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EXPERIMENTAL AND NUMERICAL INVESTIGATION OF LATENT HEAT ENERGY STORAGE IN RADIAL FINNED TUBES SUBMERSED IN PCM

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Abstract. *This paper presents the results of a numerical and experimental investigation realized on finned tubes submersed in the liquid PCM while a cold fluid at lower temperature flows inside the finned tube with the objective of using them in thermal storage systems. The model is based upon the pure conduction mechanism of heat transfer, the enthalpy formulation approach and the control volume method. A home-built numerical code is developed, tested, optimized and validated against experimental results to predict the interface position, interface velocity and the time for complete phase change. The numerical predictions were validated against experimental data produced in Thermal Storage and Heat Pipes Laboratory, on the Energy Department of the Faculty of Mechanical Engineering of the University of Campinas (Unicamp) and an agreement of approximately 96% in the final stages was found. Fins are found to increase the interface position, solidified PCM mass, and interface velocity and to decrease the time for complete phase change of the PCM.*

Keywords: *Phase change, energy storage, finned tube, interface velocity, phase change time.*

1. INTRODUCTION

In latent heat storage systems, the phase transition of the storage material from liquid to solid is used to save thermal energy. The materials whose phase change from solid to liquid is used for storing thermal energy are known as phase-change materials (PCMs). Latent heat storage systems are characterized by high volume-specific storage densities at a narrow temperature interval. The majority of the thermal energy is stored at a constant temperature. The constant storage temperature allows for a good adaptability for optimal heat transfer to the consumer. One of the most common methods used to improve thermal performance of these systems is the use of finned tubes as the heat transfer elements. These fins can be axial or radial and are usually attached to the tubes.

Ismail *et al* (2001) realized a numerical and experimental investigation in finned tubes with the objective of using them in thermal storage systems. The results confirm the importance of the fins in delaying the undesirable effects of natural convection during the phase change processes. Also, this study indicates the strong influence of the annular space size, the radial length of the fin and the number of fins on the solidified mass fraction and the time for complete phase change.

Medrano *et al* (2009) investigated experimentally the heat transfer process during melting (charge) and solidification (discharge) of five small heat exchangers working as latent heat thermal storage systems. They used commercial paraffin RT35 as PCM filling one side of the heat exchanger and water circulates through the other side as heat transfer fluid. Results show that Reynolds number in the turbulent regime is desirable for faster phase change process reduce the phase change time in about half.

Jmal *et al* (2015) studied the solidification of PCM (Paraffin C_{18}) in thermal storage coaxial tubes with internal and external fins for conditioning systems with two air passages. The numerical approach aims to study the impact of natural convection, occurring in the liquid phase, on the solidification time of PCM and the temporal evaluation of the solidification front. They concluded that in the presence of fins, energy extraction from PCM to airflow occurs at faster rate, which contributes to the reduction of the discharging time and the increase of the outlet air temperature. However, for a great number of fins (9 fins), the enhancement of the solidification process is not significant because of the effect of enhancement of PCM liquid spaces on the development of thermoconvective flow.

Kabbara *et al* (2016) conducted an experimental study on a latent heat storage system (LHESS) consisting of a tank filled with phase change material (PCM). The study included charging experiments under controlled experimental conditions with parametric alterations on the heat transfer fluid (HTF) flow rate and inlet temperature. The characterization of the LHESS showed that increasing the HTF inlet temperature during charging resulted in significantly faster melting time. The increase of flow rate did not have a significant impact during the discharge process, which can be attributed to the conduction dominant heat transfer during solidification.

Wang *et al* (2016) investigated the parameters that affect the performance of a phase change thermal energy storage (PCTES) unit using circular finned tubes. The performance of PCTES unit using circular finned tubes is best when water is used as heat transfer fluid (HTF). When the fluid flow of HTF is in a laminar state, the energy efficiency ratio and the heat storage rate are larger than that in a turbulent state. With increasing inlet velocity of HTF the energy efficiency ratio and the heat storage rate of PCTES unit change in opposite direction. The result of compromising is that when the fluid flow of HTF is laminar, the performance of PCTES unit is better.

