A NUMERICAL-EXPERIMENTAL STUDY CONCERNING ICE STORAGE TANKS

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Abstract: *A numerical-experimental investigation of a typical operation condition, associated with indirect, area-constrained, ice-on-pipe storage tanks is presented. The storage tank is simulated through a vertical annulus with the inner vertical wall representing one of the tubes packed into a typical storage tank. The outer vertical wall represents the maximum possible border for some ice layer growth, before it intersects another neighboring ice layer. In order to learn about the effect of loss in surface area and heat transfer rate, when two adjacent ice layers intersect during the ice making process, an experiment with a vertical annulus was carried out. Our task was to measure the temperature profile along the outer vertical wall over the time, with thermocouples positioned at specific locations and, to compare it with the correspondent numerical results. Regarding the annulus, the top and bottom walls of the cavity, as well as the outer vertical one were thermally insulated. The experiment begins with the water in the liquid phase, which is confined into the cavity, at the environment temperature. For the time greater than 0, it was imposed to the inner wall a prescribed typical cold temperature, smaller than the melting temperature of water (0°C).*
The mathematical model used to simulate transient natural convection of water with phase-change (solidification) was based on the finite volume method in order to solve the set of coupled conservation equations of mass, momentum and energy. It was adopted a fixed and regular grid of 140x140 nodal points and it was considered the fully-implicit time marching technique as well.

Key words: *Ice Storage Tanks, Annular cavity, Natural convection.*

1. Introduction

The present work aims to learn about the behavior of the formed solid around one vertical tube pertaining to a typical fusion latent heat storage system, such as indirect, area-constrained, ice-on-pipe storage tanks, during a process of full charging (ice making) Inside these equipments the tubes are usually positioned very close to one another and, whenever two or more ice layers intercept themselves, the thermal exchange is affected due to a reduction in the superficial heat transfer area. Other problem observed during a water solidification process is the appearance of a complex flow structure and uncommon patterns of temperature distribution in the melt, caused by the inversion density phenomenon for water around 4°C (Lin and Nansteel, 1987). This fact also affects significantly the thermal exchange between one tube of the device and its involving environment. In this sense, the shape and the growing velocity of an ice layer around and along a tube are functions of the velocity and temperature fields in the liquid region, which depend on the initial and boundary conditions to be considered.

Natural convection problems considering cold water in vertical rectangular and horizontal cylindrical annular cavities, have been dominated the research studies (Lin and Nansteel, 1987; Inaba and Fukuda, 1988; Tong and Koster, 1994; Vasseur and Robbilard, 1983 and Ho and Lin, 1990). Ho and Tu (1998) carried out a numerical study on vertical annulus, regarding natural convection of cold water from a stable laminar to oscillatory regime. The result showed that under specific conditions of the density inversion parameter, the buoyancy-driven flow experienced a Hopf bifurcation into a periodic oscillation regime at critical Rayleigh numbers. Phase-change processes around a cylinder or inside cylindrical cavities are of particular interest in thermal storage devices. Those one represent an important area and studies of performance concerning energy storage systems have been the target of many works (Laouadi and Lacroix, 199; Ismail and Abugderah, 2000 and Ismail et al., 2000). The main goal of the researches related to these systems has been the determination of the higher capacity of energy accumulation per unit volume.

West and Braun (1999) presented a numerical and experimental study of the partial charging and discharging processes, with icing, in tanks of ice storage. Stampa et al. (2001) have also presented a numerical study of natural convection in vertical annular cavity, considering the density inversion phenomenon of the water, however, without formation of the solid phase. In that study, the influence of the multi-cellular regime in the rate of heat transfer was investigated. In other study, Stampa et al. (2002) analyzed the growth of an ice layer around a vertical tube. In a numerical-experimental work presented by Benta et al. (2000), it was studied the solidification of water outside tubes, where the moving of the solid-liquid interface and the influence of the conditions for the secondary fluid in the process were experimentally investigated by using photographs. Abugderah and Ismail (2000) also investigated the solidification outside tubes vertically disposed within a shell and tube heat storage system. The results were concerned with temperature distribution for the whole domain, as well as phase change behavior in both axial and radial directions as a function of the Reynolds and Stefan numbers.

