thermophysical properties of the PCM and transfer fluid are independent of temperature. However, the properties of the PCM can be different in the solid and liquid phases.

(2) The PCM is homogeneous and isotropic.

(3) The thickness of the tube wall is assumed to be zero.

(4) The tube wall temperature is uniform along the circumference, therefore the melting in the PCM is axisymmetric around the tube.

(5) The melting of the PCM occurs at a single temperature, T^0_m.

(6) The effect of natural convection in the liquid PCM is ignored.

(7) The transfer fluid flow inside the tube is turbulent with fully developed inlet velocity and heat transfer.

The heat conduction in the PCM is described by a temperature transforming model using a fixed grid numerical method [10]. This model assumes that the melting process occurs over a range of phase change temperatures from $(T^0_m-\delta T^0)$ to $(T^0_m+\delta T^0)$, but it can also be successfully used to simulate the melting process occurring at a single temperature by taking a

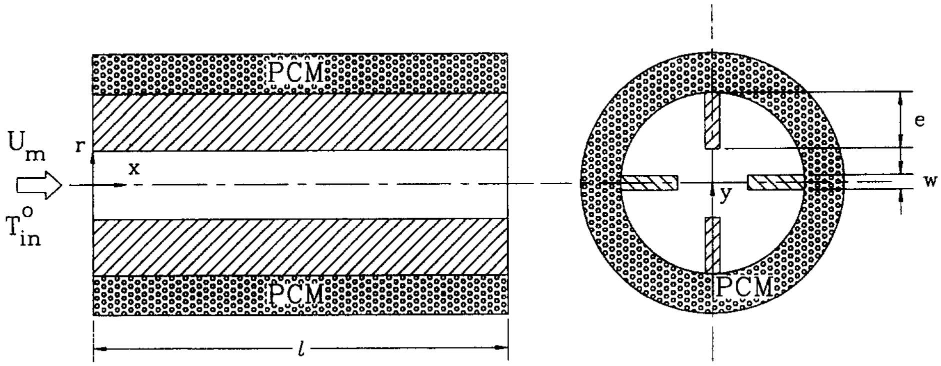

Fig. 1. Schematic of PCM energy storage system with internally finned tube.

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 governing the energy equation for the PCM is [9]

$$\frac{\partial(CT)}{\partial Fo} = \frac{1}{R}\frac{\partial}{\partial R}\left(KR\frac{\partial T}{\partial R}\right) + \frac{\partial}{\partial X}\left(K\frac{\partial T}{\partial X}\right) - \frac{\partial S}{\partial Fo}$$
$$0 < X < L, \quad 1 < R < R_o \quad (1)$$

where

$$C(T) = \begin{cases} C_s & T < -\delta T \\ \frac{1}{2}(1+C_s) + \frac{1}{2Ste\,\delta T} & -\delta T \leqslant T \leqslant \delta T \\ 1 & T > \delta T \end{cases} \quad (2)$$

$$K(T) = \begin{cases} K_s & T - \delta T \\ K_s + \frac{(1-K_s)}{2\delta T}(T+\delta T) & -\delta T \leqslant T \leqslant \delta T \\ 1 & T > \delta T \end{cases} \quad (3)$$

$$S(T) = \begin{cases} C_s\delta T & T < -\delta T \\ \frac{1}{2}(1+C_s)\,\delta T + \frac{1}{2Ste} & -\delta T \leqslant T \leqslant \delta T \\ C_s\delta T + \frac{1}{Ste} & T > \delta T. \end{cases} \quad (4)$$

The initial condition and boundary conditions are

$$T = T_i \quad 0 \leqslant X \leqslant L, \quad 1 \leqslant R \leqslant R_o, \quad Fo = 0 \quad (5)$$

$$-K\frac{\partial T}{\partial R} = \frac{1}{2}K_f Nu\left(1 - \frac{n}{2\pi}W\right)(T_f - T) + \frac{n}{2\pi}WQ_{fin} \quad R = 1 \quad (6)$$

$$\frac{\partial T}{\partial R} = 0 \quad R = R_o \quad (7)$$

$$\frac{\partial T}{\partial X} = 0 \quad X = 0, L \quad (8)$$

where Q_{fin} in equation (6) is the dimensionless heat flux conducted by the fins. It can be calculated by using the method described in the next section. The dimensionless transfer fluid temperature, T_f, can be determined by the following equation which is developed by an energy balance on a fluid control volume:

$$\frac{C_f}{K_f}\left(1 - \frac{n}{\pi}EW\right)\frac{\partial T_f}{\partial Fo} = \left(1 - \frac{n}{2\pi}W\right)Nu(T|_{R=1} - T_f) - \frac{n}{\pi K_f}WQ_{fin} - \frac{1}{2}\left(1 - \frac{n}{\pi}EW\right)Pe\frac{\partial T_f}{\partial X}. \quad (9)$$