Caron-Soupart *et al* (2016) investigated an alternative experimental performance analysis method proposed and based on the energy calculation thanks to a thorough instrumentation on the PCM side. This method enables to calculate the storage density but also the heat exchange power by defining a dimensionless characteristic time as the time necessary to store 90% of the maximal storable energy in the system. Experiments have been carried out on a single stainless steel tube, a steel tube with longitudinal fins and a copper tube with helical fins to validate the technique. The implementation of the method for the charging process demonstrated the interest of increasing the heat exchange surface to enhance the heat exchange power. The dimensionless parameters are proposed as a tool to compare some heat exchangers whatever the experimental conditions (initial and final temperatures) and the geometries (fins or not, PCM volume).

In this study, a model for the solidification of PCM around a radial finned tube with constant wall temperature is developed and solved numerically. The model is based upon pure heat conduction formulation and the enthalpy method. The finite difference approach and the alternating direction implicit scheme were used to discretize the system of equations and the associated boundary, initial and final conditions. A home-built numerical code is developed and optimized, validated and then used to investigate the effects of the diameter of fin and the tube wall temperature on the interface position, the interface velocity, the solidified mass fraction and the time for complete phase change.

2. PROBLEM MODELING

The problem under consideration can be represented by a horizontal tube fitted with external radial fins and submersed in liquid phase change material (PCM) at its phase change temperature. A circulating cold fluid flows along the tube forming a solid layer of PCM over the tube and fins surfaces. It is required to determine the effects of the fin on the phase change parameters of this process. To be able to handle this problem assume some simplifying assumptions including that the PCM is pure and of well-defined temperature varying properties, constant surface temperature over the tube and heat transfer process dominated by conduction.