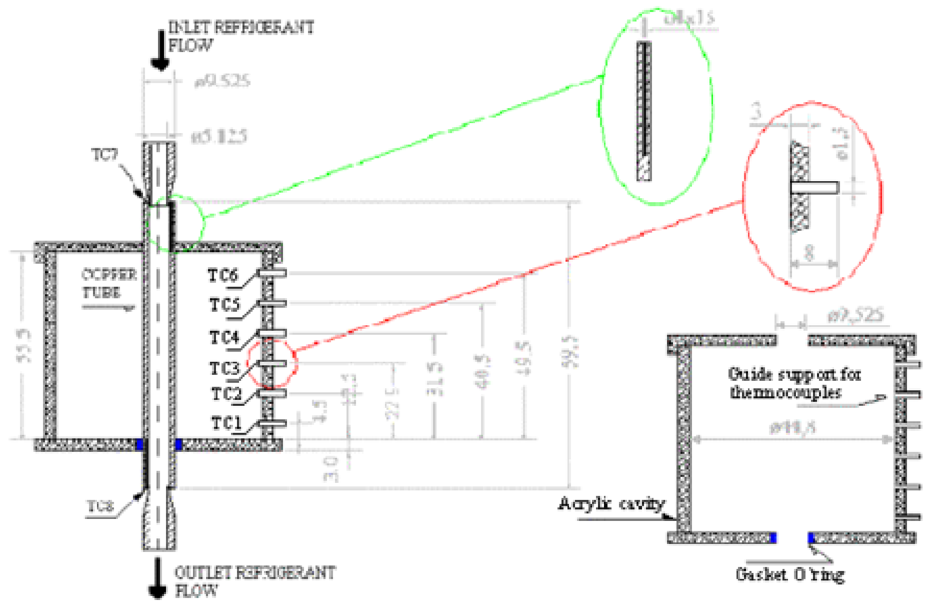


Figure 1. Test section.

The present study is intended to contribute with basic information to optimize designs of fusion latent heat storage systems. Our goal was to simulate what may occur within an indirect, area-constrained, ice-on-pipe vertical storage tank with respect to the growth of an ice layer around one of its vertical internal tubes. In order to provide further knowledge about the transient heat transfer during processes of freezing in such devices, in the present paper the problem has been analyzed both numerically and experimentally. Thus, one has considered an ice making process occurring within an insulated vertical annular cavity due to a cold inner surface of the annular space, where a fixed temperature was maintained all the time. The conditions established in the present study, correspond to a typical full charging process in such devices. A set of thermocouples was mounted at different heights over the outer vertical wall in order to yield the experimental mapping of the temperature distribution in that boundary, which represents in the present study, the maximum extend the ice layer can achieve. The numerical analysis provides results obtained by means of a control volume computer code developed specially for this work. In order to verify the simplifying hypotheses of the theoretical model, a comparison between experimental data and numerical results is made.

2. Description of experiment

A sketch of the test section is shown in Fig. 1. The test cell consists of a vertical annular cavity (outer diameter 44.8 mm, inner diameter 9.525 mm and height 55.5 mm), filled with water as being the phase-change material (PCM). The outer vertical wall of the cavity, as well as the top and bottom ones, are all made of 3-mm thick acrylic. On the other hand, the inner vertical wall consists of a thin wall copper tube, in other to enhance the internal thermal exchange. Six guide supports with inner diameter of 1,5 mm have been mounted at the external vertical wall. They are used to accommodate thermocouples to acquire the local water temperatures. These supports are vertically aligned and spaced at different heights in accordance to Fig. 1. Two orifices with diameter of 1 mm and 15 mm deep have been made at specific locations on the copper tube. They are positioned closely to the inlet and outlet of the copper tube. Thermocouples have been attached within these orifices in other to measure the wall temperatures at these two locals.