The initial condition and boundary condition of equation (9) are

$$T_f = T_i, \quad Fo = 0 \quad (10)$$

$$T_f = 1, \quad X = 0 \quad (11)$$

where the definition of the dimensionless variables can be found in the Nomenclature.

The Nusselt number in equations (6) and (9) is defined with the internal diameter of the tube as the characteristic length. It can be calculated by the following empirical correlation proposed by Edwards and Jensen [13]:

$$Nu = \frac{0.023(l_c/D)^{0.2}}{(1.35 - 0.35l_c/D)}Re^{0.8}Pr^{0.4} \quad (12)$$

where

$$\frac{l_c}{D} = \frac{(1-E)^3}{1 - \frac{n}{\pi}WE} + \left(1 - \frac{(1-E)^2}{1 - \frac{n}{\pi}WE}\right)\left(\frac{\pi}{n}\left(1 - \frac{E}{2}\right) - \frac{W}{2}\right). \quad (13)$$

The key module parameter used to evaluate the performance of a thermal energy storage system is the melting volume fraction (MVF), which can be calculated by the following equation:

$$MVF = \frac{1}{(R_o^2-1)L}\int_0^L (R_m^2-1)\,dX \qquad (14)$$

where the dimensionless radius of the melting front, R_m, can be determined after a converged temperature distribution is obtained.

HEAT CONDUCTION IN THE FINS

Since it is assumed that the heat conduction in each fin is identical, a single longitudinal fin is investigated. The heat conduction in a longitudinal fin is an unsteady two-dimensional heat conduction problem. The dimensionless governing equation of the problem is as follows:

$$\frac{\partial^2 T_{fin}}{\partial X^2}+\frac{\partial^2 T_{fin}}{\partial Y^2}+\frac{K_f Nu}{K_{fin}W}(T_f-T_{fin}) = \frac{C_{fin}}{K_{fin}}\frac{\partial T_{fin}}{\partial Fo}$$

$$0 < X < L,\quad 0 < Y < E \qquad (15)$$

with the following boundary conditions and initial condition

$$T_{fin} = T|_{R=1} \quad Y = 0 \qquad (16)$$

$$-\frac{\partial T_{fin}}{\partial Y} = \frac{K_f Nu}{2K_{fin}}(T_{fin}-T_f) \quad Y = E \qquad (17)$$

$$\frac{\partial T_{fin}}{\partial X} = 0 \quad X = 0, L \qquad (18)$$

$$T_{fin} = T_i \quad Fo = 0 \qquad (19)$$

where the dimensionless variables in equations (15)–(19) can be found in the Nomenclature.

The above equations (15)–(19) will be solved by a finite difference method [14]. After the converged solution is obtained, the dimensionless heat flux conducted by a longitudinal fin can be expressed as

$$Q_{fin} = K_{fin}\left.\frac{\partial T_{fin}}{\partial Y}\right|_{Y=0}. \qquad (20)$$

NUMERICAL SOLUTION PROCEDURE

The melting problem has been specified mathematically by equations (1)–(11). These equations can be solved by a finite difference method [14]. In this methodology, the discretization equations are obtained by applying the conservation laws over a finite size control volume surrounding the grid node and integrating the equation over the control volume. For the dimensionless thermal conductivity K, it is important to use the harmonic mean at the faces of the control volume.

Since equation (1) is a non-linear unsteady heat conduction equation, and is linked to the heat transfer from the transfer fluid and internal longitudinal fins inside the tube, iteration is needed during the calculation process. Converged results were assumed to be reached when the maximum relative change of temperature between the consecutive iterations was less than 10^{-5}. In order to obtain a numerical solution which was not affected by the grid size, different grid sizes for the same problem were tested. The grid size used for the calculation was 42(axial) × 22(radial PCM) × 22(fin height) with a dimensionless time step $\Delta Fo = 0.1$. Although smaller grid sizes and dimensionless time steps ($62\times32\times32, \Delta Fo = 0.05$) were also used to simulate a few cases, the maximum changes in the MVF were less than 0.8%.

