ENCIT 2018

17th Brazilian Congress of Thermal Sciences and Engineering
November 25th-28th, 2018, Águas de Lindóia, SP, Brazil

ENCIT-2018-XXXX

MODELING AND VALIDATION OF SOLIDIFICATION OF PCM AROUND A VERTICAL BARE TUBE

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Abstract: *Renewable energy systems are usually intermittent and needs some means of accumulating and keeping the energy when available to be used when needed with minimum of losses and in adequate rate for the process. There are three basic means for thermal energy storage; sensible heat storage, latent heat storage and hybrid thermal energy storage. Latent heat thermal energy storage system is the most preferred method due to its high thermal capacity, the small mass and the fact that the processes of charging and discharging can be realized under isothermal conditions. In the present study a model is proposed to describe the solidification process along a vertical tube of constant surface temperature (or constant heat flux) submersed in a liquid PCM. The mathematical model of the phase change process over the surface of the vertical tube is based on pure conduction, the finite volume technique and the enthalpy approach are used for the numerical treatment of the mathematical model. A home-built numerical code is constructed, optimized and validated against experimental results of the interface velocity and position showing reasonably good agreement.*

Keywords: *Energy storage, latent heat, vertical tube, phase change, PCM, interface velocity, interface position*

1. INTRODUCTION

The storage of thermal energy in the form of latent heat is the most preferred method, due to the high thermal capacity, small mass and nearly constant temperature charging and discharging processes. The latent heat concept can be used to store energy as it can be used for efficient thermal insulation in homes and thermal equipment. Most of the latent heat storage materials have low thermal conductivity which increases the time for charging and discharging the storage system and hence limiting their wide application.

Gonçalves (1996) developed a two-dimensional model of the phase change problem in a thermal storage unit with axially finned tubes with constant working fluid inlet temperature and constant convective coefficient. The mechanism of heat transfer is considered as pure conduction. The enthalpic approach and the control volume method were used in the numerical solution to descretize the governing equations and the associated boundary and initial conditions. Effects of the number of fins, fin length and tube wall temperature were presented and discussed.

Yuksel,et al. (2006) proposed a theoretical approach to predicting time and temperature during charging and discharging of latent heat storage with phase change material (PCM). The analytical solutions obtained were compared with experimental data available in the literature. They show that the decrease in the inlet temperature of the working fluid from 4ºC to 15 ºC has a very strong and dominant effect on the PCM solidification time. The effect of the working fluid and PCM capsule material on the total solidification time was investigated.

Bony and Citherlet (2007) presented a numerical model with experimental validation of heat storage with phase change materials. The research describes shapes, such as cylinders, plates and beads. Comparisons between measurements and simulations were conducted to evaluate the potentiality of the model. This model, based on the enthalpy approach, takes into account conduction and convection in the PCM.

Regin et al. (2009) presented the results of a study that had as objective to analyze the behavior of the fixed bed latent heat storage system. The fixed bed is composed of spherical capsules filled with wax (paraffin) as PCM usable with a solar water heating system. The model developed in this study uses the fundamental equations, except the phase change phenomena of PCM inside the capsules are analyzed using the enthalpy method. The equations are solved numerically, and the results obtained are used to analyze the thermal performance of both loading and unloading processes. The effects of the heat input temperature of the Stefan number, heat transfer fluid, mass flow rate and phase change temperature range on the thermal performance of the capsules have been investigated.

Rösler e Brüggemann (2011) conducted a numerical study including comparison with experiments on shell-and-tube latent heat energy storage. The system is analyzed by numerical simulation and experimental measurements. To this end, the enthalpy porosity method is extended and validated by experimental measurements.

Lago et al.(2015) investigated experimentally the fusion of PCM in a spherical shell subject to constant wall temperature and obtained experimental correlations for the time for complete fusion and the involved parameters of the process . In another work Ismail et al. (2016) investigated the parameters affecting the time for complete solidification and fusion in spherical capsules and developed correlations between the time for complete phase change and the investigated parameters.