The present study considers the whole cavity as being externally insulated. To minimize the heat transfer to the environment, the external walls have been covered with a 15 mm thick layer of foam rubber surrounded by a 20 mm thick layer of expanded polystyrene. As a uniform temperature boundary condition was chosen for the inner vertical wall, the copper tube was chilled internally by means of a refrigeration system. To guarantee a uniform distribution of temperature over the whole external surface of the copper tube, a continuously running thermal bath pump supplied the refrigerant fluid to the test cell with a high flow rate value. The arithmetic mean temperature calculated by the measured values with the thermocouples positioned at the copper tube is considered as being the constant wall temperature adopted in the numerical simulation, since these values should not differ more than 1 ºC during the run.

As can be seen in Fig. 1, eight thermocouples were mounted inside the orifices mentioned before. Those thermocouples, which were positioned over the outer vertical wall, are denoted as TC1, TC2, TC3, TC4, TC5 and TC6. They measure the water temperatures over the time, which is the object of the present study, exactly at the outer wall. Their positions with respect to an origin placed at the bottom of the cavity are informed in Tab. 1. The thermocouples mounted at the copper tube are denoted as TC7 (inlet) and TC8 (outlet). All temperatures have been measured by using 0.508 mm diameter T-type copper–constantan thermocouples of exposed junction. A digital data acquisition system, Hewlett–Packard HP3458A, has been used to measure the electromotive force (e.m.f) for each thermocouple. Such system was attached to a personal computer, where the associated HP software converted the electrical signs into temperatures by using ASTM temperature–e.m.f. tables for standardized thermocouples (ASTM, 1987). All of the thermocouples have been calibrated, simultaneously, together with the data acquisition system and the personal computer, for which a typical operating condition was simulated. Considering all the thermocouples, the uncertainties have been calculated with a reliability level of 95 % and the values obtained for each thermocouple are shown in Tab. 2.

Table 1. Positions of the thermocouples

TCno.	**1**	**2**	**3**	**4**	**5**	**6**
Z (mm)	4.5	13.5	22.5	31.5	40.5	49.5

Table 2. Uncertainties of the thermocouples

TCno.	**1**	**2**	**3**	**4**	**5**	**6**
Uncertanty (˚C)	± 0.33	± 0.21	± 0.40	± 0.22	± 0.31	± 0.17

The experiment consisted of a solidification process and it was performed in the following manner. The vertical annular cavity was filled with distilled water at an ambient air temperature. After that, the cavity was covered with its external thermic insulation. The readings of the thermocouples were verified before the experimental run, exposing them simultaneously to the internal cavity ambient, during nearly 15 min. This time of exposure was used to turn the internal temperature of the test cell homogeneous, as well as to eliminate any convection in the liquid water. In the refrigeration system, the refrigerant fluid moves in a closed circuit. There is a by-pass near the inlet of the cavity, which is utilized to circulate this fluid before the experimental run begins. In this sense, one could stabilize the operation of such system and adjust the refrigerant temperature in the desired value.

One has considered the beginning of the run, t= 0 s, when the maximum difference between the readings of any two thermocouples did not exceed 0.2 ºC and, the value adopted in the numerical simulation (T_{IN}) was the average one among all readings, being T_{IN} = 23.7 °C. For a time greater than zero, the inner wall of the cavity was kept at a temperature lower than the melting point of the water (T_m= 0°C) by means of the refrigeration system, in order to make possible the icing process. The prescribed cold temperature was settled at the value T_C= – 11.6 °C. After preliminary testing runs, one has chosen 1 hr (Fo=13.92) as the period of time to compare experimental data with numerical predictions.

3. Mathematical formulation and numerical solution

The physical system consists of a vertical annulus with height L and annular space W, being w the inner radius. It is shown in Fig. 2 for a certain instant of time, during which the phase change process is occurring. When making comparisons with experimental data, the initial and boundary conditions are those described previously. Thus, initially, all the system has its temperature settled in the experimental value of T_{IN}. The boundary conditions are considered as adiabatic at the top and bottom of the cavity, as well as at the external wall. For a time greater than zero, the experimental cold temperature, T_C , is imposed at the inner wall of the cavity.

