PCM SOLIDIFICATION AROUND THE PARALLEL PLATES OF A STORAGE SYSTEM

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Abstract. *Ice banks are relevant in balancing the demand for electricity because they accumulate energy in the form of cold during off-peak periods to be used later during periods of high consumption, especially during peak hours. This energy accumulation in the phase change material, in these cases water, is a well-accepted technology and is expanding in several fields of engineering and building applications. In this study, a thermal model based on pure conduction is developed to describe the solidification process in a parallel plate storage system. For the solution of this model, the governing equations for the solid and liquid phases and the interface moving are discretized using the finite difference approximation and totally implicit approach with variable time step. The computational code representing the thermal model for the phase change process around the parallel plates of a storage system was tested and validated against available numerical and experimental results, showing good agreement. The energy stored, interface position, interface velocity and the time for complete solidification are presented and discussed in terms of the temperature of the cold plate and the distance between the plates. As the temperature on the surface of the plate decreases, there is a reduction in the time to complete phase change and a higher solidification rate. However, increasing the distance between the plates contributes to a longer time for complete solidification. The contribution of this study through the adopted model and solution method aims at a provisional way of obtaining an evaluation of this type of storage configuration and expanding its use in commercial applications and scientific research, in addition to predicting its thermal performance parameters quickly and accurately. Therefore, the model developed to analyze the thermal performance of the storage tank contributes some new and relevant data on this type of system and be useful for designers in that specific area.*

Keywords: *solidification, thermal storage, parallel plates, phase change material*

1. INTRODUCTION

The knowledge of phenomena related to phase change, such as the solidification process, represents a factor for the design and development of various equipment in a wide field of applications, both in the industrial sector and in the academic and scientific sectors. Mathematical modeling and solution techniques for such phenomena have been studied over the years for different geometries and different Phase Change Materials (PCM), resulting in exact solutions for the easier problems, semi-analytical and numerical solutions for the most complex cases. Thus, the mathematical modeling of most of the problems presents a complex solution since the interface that separates the solid and liquid phases moves as the respective latent heat is absorbed or released. One of the main applications of these processes is the so-called "ice banks", which consist of equipment that accumulates energy in the form of latent heat at low temperatures. The main objective of the construction of the ice banks is to alleviate the peaks in electricity consumption, that is, periods in which this demand is very high. Another relevant aspect is that they allow the air conditioning system to be minor as these consumption peaks can be supplied by the stored cold.

The ice bank is a type of Latent Heat Thermal Energy Storage (LHTES) that uses water as a phase change material, which can operate in charge and discharge cycles. They are one of the promissory cold storage systems because water is an inexpensive PCM with a high calorific capacity. According to Asgharian and Baniasadi (2019), they are divided into two categories, including static and dynamic systems. In the static model, ice is made and used in the same place during the solidification and melting processes. Already in dynamic systems, ice is charging in a specific space and then transferred to another location for use in the discharge. Barz *et al.* (2018) report that these latent heat thermal energy storage systems that combine a high energy density, with the advantage of the isothermal nature of the accumulation process, integrate several industrial applications, such as concentrated solar thermal plants and air conditioning units,

cold and refrigeration production. For Ismail and Gonçalves (1999), the energy storage systems in the form of latent heat present a valuable option for energy accumulation concerning those that use it in the form of sensible heat. They have a low volume/energy ratio experienced during charge and discharge processes, which offers the most accumulated heat per unit volume. According to Mosaffa et al. (2014), these systems are designed to accumulate the lowest temperature compared to the environment temperature because the energy can be charged, stored, and discharged daily, weekly, monthly, or in seasonal cycles.

Phase change materials are widely used in thermal energy storage systems due to their high capacity for heat accumulation in latent form. According to Yilbas et al. (2015), the heat involved during the solidification process is high, and its thermal conductivity is low, which makes charging and discharging times longer. Al-Maghalseh and Mahkamov (2018) presented a comprehensive review of significant studies on thermal energy storage technologies using phase change materials. The review focused on some techniques applied to improve the performance of these systems and the methods used to analyze the problems of heat transfer. Thus, the purpose of the review was to provide a solid basis for identifying the ideal project for the various applications using these PCMs. For Berdja et al. (2019), many studies have been conducted on PCMs for thermal storage in the form of cold, focusing mainly on the application in building structures, solar energy systems, air conditioning, refrigeration equipment, and others.

Zalba *et al.* (2004) designed, built, tested, and performed an experimental installation to study PCMs with a melting temperature around 20°C to 25°C. This study analyzed the applicability of the free cooling installation to store cold outdoors at night and release it indoors during the day. In his studies, Panesi (2016) presented an experimental model tested for use in the analysis of the thermal performance of the PCM in compacted beds, compared the results numerically using the explicit finite difference method. Ismail and Lino (2011) performed an experimental investigation of the effects of radial fins and turbulence promoters in improving heat transfer by phase change external to a horizontal tube submerged in the PCM with a working fluid flowing through it. The experiments were realized on tubes without fins, finned tubes, and finned tubes with the turbulence promoter to obtain the results of interface position, interface velocity, fraction of the solidified mass, and the time for complete solidification as a function of the temperature of the working fluid, fluid mass and tube arrangement.