Ismail et al.(2015) presented the results of a numerical study on internally and externally finned annulus in which the internal tube has external fins while the external tube has internal fins. This arrangement was investigated with the objective of increasing the heat transfer rate, reducing the time for complete phase change and allowing for simultaneous charging and discharging processes. The proposed model was based upon pure conduction and its validity was established by comparison with available data. In another work Ismail et al.(2016) presented the results of an investigation on axially finned tubes to enhance the processes of charging and discharging thermal storage units. The proposed model was based upon pure conduction in the solid and liquid phases. The enthalpy approach and the finite volume method were used in the numerical treatment. The numerical predictions were compared with experimental results and relatively good agreement was observed.

This paper presents a model for a vertical tube with constant surface temperature (below the phase change temperature of the PCM) submersed in liquid PCM at its phase change temperature. A pure conduction model is adopted to formulate the governing equations while the finite volume approach is used to discretize the equations and the associated boundary conditions. A home-built numerical code was constructed, optimized. The numerical predictions were validated against experimental measurements showing relatively good agreement.

2. FORMULATION OF THE MODEL

The physical model consists of a vertical tube submerged in the PCM, the cold secondary heat transfer fluid circulates inside the tube and absorbs heat coming from the PCM initially in the liquid state. The problem under consideration is presented in Fig.1, where one can identify the symmetry circle and the representative domain of the problem.

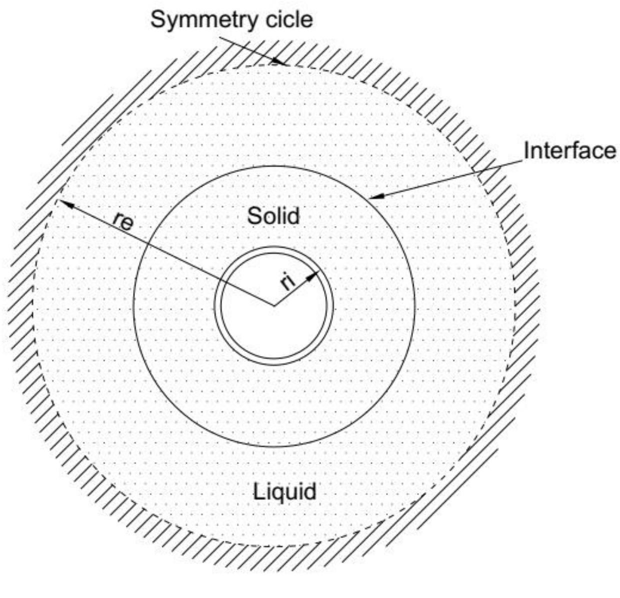

Figure 1. Dimensions of the symmetry region

Considering the process of heat transfer by pure one-dimensional conduction with no natural convection effects in the solidification process, the energy equation can be written for the whole domain and is valid for both the liquid and solid PCM. The energy equation for solid phase PCM is:

$$\rho_s c_s \frac{\partial T_s}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r k_s \frac{\partial T_s}{\partial r} \right) \tag{1}$$

The energy equation for liquid phase PCM is:

$$\rho_l c_l \frac{\partial T_l}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r k_l \frac{\partial T_l}{\partial r} \right) \tag{2}$$

The boundary conditions at the solid / liquid interface

$$\left[k_s \frac{\partial T_s}{\partial t} - k_l \frac{\partial T_l}{\partial t} \right] = \rho_S L \frac{\partial r_S}{\partial t} \tag{3}$$