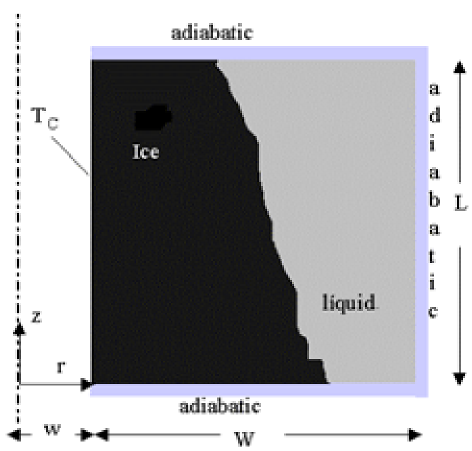


Figure 2. Physical system.

The mathematical model formulated to represent the physical system is based on the following simplifications:

(1) the fluid flow is considered laminar and the PCM is a Newtonian fluid;
(2) the physical properties are constant except the density of the liquid in the buoyancy term;

(3) the values adopted for the radius ratio are small enough, such that the buoyancy flow can be considered axis-symmetrical, i.e., two-dimensional;
(4) the difference of density between solid and liquid does not create appreciable local motion of the liquid.

The problem is then governed by the following conservation equations:

$$\mathbf{div}\left(\rho_\ell\ \mathbf{U}\right) = 0 \tag{1}$$

$$\frac{\partial(\rho_\ell \mathbf{U})}{\partial t} + \mathbf{div}(\rho_\ell \mathbf{U}\mathbf{U}) = \mathbf{div}\left[\mu_\ell\ \mathbf{grad}\ \mathbf{U}\right] - \mathbf{grad}\ p + \rho_\ell \mathbf{g} \tag{2}$$

$$\frac{\partial\left[\left(\rho_\ell\, cp_\ell\, \varepsilon + \rho_s\, cp_s\,(1-\varepsilon)\right)T\right]}{\partial t} + \mathbf{div}\left(\rho_\ell\, \varepsilon\, \mathbf{U}\, cp_\ell\, T\right) = \mathbf{div}\left[\left(k_\ell\, \varepsilon + k_s(1-\varepsilon)\right)\mathbf{grad}\, T\right] - \frac{\partial\left(\rho_s\, \varepsilon\, \Delta h_{lat}\right)}{\partial t} \tag{3}$$

where ρ is the specific mass and μ the absolute viscosity. The subscripts ℓ and s correspond to the liquid and solid properties of PCM. **U** is the velocity vector, p is the pressure, **g** is the gravitational acceleration vector. The conservation equations of mass and momentum are solved for the liquid phase only, whereas the conservation equation of energy is solved in the whole domain, for both solid and liquid phases. For this equation, the thermo-physical properties of the PCM expressed by k e cp, are thermal conductivity and constant pressure specific heat, respectively. T is the temperature of the material and t is the time. Δh_{lat} is the latent heat of fusion and, ε is the volume fraction, which is defined as

$$\varepsilon = \frac{\forall_\ell}{\forall} \tag{4}$$

where $\forall_\ell$ and $\forall$ are the liquid volume and the total volume, respectively.

Initially, the PCM is in the liquid phase, thus the volume fraction ε is equal one in the whole domain. ε is set to zero at the regions, where the temperature of the material reaches the value of the fusion temperature of water, T_m (0°C). The last term on the right-hand side of Eq. (3) is different from zero only in the regions where the change of phase is occurring.

In the past, many correlations have been proposed to represent the density of cold water as a function of temperature such as rational function (Kell, 1967; Seki et al., 1978) and polynomial approximation (Watson, 1972). Other correlations include Chen and Millero (1976), Poulikakos (1984), and Gebhart and Mollendorf (1977). Although most of these correlations are in close agreement, the relation from Gebhart and Mollendorf will be used here due to its high precision and simple form. Thus, according to assumption (2), in the third term on the right-hand side of Eq. (2), the following density-temperature relationship of water was adopted for the liquid density

$$\rho_\ell = \rho_{\ell,max}\left[1 - rsp\left|T - T_{max}\right|^b\right] \tag{5}$$

where $\rho_{\ell_{MAX}} =$ 999,972 kg/m^3, rsp = 9.297 x 10^{-6} (°C)$^{-b}$, $T_{max} =$ 4.029 °C and b = 1.895. This relationship takes into account the nature of the inversion-density in the water.