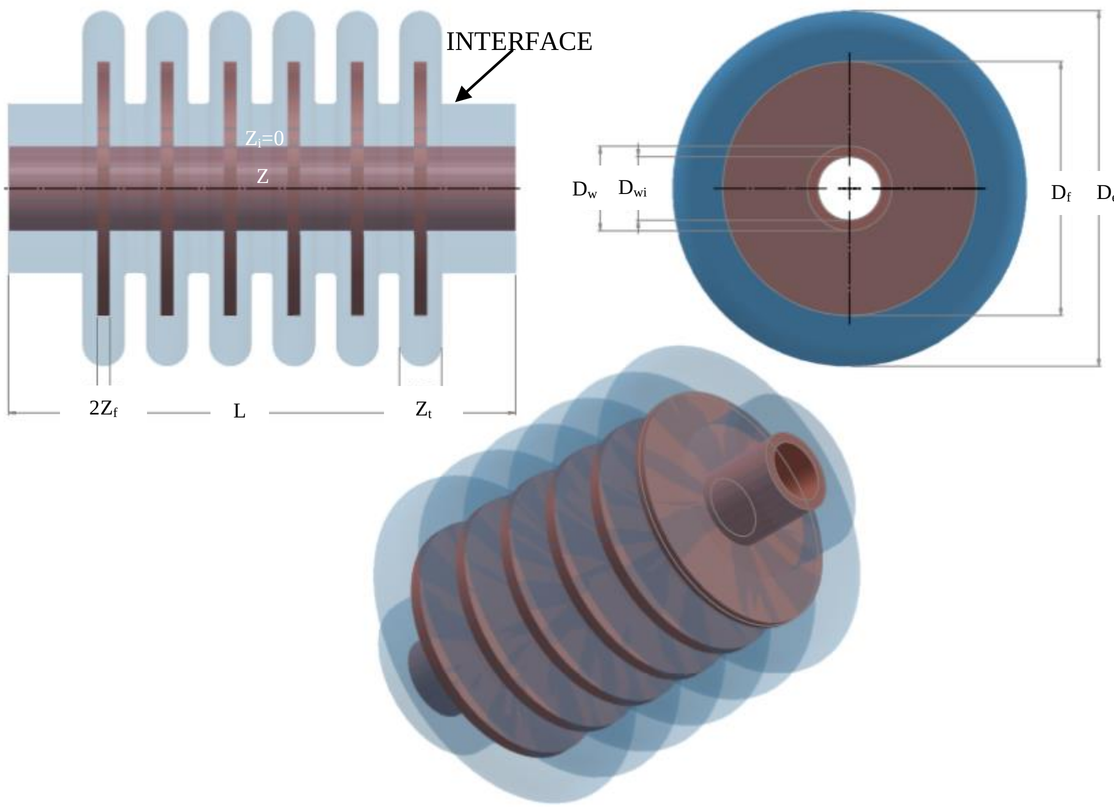

Figure 1. Layout of the problem, where: D_e = diameter of symmetry, D_f = diameter of fin, D_w = diameter of tube, D_{wi} = inner tube diameter, L = length of the tube, z = symmetry, z_i = initial symmetry, z_t = final symmetry, z_f = fin symmetry

The energy equation in cylindrical coordinates for the PCM solid phase is:

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

The energy equation for the PCM liquid phase is:

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

The boundary conditions at the interface can be written as:

$$\left(k_s \frac{\partial T_s}{\partial r} - k_l \frac{\partial T_l}{\partial r} \right)\left(1 + \left(\frac{\partial s}{\partial z} \right)^2 \right) = \rho_s L \frac{\partial s}{\partial t}; \qquad r = s(t) \quad (3)$$

$$\begin{array}{lll} T_s = T_l = T_m & & r = s(t) \\ At & \mathrm{r} = \mathrm{r_w}; & T = T_w \\ At & \mathrm{r} = \mathrm{r_e}; & \dfrac{\partial T}{\partial r} = 0 \\ At & \mathrm{z} = \mathrm{z_i} = 0; & \dfrac{\partial T}{\partial z} = 0 \\ At & \mathrm{z} = \mathrm{z_t}; & \dfrac{\partial T}{\partial z} = 0 \end{array} \quad (4)$$

The initial and final conditions can be written as:

$$\begin{array}{lll} T(r,z,t=0) & = & T_m + \Delta T \\ T(r,z,t_f) & = & T_m - \Delta T \end{array} \quad (5)$$

where ΔT is half of the phase change temperature range.

The above phase change problem is solved numerically following the enthalpy approach adopted in Ismail, *et al.*, 2001. The above model based upon pure heat conduction formulation is treated by the enthalpy method, while the finite difference approach and the alternating direction implicit scheme were used to discretize the system of equations and the associated boundary, initial and final conditions. The details are omitted here for brevity. The set of equations of the model and the boundary and initial conditions were implemented in a home-built computational code which was thoroughly tested and optimized. The input parameters used are finned copper tube with 1 m length and 15 mm tube diameter, the thickness of the fins is 3 mm, the spacing between the fins is 60 mm and the PCM is water and the range of phase change is 0.1°C. Numerical tests were realized to ensure that the results are independent of the number of grid points. The simulations were realized for number of grid points of 100 and for convergence precision of 0.0001. Figure 2 shows the simulated mesh test according to the parameters adopted, where D_e/D_w is the ratio between the diameter of symmetry and the diameter of tube, and D_f/D_w is the ratio between the diameter of fin and diameter of tube.