The initial and final conditions can be written as:

$$T(r,t=0) \geq T_m^+ + \Delta T \tag{4}$$

$$T(r,t=t_f) \leq T_m^- - \Delta T \tag{5}$$

The specific heat values and latent heat per volume unit are defined:

$$C_S = \rho_S c_S \tag{6}$$

$$C_L = \rho_L c_L \tag{7}$$

$$\lambda = \rho_S L \tag{8}$$

For facilitate the numerical calculations, Eqs. (1, 2) and the initial and final boundary conditions were put in dimensionless forms by using the these new dimensionless variables :

$$\theta = \frac{T - T_m}{T_m} \quad ; R = \frac{r}{(r_i^2)^{1/2}}; Fo = \frac{k_s t}{C_s r_i^2}; \tilde{C} = \frac{\overline{C}}{C_s}; \tilde{K} = \frac{\overline{k}}{k_s}; Ste = \frac{C_s(T_m^+ - T_{bi}(0))}{\lambda}; Bi = \frac{hr}{k} \tag{9}$$

According Bonacina et al. (1973) approach the governing equations and the associated boundary, the initial and final conditions it was to possible to cast the model equations in the form as in Eqs. (10,11) expressed in terms of the new variables as:

$$\tilde{C}(\theta)\frac{\partial \theta}{\partial Fo} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\tilde{k}(\theta)\frac{\partial \theta}{\partial R} \right) \tag{10}$$

$$\theta = 1 \quad for \quad Fo = 0; \quad \theta = 1 - 2\xi \quad for \quad Fo = Fo_f \tag{11}$$

For the case of constant tube wall temperature the boundary conditions in terms of the new variables are given by:

$$\left. \frac{\partial \theta}{\partial r} \right|_{R=1} = 1 \tag{12}$$

Symmetry circle:

$$\left. \frac{\partial \theta}{\partial r} \right|_{R=\frac{r_e}{r_i}} = 0 \tag{13}$$

For the case of constant heat flux the condition boundary considering the constant flow in the surface of the vertical tube. Boundary condition at the wall of the tube considered constant flow dimensionless variables:

$$\left. \frac{\partial \theta}{\partial r} \right|_{R=1} = Bi\theta \tag{14}$$

Symmetry circle:

$$\left.\frac{\partial \theta}{\partial r}\right|_{R=\frac{r_e}{r_i}} = 0 \tag{15}$$

The set of equations of the model and the boundary and initial conditions were implemented in a home-built computational code. Numerical tests were conducted to ensure that the predictions are independent of the choice of the grid size. The number of grid points along the radial direction of the vertical tube is 1000 points and the time step is taken as 10^{-6} s. These values are used in all the numerical simulations.

3. ANALISYS EXPERIMENTAL

Figure 2 shows the layout of the experimental test bench for conducting the experiments on the solidification of PCM (water) around the bare vertical tube. The test rig is composed of a compression refrigeration system composed of a vapor compression refrigeration unit, condenser, expansion device and heat exchanger which is responsible for cooling the working fluid (Ethanol) to the desired temperature for the test. The test section where the vertical bare tube is connected to the secondary fluid circuit is made acrylic cylinder of 15 mm thickness, 300 mm in diameter and 260 mm in height with both the top and bottom made of thick acrylic sheet of 20 mm thickness. The cold Ethanol enters the test tube at the bottom and leaves at the top back to the secondary fluid tank and back to the heat exchanger where it is cooled back to the preset temperature. The primary refrigerant from refrigeration compressor passes through the coil immersed in the heat exchanger where the Ethanol is placed in the shell of the heat exchanger. The mass flow of the secondary fluid is adjusted by a control valve and measured by a calibrated orifice plate where the pressure difference is measured by a calibrated differential pressure manometer. Following the solidified mass around the test tube is done by a high resolution digital camera where photographs of the tube are taken at the preset intervals. A precision scale is fixed very close to the tube to serve as a reference for tracking the real position of the solidification front. Calibrated thermocouples type T, are fixed at entry and exit of the vertical tube, in the PCM test section (tank), in the tank of the secondary fluid.

The thermocouples were calibrated to within ±0.5 ºC, the image conversion precision is done to within ±0.5mm and the mass flow rate of the secondary fluid (Ethanol) was calibrated to within ± 10^{-4}kg/s. Error propagation analysis is also done.