Equations (1)–(3) have been made dimensionless in order to individualize the significant parameters of the problem. So, the non-dimensional variables are defined as follows:

$$\mathbf{U}^* = \frac{\rho_{\ell,max}\,\mathbf{U}\,W}{\mu_\ell} \qquad P = \frac{p^*\ \rho_{\ell,max}}{(\mu_\ell / W)^2} \qquad Z = \frac{z}{W} \tag{6}$$

$$\theta = \frac{T - T_C}{T_{IN} - T_C} \qquad Fo = \frac{\mu_\ell\ t}{\rho_{\ell,max}\ W^2} \qquad R = \frac{r}{W} \tag{7}$$

where p* is the modified pressure, defined as $p^* = p + \rho_{\ell,max}\ g\ z$. The conservation equations may be written in dimensionless form as follows:

$$\mathbf{div}\left(U^* \right) = 0 \tag{8}$$

$$\frac{\partial\ \left(U^*\right)}{\partial\ Fo} + \mathbf{div}\ \left(U^*\ U^*\right) = \mathbf{div}\ \left[\ \mathbf{grad}\ \ U^*\right]\ -\ \mathbf{grad}\ P\ +\ Gr\ \left|\theta\ -\ \theta_{max}\right|^b\ k \tag{9}$$

$$\begin{aligned}&\frac{\partial\left(\left[\varepsilon+\rho^* cp^*(1-\varepsilon)\right]\theta\right)}{\partial\, Fo} + \mathbf{div}\left[\varepsilon\, \mathbf{U}^*\ \theta\right] = \\ &\quad \frac{1}{Pr}\,\mathbf{div}\left\{\left[\varepsilon + k^*(1-\varepsilon)\right]\mathbf{grad}\,\theta\right\} - \\ &\qquad - \left\{\left[\frac{\rho^*}{Ste_\ell + \left(Ste_s \big/ cp^*\right)}\right] - \left(1+\rho^* cp^*\right)T_C^*\right\}\frac{\partial\,\varepsilon}{\partial\, Fo}\end{aligned} \tag{10}$$

where the dimensionless parameters are:

$$\rho^* = \frac{\rho_s}{\rho_{\ell,max}} \qquad k^* = \frac{k_s}{k_\ell} \qquad cp^* = \frac{cp_s}{cp_\ell} \qquad RR = \frac{W+w}{w} \qquad AR = \frac{L}{W} \qquad Pr = \frac{\mu_\ell\, cp_\ell}{k_\ell} \tag{11}$$

$$Gr = \frac{g\ rsp\ (T_{in} - T_c)^b\ W^3}{(\mu_\ell / \rho_{\ell,max})^2} \qquad Ste_\ell = \frac{cp_\ell\left(T_{in} - T_m\right)}{\Delta h_{lat}} \qquad Ste_s = \frac{cp_s\left(T_m - T_c\right)}{\Delta h_{lat}} \qquad T_C^* = \frac{T_C}{T_{in} - T_C} \tag{12}$$

It can be noticed that the problem is governed by the Prandtl number, Pr, Grashof number, Gr, Stefan number, Ste, ratio of specific mass, ρ^***,** ratio of thermal conductivity, k^*, ratio of constant pressure specific heat, cp^*, power b, cooling factor, T_C^*, and by the geometric parameters RR (radius ratio) and AR (aspect ratio) and the initial condition is $\theta_{in} = 1$**.**

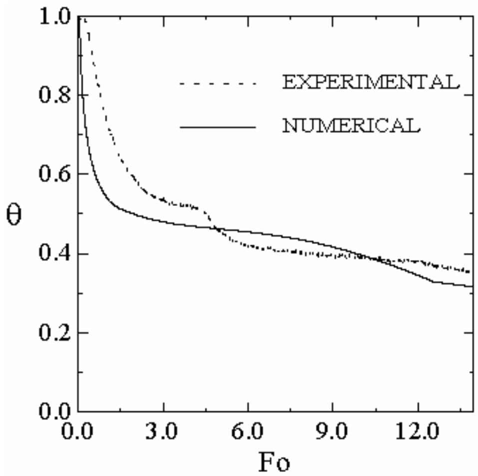
The present two-dimensional and transient problem has been treated numerically using the control volume method (Patankar, 1980). The convective and diffusive fluxes that cross all faces of each control volume were handled by the Power Law scheme. It has been considered the fully-implicit time marching technique and the solution method used for the algebraic linear equations was the TDMA line-by-line solver with a block correction method (Patankar, 1980). The SIMPLE algorithm has been used for pressure–velocity coupling (Patankar, 1980).

