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ANALYSIS AND PRE-CALCULATION OF A PCM THERMAL STORAGE SYSTEM

Kamal A. R. Ismail
Dept^{o} de Eng. Térmica e Fluidos, Faculdade de Eng. Mecânica-UNICAMP
C.P. 6122, CEP 13083-970 Campinas,SP, Brasil, E-mail kamal@fem.unicamp.br

Mabruk M. Abugderah
Universidade Estadual do Oeste do Paraná
Dept^{o} de Eng. Química, C. P. 520, CEP 85900-000 Toledo, PR, Brasil
E-mail mabruk@yahoo.com

ABSTRACT. *A phase change thermal storage system of the vertical tube type is studied using a fixed grid numerical model. 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 control volume finite difference approach described by (Patankar, 1980) is used to solve the equations describing the phase change thermal storage system. The radial temperature distribution, the phase change interface position and the latent and sensible heat accumulated along the system axial length are shown. It is also shown in this work a procedure for thermal performance prediction of a pcm thermal storage system.*

Key words: *Phase change material, Thermal performance, Solidification front*

1. INTRODUCTION

Phase change problems have a limited number of analytical solutions. Most of the available solutions apply to simplified and idealised one dimensional systems. Numerical methods appear to offer a more practical approach to solve a wide range of phase change problems. Over the last years many authors have been preoccupied with this type of problem solving and various techniques have been developed, but the major part of them focused on phase change problems controlled by diffusion or involving natural convection (Cao and Faghri (1991)). 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 phase change energy storage system is numerically modelled. The details of the system construction parameter are shown in Fig. (1). First the model is compared with other results obtained from the literature then it is used to analyse a two dimensional phase change storage system in cylindrical coordinates. The influence of various parameters on the performance of the system; such as the time period, Reynolds and wall materials; is carried out. It is also presented a procedure of a phase change thermal performance prediction.

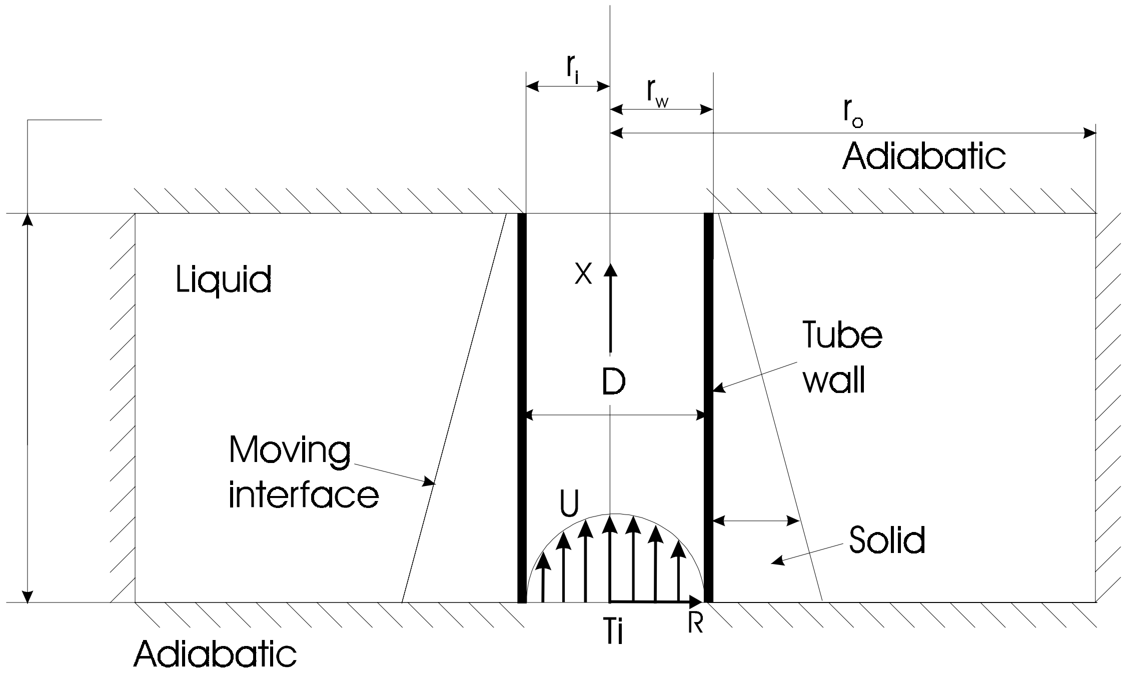

Figure 1- Schematic diagram of the PCM thermal storage system

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, 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

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

Where $K_{sl} = \dfrac{k_s}{k_l}$, and $C_{sl} = \dfrac{c_s}{c_l}$, 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: $X = 0$; $0 < R < R_i$: $\theta = 1$

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

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

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

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

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

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 phase change model was checked against other numerical results for two dimensional freezing problem in cylindrical coordinates. The obtained results 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 table (1).