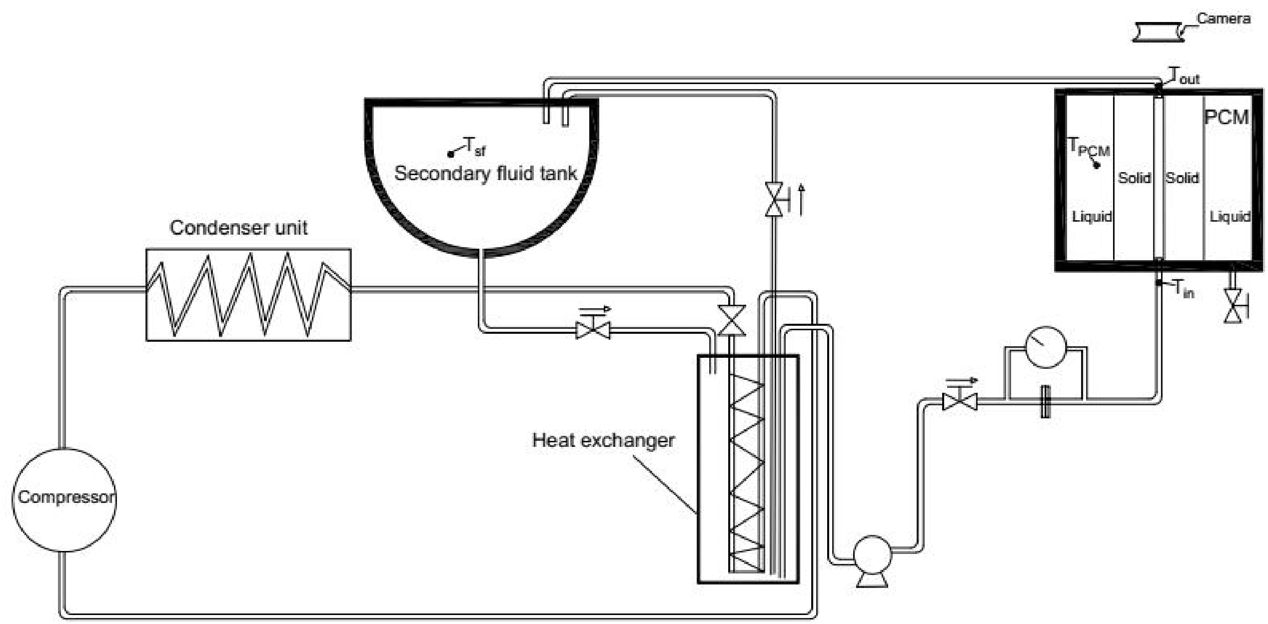

Figure 2. Experimental workbench layout

Measurements were taken when the desired testing conditions were achieved, that is the temperature of the working fluid to flow through the vertical tube, temperature of the Ethanol tank, temperature of the PCM in the test section (tank). Achieving these conditions the different measurements are registered and the test is started immediately. During the first two hours each 5 minutes period all measurement points are registered and a photograph of the bare tube is taken. During the third and fourth hours measurements are registered each 30 minutes interval. After that the time interval is increased to 50 minutes until the end of the test. The test is considered terminated when no significant variation of the interface position is observed during three successive time intervals. A typical photograph of the vertical tube is shown in Fig. 3 where the interface position is tracked and converted to real dimension by using the program Tracker

Figure 3. Photograph of the vertical tube with solidified PCM around it

4. RESULTS AND DISCUSSIONS

Experimental measurements realized on the bare tube submersed in liquid PCM show that the variation of interface position with time for three different wall temperatures as can be seen in Fig. 4. One can observe that decreasing the wall temperature increases the interface position, that is, more PCM is solidified. This is due to the fact decreasing the wall temperature increases the temperature difference which provokes more heat transfer from the tube surface to the PCM and hence increasing the interface position.