A fixed and regular grid has been adopted on the physical domain. The mesh and time step have been defined after a grid test in which their correspondent chosen values did not present significative variation on the numerical solution. A mesh size of 140 × 140 nodal points was defined, with a time step of 0,65 s. Solutions were considered converged at a particular time step if they had residuals of mass and energy less than 10^{-6} for at least eight consecutive iterations

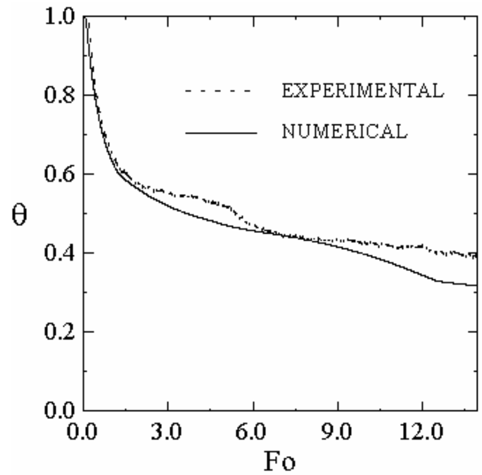
4. Results and discussion

The physical validity of the mathematical model described above may be studied by comparison of predictions with experimental data. The results of the comparison are reported in Fig. 3, in which temperature profiles evaluated at the locals defined in Fig. 1 are shown. In order to help the analysis of Fig. 3, three intermediate numerical results are presented in Fig. 4 through streamlines, isotherms and amount of formed ice, once the cavity has been kept quite insulated during the experimental run.

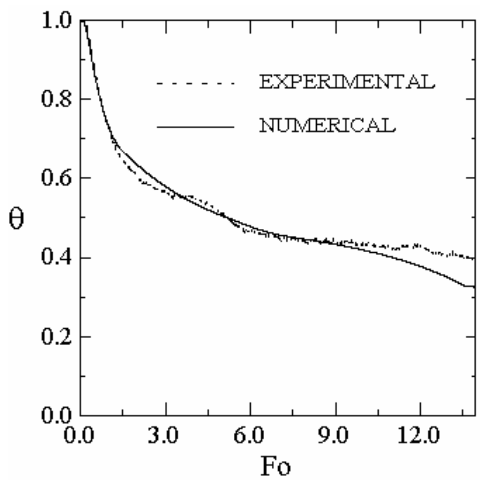
Regarding Fig. 4, for Fo=3 (a) one can see two cells of flow in the melt. The largest cell is a strong thermally driven clockwise circulation flow, in which up flow of bulk liquid occurs along the insulated wall (left side of the figure) towards the top of the cavity. As the bulk temperature at the bottom right corner of the annulus stands about T_{INV} (≈ –4°C), a weak secondary counterclockwise cell, confined by the maximum density contour, can be seen to rise and rotates due to inertia and drag effects. The thermocouple TC1 is located at the deeper portion of the cavity and, from the beginning of the run up to this value of the Fourier number, it has been measuring temperature values within the colder region of the largest cell. From Fig. 3 one verifies that its experimental profile stands above the numerical profile. Yet, the experimental and numerical profiles corresponding to thermocouples TC2 and TC3 present an excellent agreement in this range of time. The thermocouples TC4, TC5 and TC6 are all positioned at the superior part of the cavity. But in this case, all their experimental profiles stood below their correspondent numerical profiles. More precisely, only the upper thermocouples TC5 and TC6 had a sensible discrepancy between experimental data and numerical predictions,