Table 1. Parameters used by Cao and Faghri (1991)

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

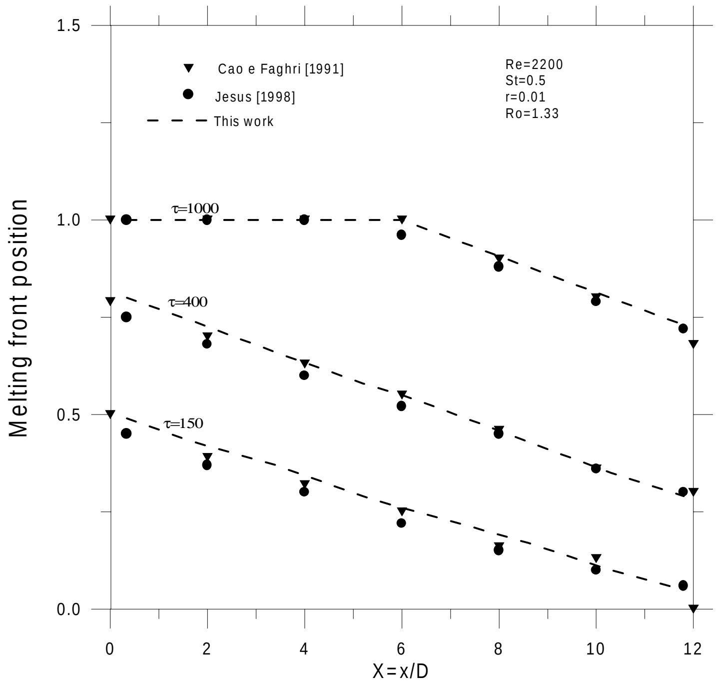

Figure 2- Solidification front position.

After checking the validity of the model, the numerical calculation for the pcm storage system was then conducted to solve a system consisting of n-Eicosane as the phase change material and water as the working fluid. The physical and the system geometrical parameters are given in tables (2-4).

Table 2. Physical properties of n-Eicosne

	ρ (kg/m^3)	c_p (J/kg.K	k (W/m.K)	λ (J/kg)
Solid	778	2010	0.15	247300
Liquid	856	2210	0.15	

Table 3. Physical properties of the working fluid (water)

ρ (kg/m^3)	c_p (J/kg.K)	k (W/m.K)	ν (m^2/s)	Pr
998	4225	0.57	0.012	7.2

Table 4. Operational and geometrical parameters

Tube internal diameter (m)	0.025
Tube wall thickness (m)	0.0015
Tube external diameter (m)	0.028
Tube length (m)	1.5
Working fluid velocity (m/s)	0.25

Initially the system was considered as liquid at its melting temperature, T_m. The heat transfer fluid inters the tubes with lower temperature than the temperature of the phase change material. The energy of the phase change material is stored as both latent and sensible heat. The stored sensible heat is that used to lower the solid phase temperature to the temperature of the HTF. The grid size used is 60 (axial) $\times$ 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 phase change temperature ε is taken to be 0.001, the system initial temperature is $-\varepsilon$, and outer radius r_o/D=2.0.

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 in the figure 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 melting interface progresses to the right indicating greater energy storage.