Figure 5 shows the variation of the interface velocity with time for different temperatures of the tube wall. As can be seen initially the interface velocity is extremely high due to low thermal resistance between the tube wall and the surrounding PCM. With time a layer of solid PCM is formed and this increases the thermal resistance causing the interface velocity to decrease. This process continues until the solidified mass is too large increasing consequently the thermal resistance to the extent that there is virtually no advance in the interface position due to the extremely low interface velocity.

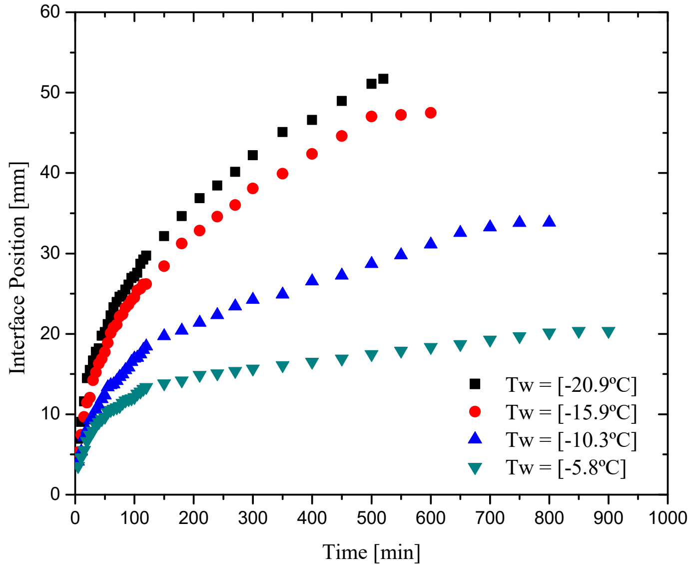

Figure 4. Experimental variation of the interface position with time

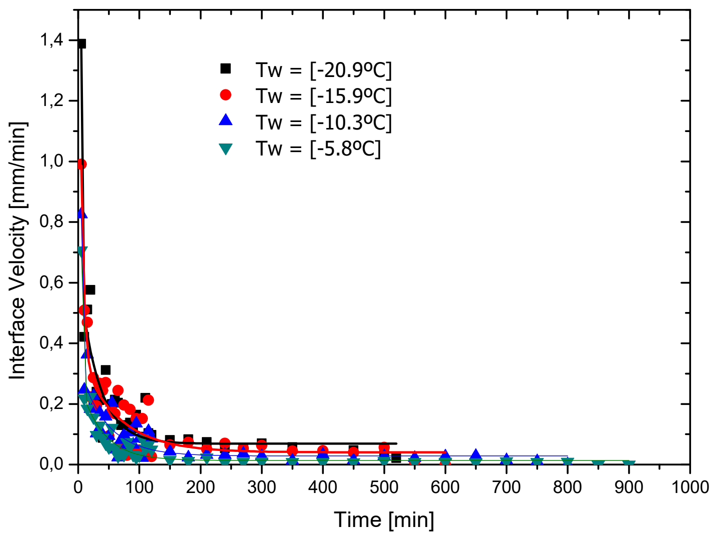

Figure 5. Experimental variation of the interface velocity with time

In order to validate the home-built numerical code and the numerical predictions experiments were conducted by the authors for the case of constant wall temperature and constant heat flux. Figures 6 and 7 are presented for the case of constant wall temperature. As can be seen the agreement is very good indicating that the numerical code can predict with good precision the interface position of the solidified PCM around the bare tube.

Figure 7 shows the variation of the interface velocity with time compared with the numerical prediction. As one can observe the agreement is extremely good confirming the adequacy of the numerical code for predicting the interface velocity with good precision.

Figures 8 and 9 are for the case of different wall temperature. Observe that the dimensionless time is calculated according to equation (9).