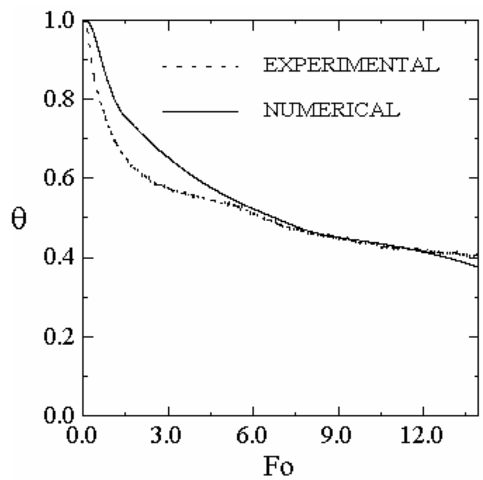
Thermocouple TC1



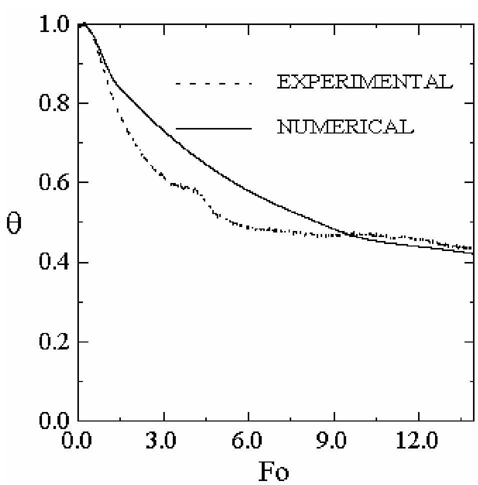
Thermocouple TC2



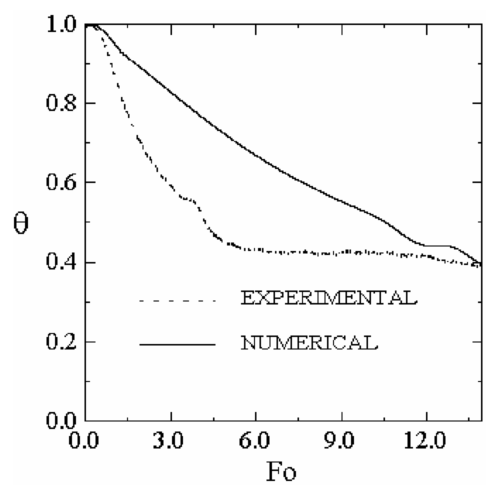
Thermocouple TC3



Thermocouple TC4



Thermocouple TC5



Thermocouple TC6

Figure 3. Comparison between experimental and numerical temperature profiles.

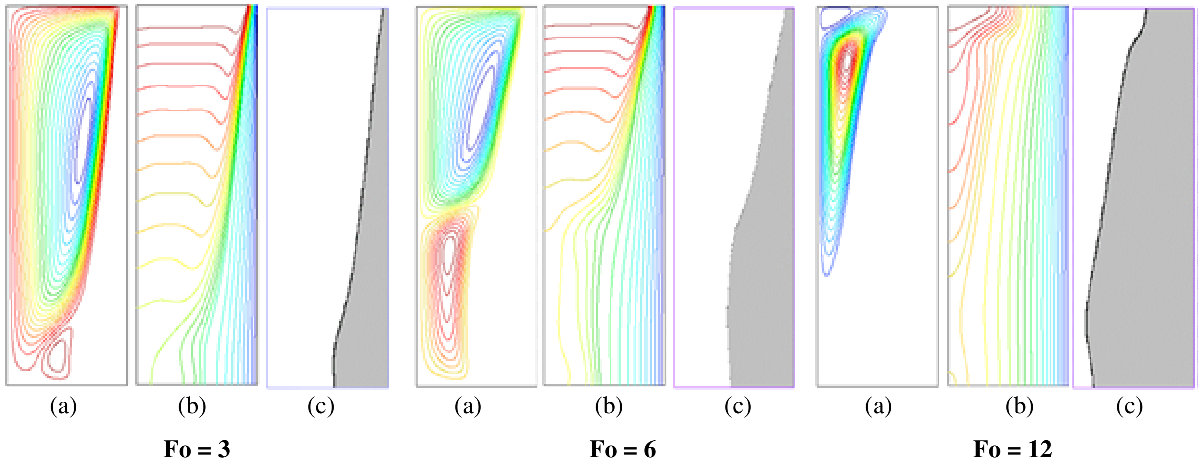


Figure 4. Intermediate numerical results: (a) streamlines; (b) isotherms and (c) formed ice.