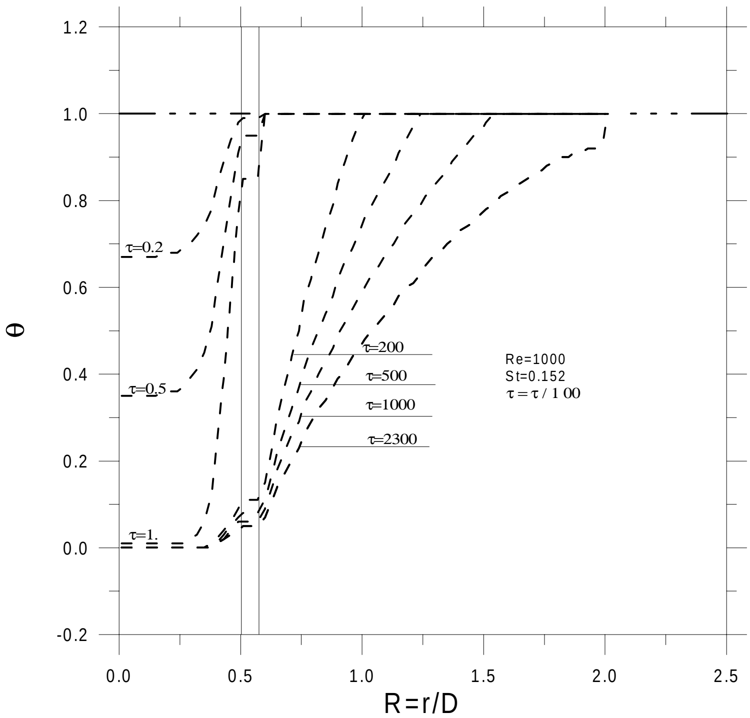

Figure 3- radial temperature distribution at the middle of the pipe.

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 $2.3\text{x}10^4$ the solidification interface has reached the outer radius of the system for X<40, while some of the pcm remains liquid for X>40. The reason for that is the high thermal conductivity which facilitates the heat 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 (5), shows the solidification front position as a function of the dimentionless axial length for different Reynolds numbers. It can be seen that for Reynolds number greater than or equal to 1500 the solidification front has reached the outer radius for X<40. It can also be noted that the influence of the Reynolds number is insignificant when passed 1500.

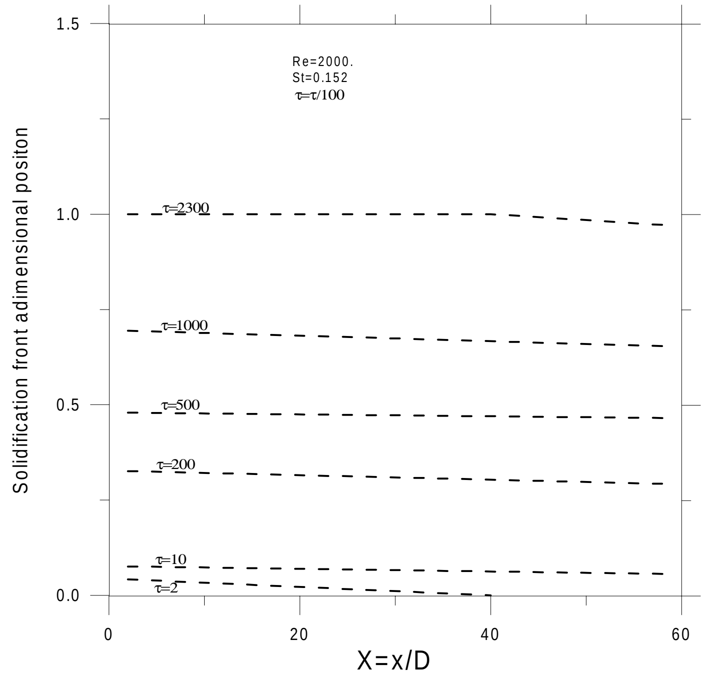

Figure 4- Location of solidification front in the axial direction

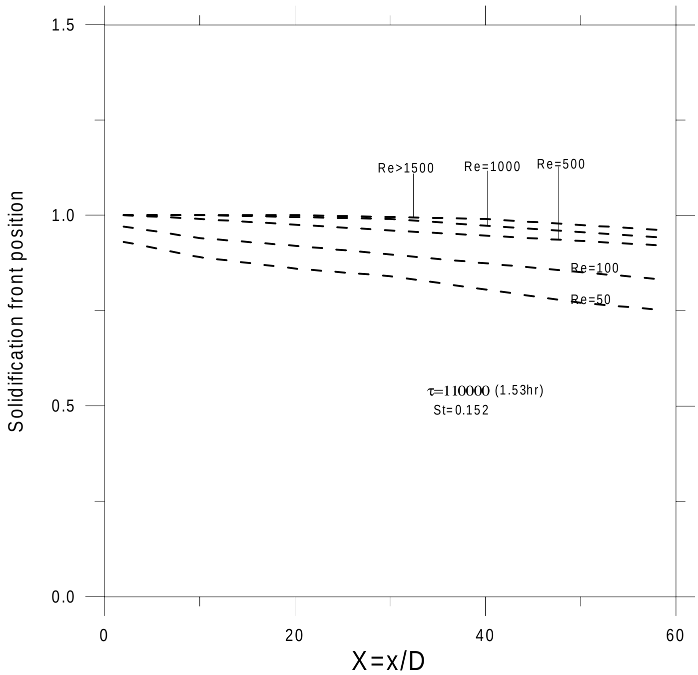

Figure 5- Position of the solidification front for different Reynolds numbers.