In their study of the energy rating of a thermal storage unit with several PCMs, Mosaffa et al. (2014) analyzed that heat released from the material during the night and after melting used to cool the interior of buildings during the hot hours of the day. The discharge process is carried out at night when the ambient temperature is low compared to the temperature of the buildings indoors. Heat is absorbed by the PCM when that ambient temperature exceeds the thermal comfort limit. Then, the hot air to be cooled passes through the storage unit, and the phase change material absorbs heat from the air during the melting process.

According to Deng et al. (2019), finned geometries in the presence of the phase change materials are one of the most efficient methods to improve the energy transfer between the PCM and the heat transfer fluid. Thus, an appropriate arrangement of these extended surfaces performs a significant role in the project of a thermal energy storage unit in the form of latent heat. Candanedo *et al.* (2013) proposed a model based on a predictive control algorithm to find an appropriate combination of chillers and ice banks to provide the necessary cooling in a building at minimum cost within an electricity profile. The system produced ice during the solidification or charging process when the cost of electricity was lower. Ismail and Jesus (2001) realized a numerical analysis of the solidification of the PCM on a cylinder in their study on the ice bank. A pure conduction model was applied for the heat transfer inside the PCM and on the walls and modeled the convection process inside the tube based on the specific Nusselt number. Consequently, the temperature distribution during the process, the interface velocity at the phase change, and the energy stored in the system were determined.

The numerical and analytical analyzes of the processes involving solidification are complex. The analytical solutions are limited due to the high mathematical level observed in the phenomenon, while most numerical solutions require high computational costs. In such cases, the general solutions involve three-dimensional transient analyzes of the temperature distribution in a body whose physical properties depend on the temperature.

Teggar and Mezaache (2013) presented a conduction model that described the internal solidification of a phase change material, in this case, water, inside a flat plate type cold storage unit. In this study, an enthalpy method with a finite control volume approach was used and validated the model by comparison with experimental results available in the literature. Ismail *et al.* (1999) presented the results of a numerical and experimental study of parallel flat plate ice banks to identify the relative importance of geometric and operational parameters, as well as the thermal performance of the system. The model was based on the one-dimensional phase change problem and performed its numerical solution using the finite difference method in a fixed grid scheme. The model was based on the one-dimensional phase change problem and its numerical solution was performed using the finite difference method in a fixed grid scheme. The stored energy, the interface position, and the time for complete solidification were discussed through the independent parameters such as the initial temperature of the PCM, the temperature on the cold plate surface, and the distance between them.

Bechiri and Mansouri (2015) developed an analytical solution to study the volumetric effects of heat generation during the solidification process of a phase change material encapsulated in a horizontal cylindrical thermal energy storage container. For the solution of the energy equations representing the solid and liquid phases and in the transient

regime, the variable separation method and the exponential integral function were used. The results were satisfactory to the experimental data previously reported. Ismail *et al.* (2015) showed the results of a numerical study on the solidification of the PCM around a curved tube with a flow of cooling fluid inside it. The numerical results of the work were validated based on existing experimental results, where they were satisfactory in predicting the formation of ice on helical coils and curved tubes used in latent heat storage systems.

From the above literature review, it is possible to observe some works referring to the phase change processes in some specific geometries. Due to the reduced studies involving systems with parallel flat plate configuration, this study aims to evaluate these units by the analysis of a thermal model and simulation of a latent thermal energy storage unit by obtaining relevant parameters such as the interface position, interface velocity, stored energy, and time for the phase change complete during the solidification.

The contribution of this study through the model and solution method adopted aims at a provisional way of obtaining an evaluation of this type of storage configuration and expanding its use in commercial applications and scientific research. Parallel plate type latent heat storage devices are not so usual, although simple to manufacture, easy to be modulated in the formation of the larger capacity units and be incorporated into existing installations. These desirable characteristics have become the motivating factors for the study, together with a few pieces of information available in the literature about these flat plate type storage units.