mainly for the latter case. Even so, this difference becomes less evident for increasing Fourier numbers. One supposes this fact may be attributed to the way that the boundary condition of uniform temperature at the inner vertical wall of the cavity was experimentally imposed. As mentioned before, the inner vertical wall is chilled by the refrigerant fluid,, which flows downward through the copper tube. Thus, the whole copper tube has its initial temperature suddenly reduced up to the time that a stable and fixed cold wall temperature is achieved. This process lasts a short period of time and during it, the upper part of this tube is chilled first than the inferior one. So, as thermocouple TC6 is positioned just 6 mm below the inlet of the cavity, this effect is much more pronounced.

Analyzing Fig. 4, from Fo= 3 to Fo= 6 one can notice that the global temperature decay inside the cavity turns the secondary cell bigger than before, moving the maximum density contour towards the top of the cavity due to the reduction of the temperature gradients. So, around Fo=6 the thermocouple TC1 measures temperature values within the colder region of the secondary cell, which rotates in opposite sense with respect to the primary cell. In this case, one can observe in Fig. 3 an inversion of positions between the experimental and numerical profiles. The crossing of the two curves must be related to the change of cell during which either the acquisition of data by this thermocouple or the numerical predictions have accompanied this change of trend. Continuing the analysis of Fig. 3 for the range mentioned above, it is verified a very good agreement between experimental and numerical profiles for the thermocouples TC2, TC3 and TC4, with great significance for the fitting performed by the mathematical model in the case of thermocouple TC3. Regarding thermocouples TC5 and TC6, the distance between the experimental and numerical profiles still can be explained by the suppositions assumed before.

The final range of time to be analyzed is concerned to Fo= 6 to Fo= 12. According to Fig. 4, during this period of time the secondary counterclockwise cell becomes the dominant thermal mechanism because the clockwise cell offers less resistance to its growth. Basically, all the experimental data are acquired within the secondary cell, which rotates more and more slowly. According to Fig. 3 for this range of time, almost all the numerical predictions presented excellent results, mainly in the case of thermocouple TC4. Only for thermocouple TC6, the numerical predictions presented reasonable discrepancy. However, as already mentioned before, some experimental operational fail may have caused this effect.

5. Conclusions

The present study had as its main goal to contribute with basic information to optimize designs of fusion latent heat storage systems. To do so, a numerical-experimental investigation of a typical operating condition, associated with indirect, area-constrained, ice-on-pipe storage tanks is presented, during an ice making process in such devices. A vertical annulus with the inner vertical wall representing one of the tubes packed into a typical storage tank and, the outer vertical wall representing the maximum possible border for some ice layer growth, before it intersects another neighboring ice layer, was chosen as the physical system. The study consisted in measuring temperature profiles along the outer vertical wall over the time, with thermocouples positioned at specific locations and, to compare it with the correspondent numerical results.

The mathematical model used to simulate transient natural convection of water with phase-change (solidification) was based on the finite volume method in order to solve the set of coupled conservation equations of mass, momentum and energy. In order to better understand the comparisons between experimental and numerical profiles, one has launched of graphics in which streamlines, isotherms and ice formed were plotted for specific values of the Fourier number. The capacity of the mathematical model in predicting the transient temperature behavior at the chosen locals inside the cavity was evaluated considering three ranges for the variation of the Fourier number. Firstly, from an overall qualitative point of view, the mathematical model gives trends of temperature very similar to those experimental. For all the ranges the mathematical model was capable to follow physical variations in the melt and, excellent fittings were obtained for some experimental profiles. Particularly, for the two thermocouples positioned closely to the top of the cavity, their numerical profiles presented a reasonable discrepancy in relation to their correspondent experimental ones. However, we believe that behind of this fact is the experimental way the cooper tube (inner vertical wall) was chilled to produce a boundary condition of uniform temperature. Nearly to the end of the last range, one could note the approximation between experimental and numerical profiles for these two thermocouples. So, in an overall sense, the mathematical model was able to predict a multicellular flow structure, typical of natural convection under influence of the inversion density phenomenon of water, near 4 ºC, with a very good performance.

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