Furthermore, in order to verify the numerical results obtained by the above method, the overall energy balance is checked during the calculation process. At any point in time, this energy balance should show that the changes in internal energy of the PCM and the fins should be equal to the total energy supplied by the transfer fluid inside the tube. The dimensionless overall energy balance can be expressed as

$$MVF\pi(R_o^2-1)L+2\pi Ste\int_0^L\int_1^{R_o}(T-C_sT_i)R\,dR\,dX$$

$$+nWSte\int_0^L\int_0^E C_{fin}(T_{fin}-T_i)\,dY\,dX$$

$$=\frac{1}{2}RePrK_fSte(\pi-nWE)\int_0^{Fo}(1-T_f|_{X=L})\,dFo. \qquad (21)$$

The left-hand side of equation (21) represents the thermal energy stored in the PCM and fins, while the right-hand side represents the thermal energy supplied by the transfer fluid. The energy balance calculation at any point time, for each case, showed that the numerical deviation between the two sides of equation (21) was always smaller than 1%.

RESULTS AND DISCUSSION

Before studying the effect on the internal longitudinal fins on the heat transfer of the PCM thermal energy storage system, the code was examined by simulating the one-region phase change problem presented by Sparrow and Hsu [15]. Figure 2 compares the results of the present study for a dimensionless thickness of the freezing layer with the numerical solution presented by Sparrow and Hsu [15]. Sparrow and Hsu's solution was a function of Biot number, $K_fNu/2$, and Stanton number, $Nu/RePr$. The case in which the comparison was made corresponds to $K_fNu/2 = 5.0$ and $Nu/RePr = 0.003$. In the present calculation, the parameters such as K_f and Re were determined by Biot number and Stanton number. The Nusselt number, Nu, found in the Biot number and Stanton number was calculated by the well-known Dittus–Boelter correlation [16] for a given Prandtl number. It can be seen from Fig. 2 that there was very good agreement

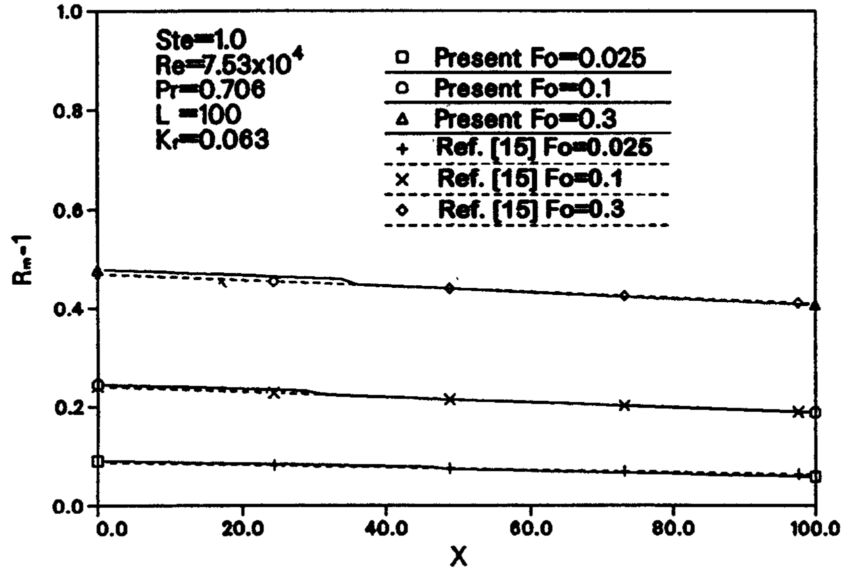

Fig. 2. Comparison of the present study result of the melting front with ref. [15].

between the result of the present study and the numerical result of ref. [15]