Figure (6), shows the influence of the tube wall material on the 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 by the pcm.

Phase change system thermal performance is a function of many parameters among others such as the Reynolds and Stefan numbers, the material of tube wall and the length of the system. It is very difficult to specify all of the operational and geometrical specifications of a phase change thermal project. In many cases some parameters can be specified by the nature of the system or by the environment. As an example, the Reynolds number can be specified by an available equipment or by an available resource if the working fluid were taken directly from the environment (for example it can be taken from a river). Will be presented as a case study the prediction of the thermal performance of a phase change thermal storage system in which n-Eicosane will be used as the phase change material and water as the working fluid. One of the most important parameters to be specified in the pre-calculation of this type of systems is the symmetry radius which would be used to calculate the necessary number of tubes to discharge a specific quantity of heat in a desired operation time cycle.

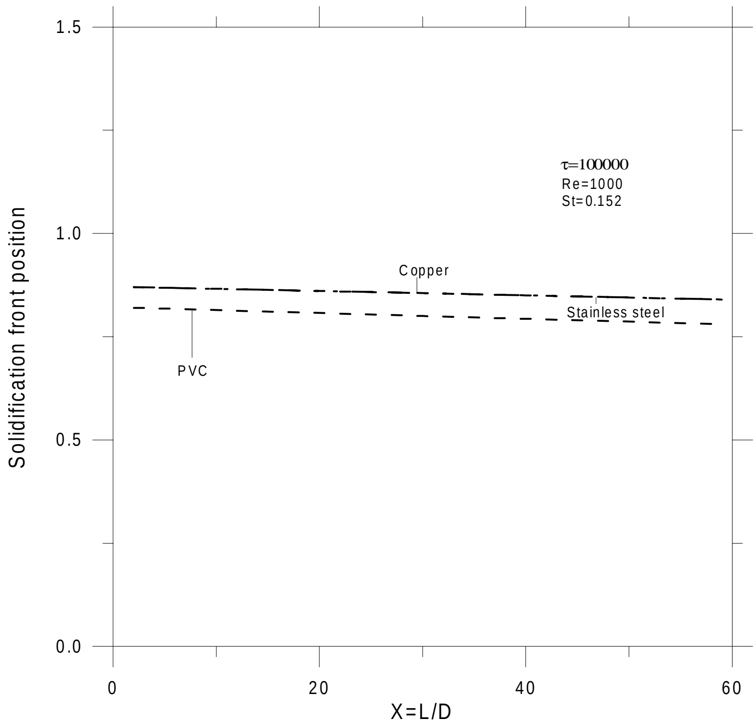

Figure 6- Position of the solidification front for different tube wall material.

3.1 Prediction of the thermal performance of a pcm thermal storage system

Most of the phase change thermal storage systems of the shell and tube type consist of more than one tube. The basic requirement in the pre-calculation of the pcm thermal storage system is the specification of the necessary tube number to discharge a certain quantity of heat. This type of analysis can be conducted in the following manner:

Given or specified parameters are:

- Maximum heat to be stored by the system;
- Storage cycle;
- Phase change material.

The Reynolds number can be obtained from a graphical method. Taking in account that the physical properties of the working fluid are constant, the Reynolds number becomes dependent on the variables: the tube diameter and the working fluid velocity. These two variables can take a range of values for a determined Reynolds number, but the specification of these variables would be tied to the available resources and economical analysis.

The phase change material is specified by the requirement of the project, and therefore it can be determined by its phase change temperature. Taking into account that the thermal properties of the phase change materials are constant, the working fluid inlet temperature can be calculated using Stefan number, which can be specified by a graphical method.

The material of the tube wall is an important parameter in the prediction and the design of the phase change storage system, 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 (7) presents the time of complete solidification versus the outer radius for different Stefan numbers. The Stefan number in this figure changes only with the inlet temperature of the heat transfer fluid, since, the properties of the HTF are considered to be constant. The Stefan numbers *0.06, 0.15, 0.20, 0.24* and *0.29* correspond to the inlet temperatures *30*, *20, 15, 10* and *5^oC*, respectively. Observe that the adequate choice of the Stefan number (which would permit the estimation of the HTF inlet temperature) would be about 0.15, because this value corresponds to the ambient temperature and the a higher value of the Stefan number does not change much the total heat stored. Other values of the inlet temperature of the working fluid require higher cost. For the given operational and geometrical parameters, the symmetry radius is found to be about *2.75D*, Fig. (7).