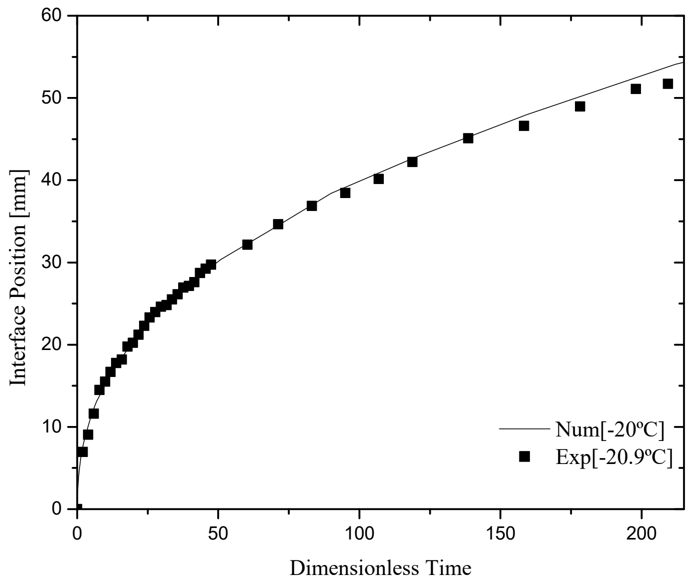

Figure 6. Comparison of the experimental and numerical interface positions for the case of Tw=-20 ºC

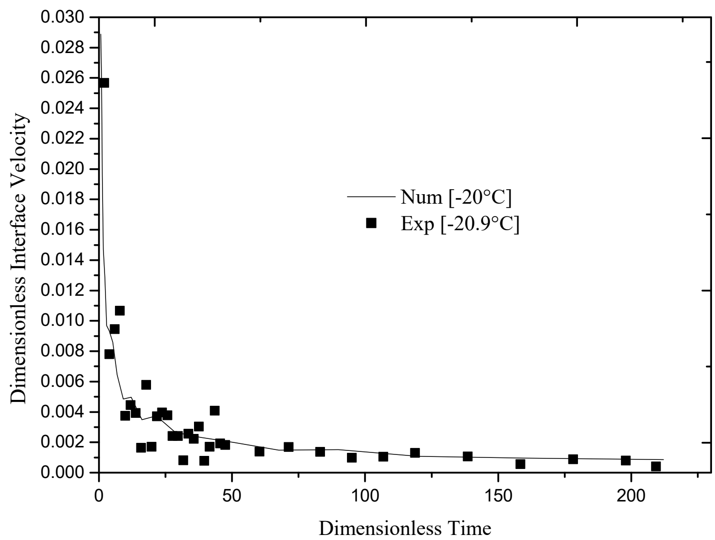

Figure 7. Comparison of the experimental and numerical interface velocity for the case of Tw=-20 ºC

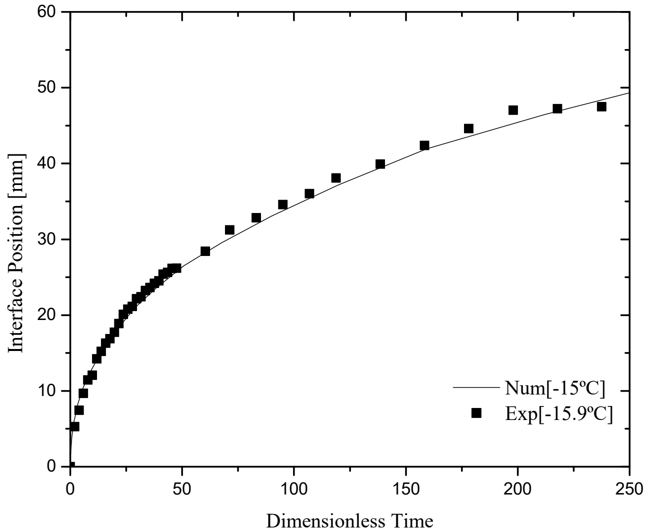

Figure 8. Comparison of the experimental and numerical interface position for the case of Tw=-15 ºC