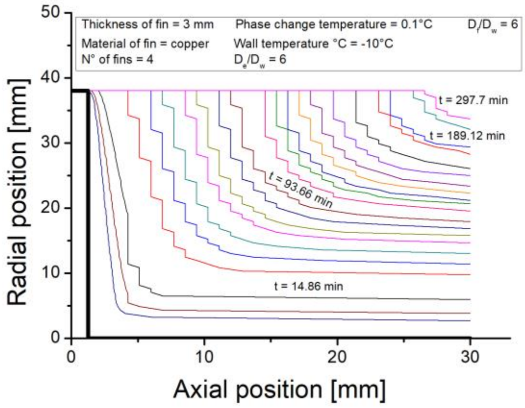

Figure 2. Solid - liquid interface for different solidification times

3. EXPERIMENTAL RIG AND TEST PROCEDURE

The general scheme of the experimental system is shown in Figure 3 and 4 are composed of a compression refrigeration circuit and a secondary circuit for cooling the working fluid (Ethanol). The test set up is composed of a compression refrigerant circuit, secondary fluid circuit, coiled tube heat exchanger submersed in the secondary fluid tank, the test section of the finned tube which is connected to the secondary fluid circuit. The secondary fluid is ethanol cooled by the refrigerant flowing through the coiled tube heat exchanger and its temperature and mass flow rate are controlled as required.

The test section is of rectangular shape built from 15 mm thick acrylic sheet with the test tube extended across the test section filled with PCM (water) whose initial temperature can be varied as desired. High resolution digital camera is used to photograph the finned tube and the reference scale. The reference scale is used to convert the image dimensions to real values. Calibrated thermocouples type T, are fixed at inlet and outlet of the finned tube, in the PCM test tank, along the finned tube and in the secondary fluid tank. Calibration of thermocouples and orifice plate were realized and error analysis and propagation in the results were done and the final results indicate a calibration error in the thermocouples of ± 0.5 °C, image conversion precision of ± 0.1mm while the mass flow rate (measured by a calibrated orifice plate) of ± 10^{-4} kg/s.

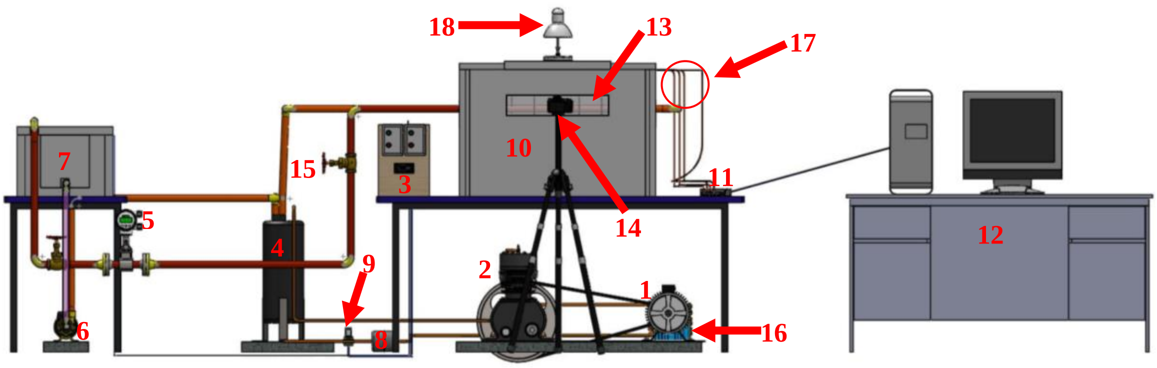

Figure 3. Test bench scheme. 1:Electric motor; 2:Compressor; 3:Set point; 4:Heat exchanger; 5:Flow meter; 6:Pump; 7:Alcohol tank; 8:Oil filter; 9:Solenoid valve; 10:Test section; 11:Signal acquisition board; 12:Computer; 13:Finned tube: 14:Digital camera; 15:Valve; 16:Condensing unit; 17:T-type thermocouples; 18:Lamp

Figure 4. Real test bench scheme of refrigeration system. (a) 1:Condensing unit; 2:Pump; 3:Compressor. (b) 1:Alcohol tank; 2:Set point; 3:Test section; 4:Pump; 5:Digital camera; 6:Heat exchanger; 7:Orifice plate