Figure 3 shows the effect of internal longitudinal fins on the melting front radius. In order to study the effect of internal longitudinal fins on the melting process, the diameter of the tube is assumed to be unchanged. Therefore, the flow area of the tube will be decreased if internal longitudinal fins are added. If the average velocity of the fluid is also kept unchanged, the flow rate of the transfer fluid will be decreased. However, the decrease of flow rate can not be allowed from the engineering view point. Therefore, the flow rate of the transfer fluid needs to be kept constant for both plain tubes and internally finned tubes for comparison. This means that the average velocity of the transfer fluid should be increased after the addition of internal longitudinal fins. In order to keep a constant flow rate, the Reynolds number defined by using the diameter of the tube as a characteristic length should be correspondingly increased. The Reynolds number after adding the internal fins should be

$$Re = \frac{\pi}{(\pi - nWE)} Re_o. \tag{22}$$

As can be seen from Fig. 3, the melting process is significantly accelerated by adding internal fins. The acceleration of the melting process at the entrance section is slightly more significant than that at the end section of the tube. The acceleration of the melting process is caused by two reasons: the Nusselt number

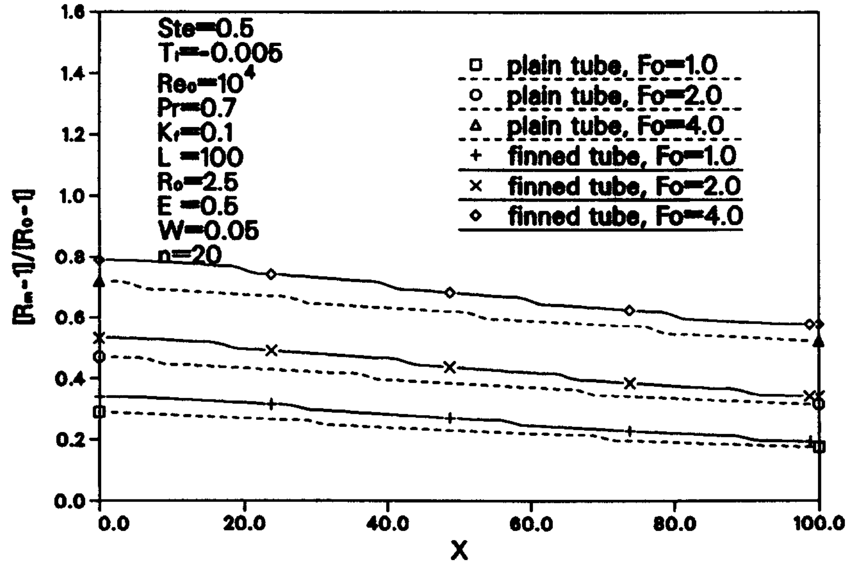

Fig. 3. Effect of internal fins on the melting front.

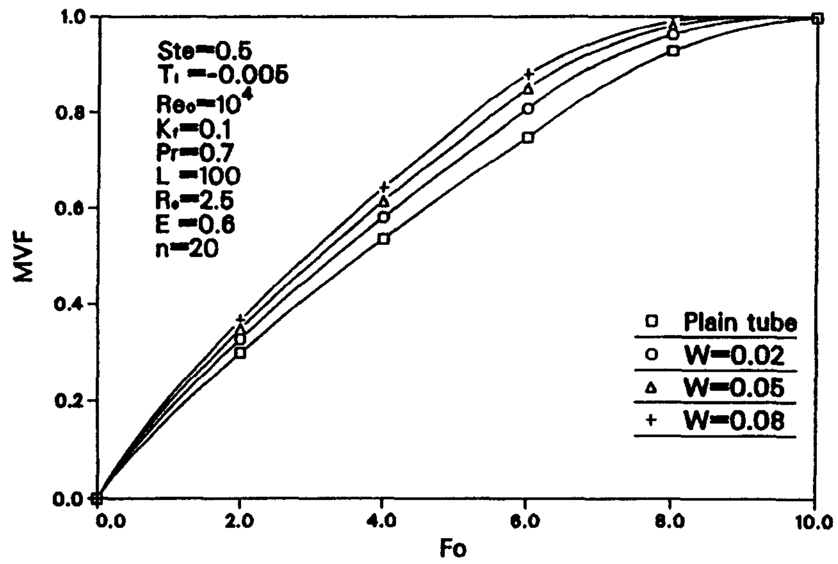

Fig. 4. Effect of fin thickness on the *MVF*.

inside the tube is increased due to the effect of the internal longitudinal fins; and the internal longitudinal fins directly conduct heat to the PCM.