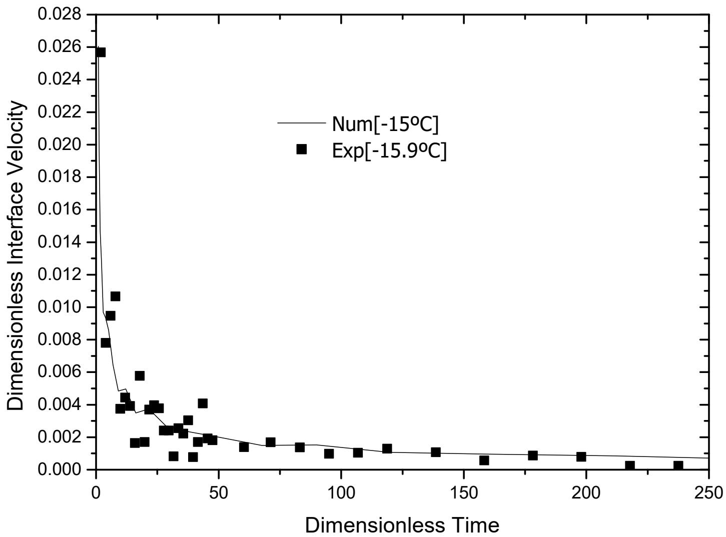

Figure 9. Comparison of the experimental and numerical dimensionless interface velocity for the case of Tw=-15 ºC

In order to validate the home-built numerical code and the numerical predictions experiments were conducted by the authors for the case of constant heat flux as in Figures 10 and 11. As can be seen the agreement is good although the Biot number is relatively small due to experimental limitations. Nevertheless the agreement indicates that the numerical code can predict with good precision the interface position of the solidified PCM around the bare tube. Figure 11 shows comparison between experimental measurements and the predicted results for the case of constant heat flux. Again as can be verified the agreement is good over most of the time range. Figures 12 1nd 13 are plotted for different value of wall temperature.

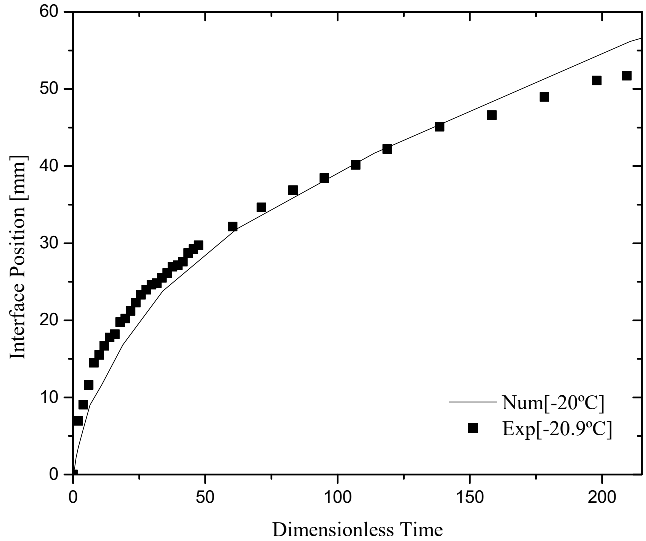

Figure 10. Comparison of the experimental and numerical interface position for wall temperature of Tw=-20 ºC for the case of constant heat flux