Measurements were usually taken when the desired testing conditions were achieved, that is the temperature of the working fluid in the test finned tube, temperature of the Ethanol tank, temperature of the PCM, and the mass flow rate of the secondary fluid. Under these initial conditions the chronometer is started after registering all initial conditions. During the first hour each 2 minutes period all the readings of the measurement points are registered and a photograph of the finned tube is taken. During the second and third hours measurements are registered each 15 minutes interval. After that, the time interval is increased to 30 minutes until the end of the test. The test is considered finished when no change in temperature or interface position is registered along three successive time intervals. The interface position is tracked and converted to real dimension by using the program Tracker and the reference scale as shown in Figure 5.

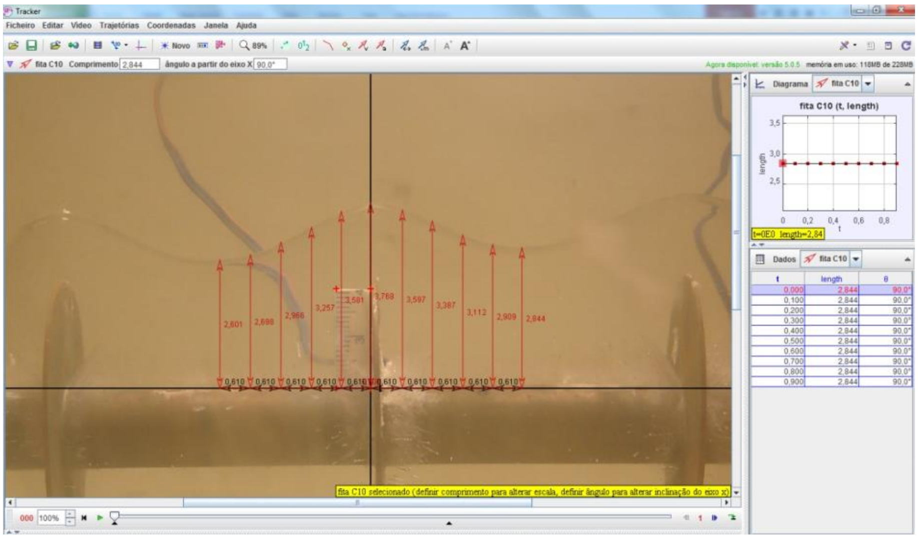

Figure 5. Tracker software with the finned tube positioned for digitalization of the interface position

4. RESULTS

Some of the experimental and numerical results obtained for the finless and finned tube will be presented in this section. The Figure 6 show the effects of incorporating radial fins external to the tube presented for the interface position after ten hours of tests. We can observe the increase of interface position due the increase of heat transfer area caused by fins as well the decrease of the wall temperature increases the position of the interface. We can also observe that the reduction of the wall temperature and hence increases the position of the interface enhancing the mass of formed ice as can be seen in Figure 7.

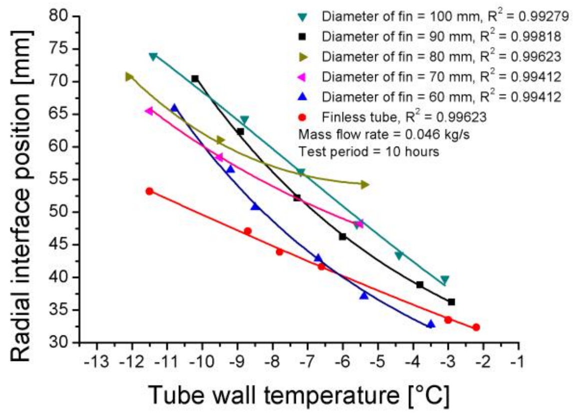

Figure 6. Comparison of the radial interface position for finned and finless tube after 10 hours of test.