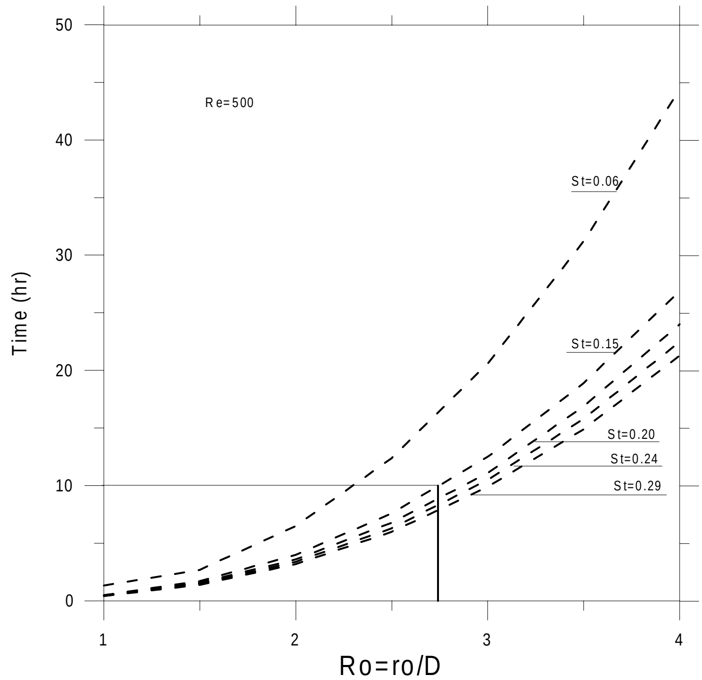

Figure 7- Time cycle versus outer radius for different Stefan numbers

The outer radius is used to estimate the total heat stored by one tube in the system from Fig. (8). The sensible heat stored in the system is calculated by the relation

$$Q_l = 2\pi D^3 c_p \rho_s \left(T_{in} - T_m\right)\int_{R_i}^{R_o}\int_0^{L/D}\left(\theta - \theta_i\right) R dR dX \tag{9}$$

and the latent heat stored in the system is calculated by the equation

$$Q_l = \pi D^3 \rho_p \lambda \int_0^{L/D} \left(R_{\text{int}}^2 - R_w^2 \right) dX \tag{10}$$

The number of tubes is calculated by dividing the total heat, desired by the phase change storage system, by the heat stored by one tube.

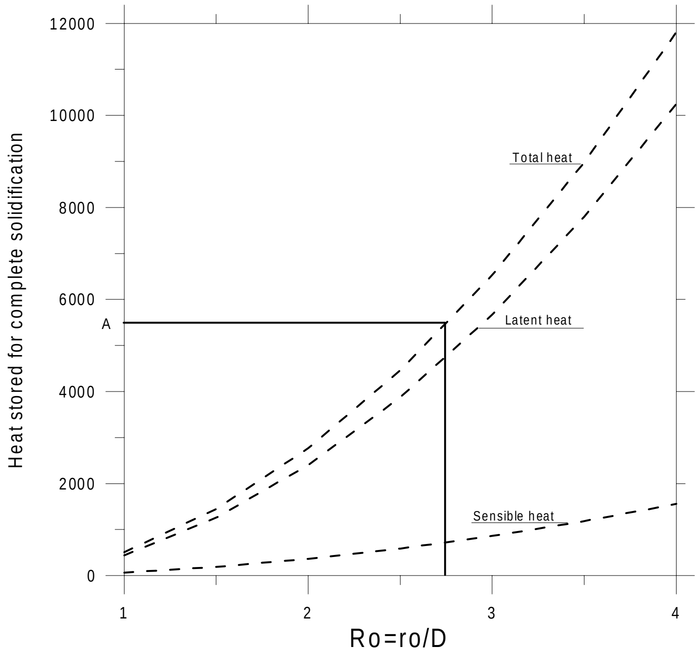

Figure 8- Heat accumulated versus the outer radius of solidification

3. CONCLUSION

A phase change thermal storage system was 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 completely developed. The influence of various parameters on the thermal performance of a phase change storage system were investigated. The most significant parameters are the Reynolds and Stefan numbers, the outer radius of solidification and the tube wall material. The prediction of thermal performance of this type of systems depends on the total heat stored, the cycle time and the other investigated parameters. Graphs were used to estimate the outer radius of solidification from which the number of tubes necessary to discharge the desired quantity of heat during the cycle time.

4. REFRENCES

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