Figure 4 shows the effect of internal fin thickness of the *MVF*. It can be seen that the *MVF* can be significantly increased by adding internal longitudinal fins. The *MVF* increases with a corresponding increase in internal fin thickness. However, when the dimensionless internal fin thickness is larger than 0.05, the effect of internal fin thickness on the *MVF* is not significant. Therefore, a dimensionless internal fin thickness of $W = 0.05$ is used in the following calculations. Figure 5 shows the effect of the internal fin height on the *MVF*. As can be seen, the *MVF* can be significantly increased by increasing the internal fin height. Figure 6 shows the effect of the internal fin numbers on the *MVF*. The *MVF* increases with an increase of internal fin number. However, after the internal fin number reaches 20, any increase in fin number is not an efficient way to enhance the melting process.

Figure 7 shows the effect of the Reynolds number on the *MVF*. The *MVF* for both plain tubes and internally finned tubes at two different Reynolds numbers are given in Fig. 7. The internal fin is shown to be an efficient way of enhancing the melting process for $Re_o = 10^4$. However, for $Re_o = 5 \times 10^4$, the increase of *MVF* by internal fins is not very significant. This is because the proportion of thermal resistance at the transfer fluid side will decrease with an increase

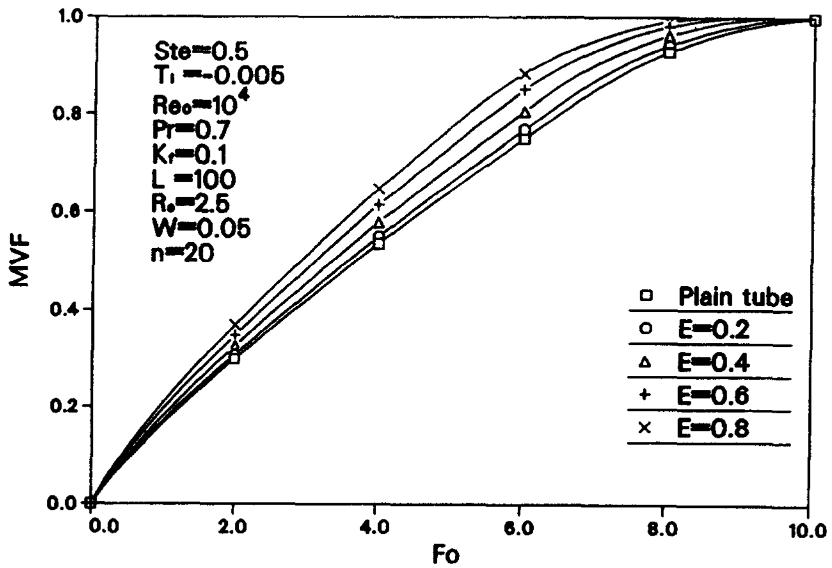

Fig. 5. Effect of fin height on the *MVF*.

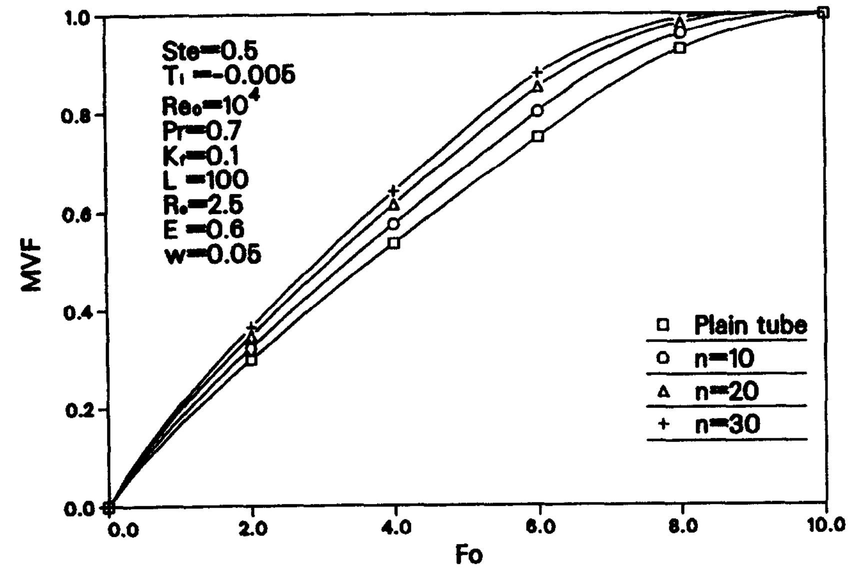

Fig. 6. Effect of fin number on the MVF.

in Reynolds number. Therefore, using internally finned tubes is an efficient way to enhance the melting process only for low Reynolds number.