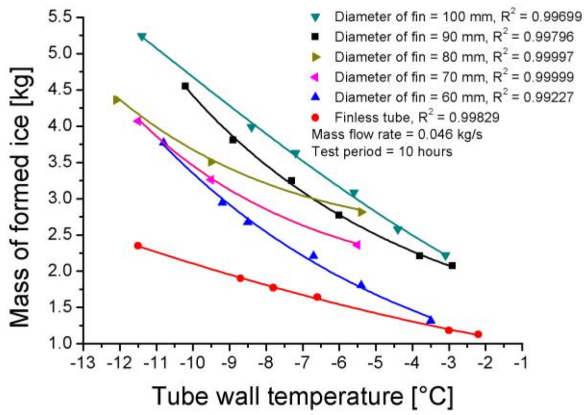

Figure 7. Comparison of the mass of formed ice for finned and finless tube after 10 hours of test.

The interface velocity is also enhanced due to incorporating fins on the tube where the cold ethanol is flowing. One can also observe that reducing the tube wall temperature increases the interface velocity due to the increase of the thermal gradient between the increased finned tube surface and the PCM as can be seen in the Figure 8. Similar results are found for other time intervals and are ommited for the sake of brevity.

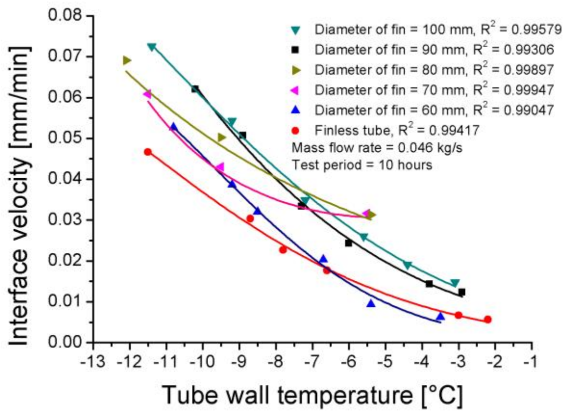

Figure 8. Comparison of the interface velocity for finned and finless tube after 10 hours of test.

Figure 9 shows the variation of formed mass with the increase of the mass flow rate of the ethanol and for three different geometries of tubes at wall temperature of -10°C. As can be seen in Figure 7 the decrease of the tube wall temperature increases the mass of the formed ice due to the increase of the temperature gradient between the surface of the tube wall and the PCM surrounding the tube. Also one observes the increase of the formed ice with the increase of the mass flow rate which causes the increase of Reynolds number and hence the internal heat transfer coefficient. The stored energy can be evaluated using the results shown in Figure 10 where the sensible heat was ignored because it is very small in comparison with the latent heat.

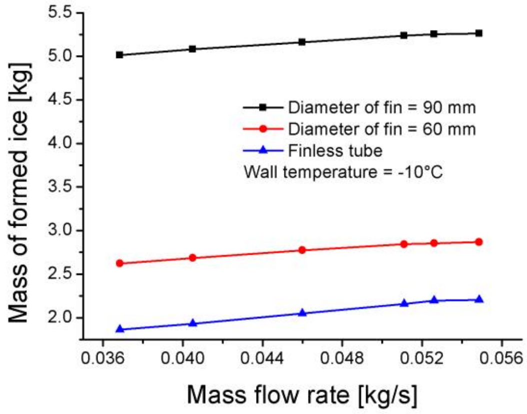

Figure 9. Variation of mass of formed ice with mass flow rate for three different tubes

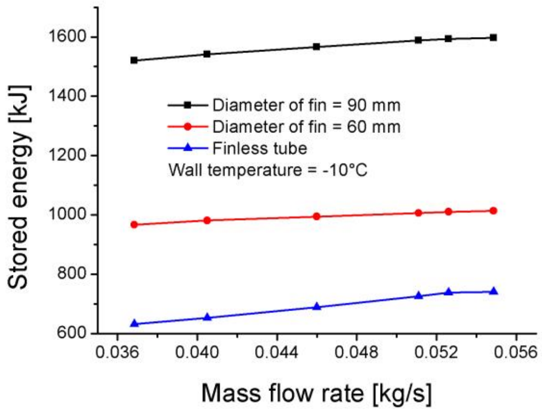

Figure 10. Variation of stored energy with mass flow rate for three different tubes