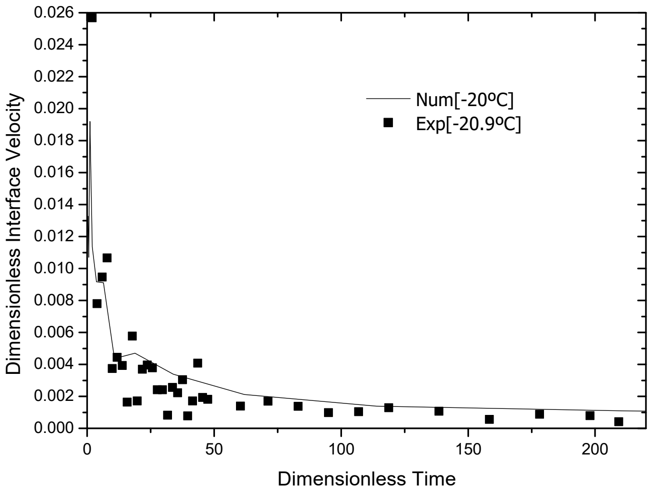

Figure 11. Comparison of the experimental and numerical interface velocity for wall temperature of Tw=-20 ºC for the case of constant heat flux

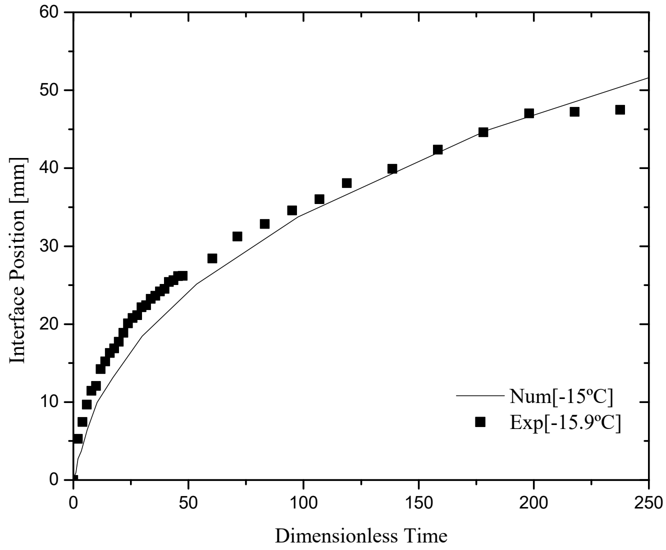

Figure 12. Comparison of the experimental and numerical interface position for wall temperature of Tw=-15 ºC for the case of constant heat flux

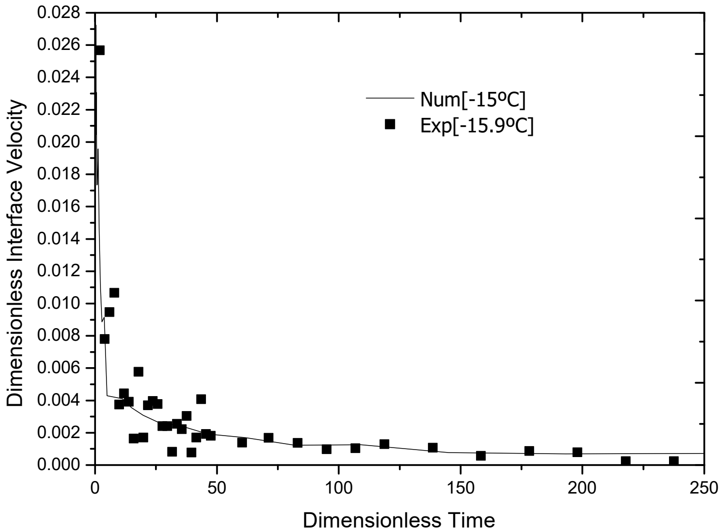

Figure 13. Comparison of the experimental and numerical interface velocity for wall temperature of Tw=-15 ºC for the case of constant heat flux

5. CONCLUSION

In this study a thermal model is developed for the phase change around vertical bare tube under constant wall temperature and constant heat flux conditions. The home-built numerical code was optimized and validated against experimental results indicating good agreement and confirming its adequacy for predicting the phase change characteristics of PCM around vertical bare tube.

6. ACKNOWLEDGEMENTS

The first author wishes to thank FAPEMA for the study scholarship and the second author wishes to thank the CNPQ for the PQ research grant.

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