Figure 8 shows the effect of dimensionless thermal conductivity of transfer fluid, K_f, on the MVF. It can be seen that the internal longitudinal fins are more efficient for smaller K_f. This is because the proportion of thermal resistance at the transfer fluid side will increase if the dimensionless thermal conductivity of the transfer fluid is decreased. To enhance the heat transfer at the main thermal resistance location is an efficient way to enhance the heat transfer in a heat exchanger.

Figure 9 shows the effect of initial subcooling on the MVF. The initial subcooling in the PCM results in a slow melting process, and the effect of internal fins is more significant if the initial subcooling exists in the PCM.

Figure 10 shows the effect of the internally finned tube structure (n, W, E) on the ratio of the heat transfer enhancement, R, at $Fo = 6$. The definition of the ratio is

$$\eta = \frac{MVF|_{\text{finned tube}}(n, W, E)}{MVF|_{\text{plain tube}}}. \tag{23}$$

As can be seen, the increase of the fin thickness resulted in a more significant increase of the MVF for higher fins. On the other hand, the increase of the fin height resulted in a more significant increase of the MVF for the thicker fins. For the thicker and higher internal fins the increase of the MVF by increasing the number of the fin is more significant.

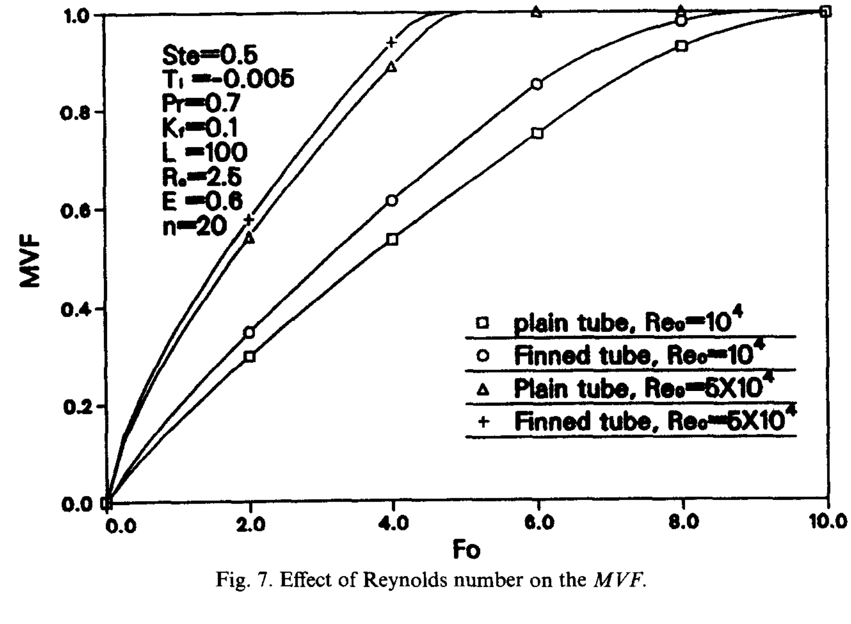

Fig. 7. Effect of Reynolds number on the MVF.

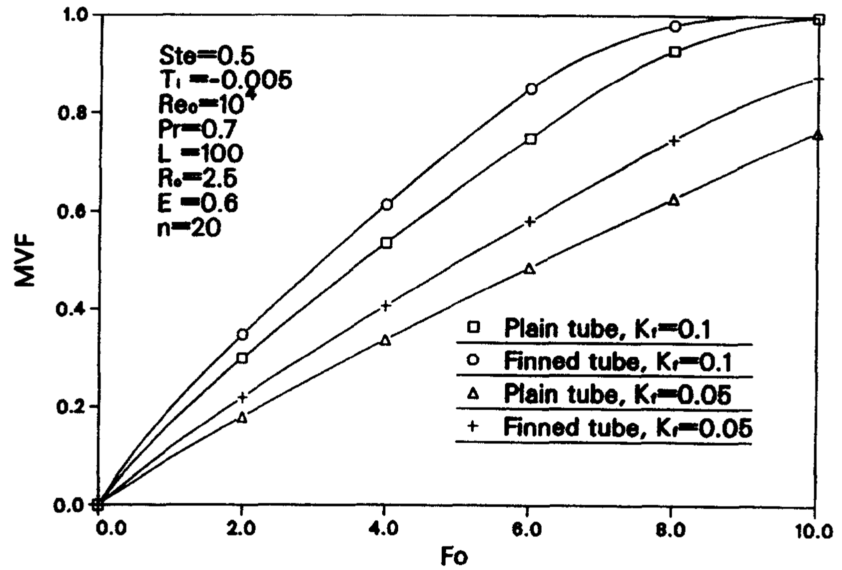

Fig. 8. Effect of dimensionless thermal conductivity of fluid on the *MVF*.