Figure 11 shows the variation of the time for complete phase change with the mass flow rate of the ethanol and with the wall temperature of -10°C. As can be seen the increase of the mass flow rate reduces the time for complete phase change (solidification of PCM) due to the increase of the internal heat transfer coefficient as mentioned before.

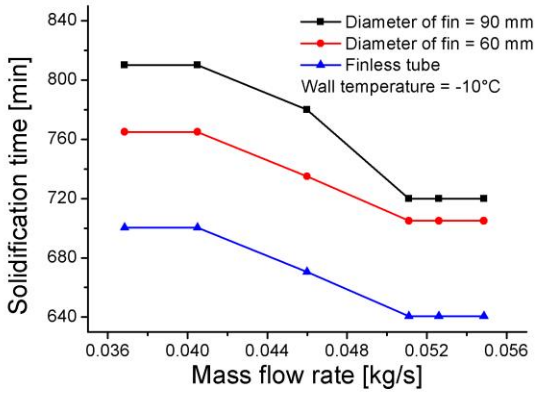

Figure 11. Variation of the time for complete solidification with mass flow rate for three different tubes

Figure 12 shows the variation of the radial interface position with time for all tubes investigated in this work, wall tube temperature ranges were used as close as possible to obtain a good comparison. As can be seen the gradient of the interface position with respect to time continuously decreases due to the increase of the thermal resistance between the tube wall and the PCM around the tube until finally reaches almost zero.

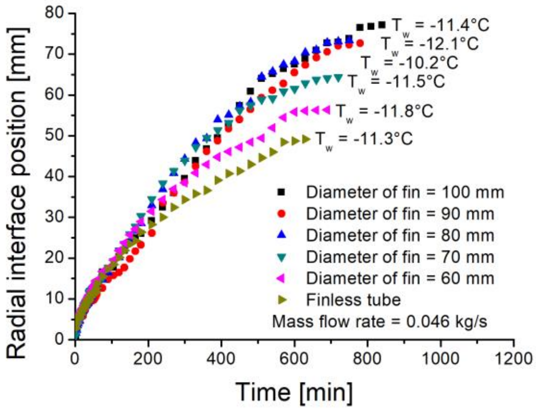

Figure 12. Variation of the radial interface position with time for different diameter of fins

Figure 13 shows the variation of the interface velocity for the same conditions as in Figure 12. One can observe that the interface velocity decreases with time due to the increase of the thermal resistance. Also the increase of the mass flow rate increases the interface velocity but the differences are too small due to the limited mass flow rate realized in the tests.

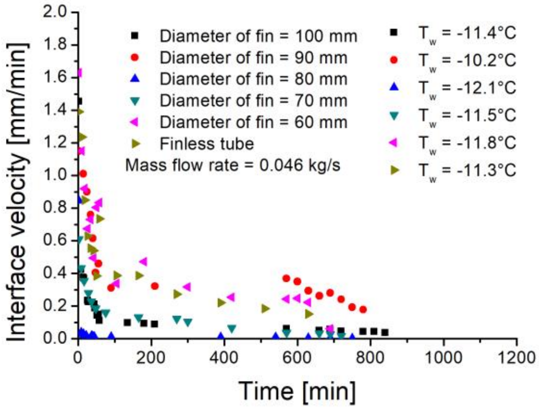

Figure 13. Variation of the interface velocity with time for different diameter of fins

Figure 14 shows a comparison between the predicted interface position and experiments for the case of finned tube. The agreement is relatively good except in the initial stages where the difference can be attributed to small calibration errors by Tracker, the scale used to obtain the measurements and by air inclusions caused by the crushed ice used to cool the water that remains attached to the surface of the tube, these inclusions decrease as the experiment time goes on until but ice growth in the early stages is lower. Similar effects are found in Figure 15 of the interface velocity where in the initial itervals the numerical predictions overestimate the interface velocity.

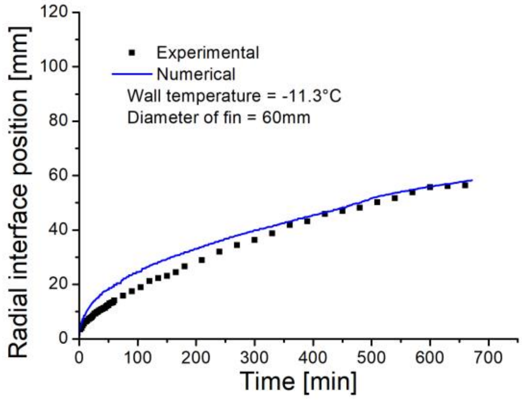

Figure 14. Comparison of the predicted interface position with experiments for the case of finned tube.

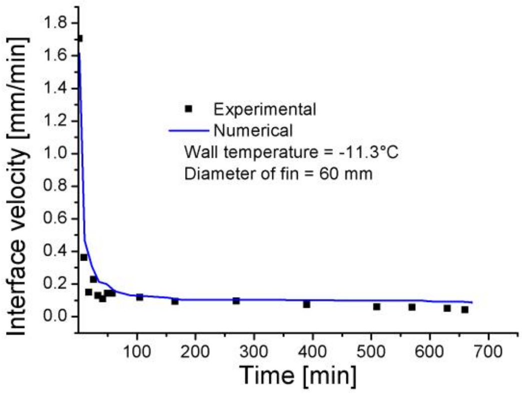

Figure 15. Comparison of the predicted interface velocity with experiments for the case of finned tube

Figure 16 shows the numerical prediction for four diferentes diameters of fins at the wall temperature of -10°C. As can be seen how much larger the diameter of fin as longer the solidification time. The convectives effects were neglected in this simulation, considering only the conduction effects of the finned tube material, copper in this case.

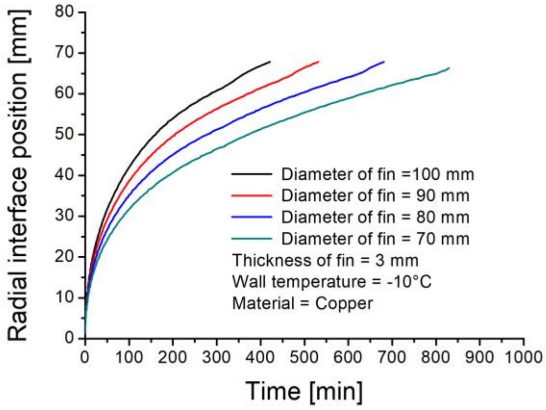

Figure 16. Numerical prediction of radial interface position for differents diameters of fins at wall temperature of -10°C

5. CONCLUSIONS

This paper presents the results of an experimental and numerical study of the effects of fins on the problem of solidification of PCM around a tube submersed in a tank full with PCM in the liquid phase. A home-built program based on the proposed conduction model was validated against experimental results indicating relatively good agreement. Comparisons of the numerical predictions with the experimental results showed a good concordance for lower number of mesh points, but the computational cost increases as the mesh points increase. The choice of the temporal step also influences the accuracy of the results, the smaller the step, the more precise the results, but the time to convergence increases as the time step is decreased. Several tests in the mesh configuration were made until reaching an ideal configuration, which was used in this work. It is found that fins increase the interface position, increase the solidified PCM mass, increase the associated accumulated energy, enhance the interface velocity and reduce the time for complete phase change of the PCM.

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