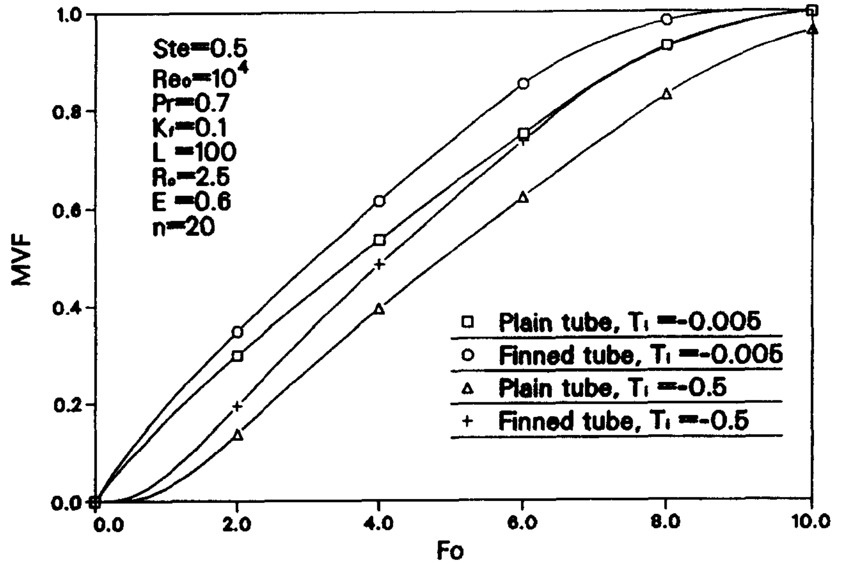

Fig. 9. Effect of initial subcooling on the *MVF*.

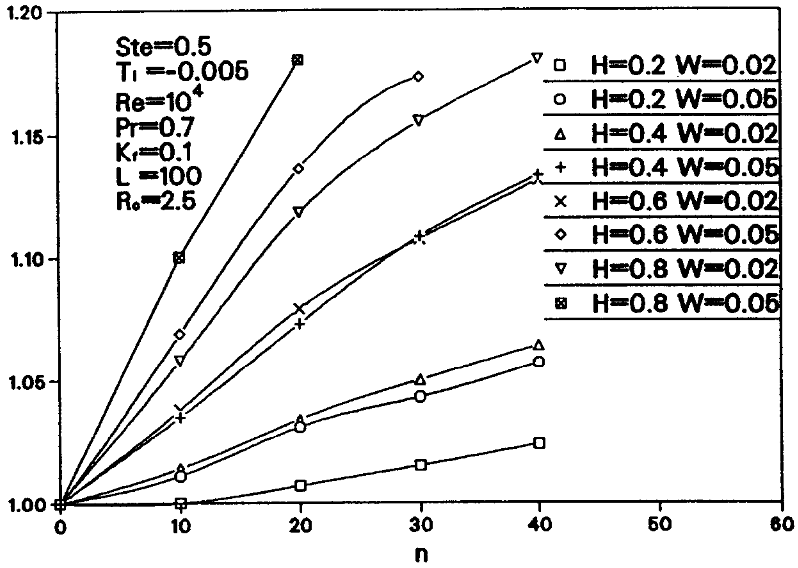

Fig. 10. Effect of the internally finned tube structure on the heat transfer enhancement ratio.

CONCLUSIONS

The heat transfer enhancement in the latent heat thermal energy storage system by using internally finned tubes was studied numerically. The comparison between plain tubes and internally finned tubes is based on the same tube diameter and flow rate of the transfer fluid. The *MVF* can be significantly increased by increasing the thickness, height and number of fins. The internal fins are a more efficient way of enhancing the melting heat transfer for a transfer fluid with low thermal conductivity at low Reynolds numbers. If initial subcooling exists in the PCM, the effect of internal fins on the enhancement of the melting process is much more significant.

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