2. MODEL DESCRIPTION

2.1 Physical model

The parallel plate type heat storage unit consists of a set of interconnected cooling plates through which the cold fluid circulates, as shown in Figure 1. The storage tank consists of a set of parallel plates through which a heat transfer fluid flows with a temperature below 0°C. These cold plates are heat exchangers made with aluminum or copper material so maintain a constant and uniform temperature on their surface. The liquid PCM starts to solidify, moving in the normal direction of the plate surface to the right and the left, as shown in the dashed lines during the loading process. At the end of the process, the solidification front does not advance and, thus, the storage tank is fully charged. In this situation, the two fronts between two successive plates will meet in the middle of the distance between them, called the symmetry line or adiabatic line, since the temperature gradient in that location is zero.
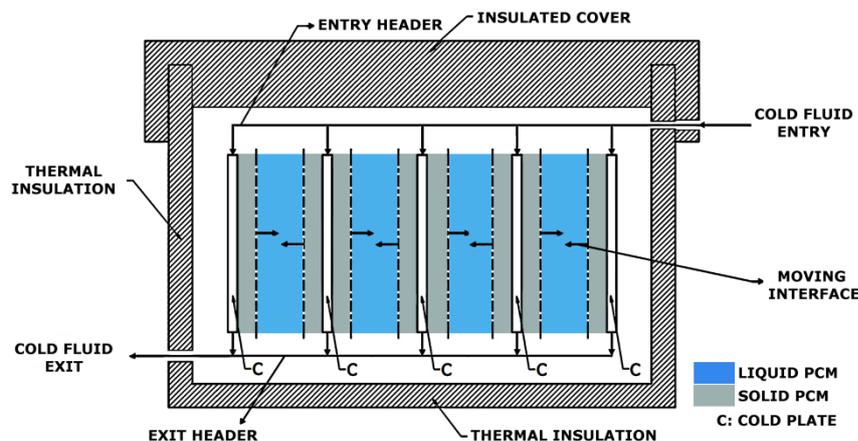

Figure 1. Representation of the flat plate type latent heat storage unit during solidification.

According to Ismail *et al.* (2021), at the initial time $t = 0$, the liquid water confined to the space between the plates is at a temperature higher than the phase change temperature, T_m. The charging process begins when the surface of the flat plate is cooled to a constant temperature below 0°C by the cold fluid that circulates through it and, the solid-liquid interface begins to move to the symmetry region. It is assumed that there is no fluid movement and the transfer of heat from the liquid to the solid occurs exclusively by conduction, disregarding the convective effects.

2.2 Governing equations

Before describing the mathematical model through its governing equations, the simplifying hypotheses assumed that were considered relevant for the development of this model are described below. The proposed model is of the one-dimensional transient type in the direction of the interface movement and perpendicular to the plate surface. Halfway through the distance between two successive plates, the phase change process or symmetry condition ends, and the temperature at the surface of the plate is kept constant.

The thermophysical properties of PCM and their correlations were calculated from Cho and Sunderland (1974) and Fukusako and Yamada (1993). The mathematical formulation of the phase change problem, where the temperatures $T_s(x,t)$ and $T_l(x,t)$ for the solid and liquid phases, respectively, is described by Eq. (1) and Eq. (2) as follows:

Solid phase

$$\frac{\partial^2 T_s(x,t)}{\partial x^2} = \frac{1}{\alpha_s}\frac{\partial T_s(x,t)}{\partial t}; \quad 0 < x < s(t) \tag{1}$$

Liquid phase

$$\frac{\partial^2 T_l(x,t)}{\partial x^2} = \frac{1}{\alpha_l}\frac{\partial T_l(x,t)}{\partial t}; \quad s(t) < x < B \tag{2}$$

The additional equation, Eq. (3), is determined by considering an energy balance at the interface at $x = s(t)$, declared as:

Interface

$$k_l \frac{\partial T_l}{\partial x} + \rho Q_L \frac{ds(t)}{dt} = k_s \frac{\partial T_s}{\partial x}; \quad x = s(t) \tag{3}$$

where Q_L, α, ρ, k, B, and $s(t)$, are the latent heat of solidification per unit mass associated with the phase change, thermal diffusivity, density, thermal conductivity, the half the distance between the plates, and interface position, respectively. Figure 2 shows the coordinates of the one-dimensional solidification process.
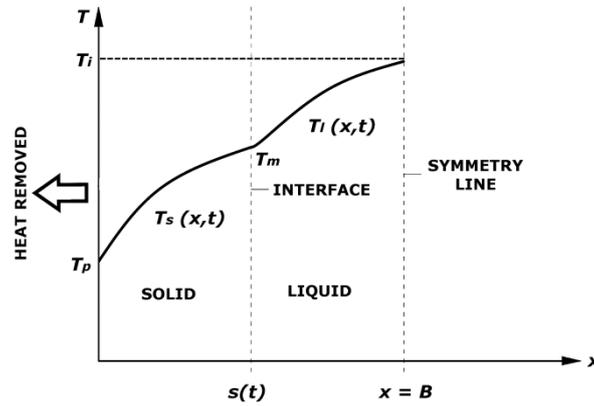

Figure 2. Representation for one-dimensional solidification.

The stored energy is composed of sensible heat, evidenced by the decrease in the temperature of the PCM and latent heat when this same material changes phase. According to Ismail *et al.* (1999), the sensible heats of the solid and liquid phases be calculated by integrating the temperature distribution curve of these phases in the PCM. Sensitive heats in solid and liquid phases are given, respectively, according to Eq. (4) and Eq. (5).

$$Q_{sens(s)} = A_{sup}\,\rho_s\,c_{p(s)} \int_0^{S(t)} T(x)dx \tag{4}$$

$$Q_{sens(l)} = A_{sup}\,\rho_l\,c_{p(l)} \int_{S(t)}^{B} T(x)dx \tag{5}$$

The latent heat is expressed in terms of the position interface $s(t)$, the latent heat of solidification per unit of mass (L), the density (ρ) and the surface area of the plate in contact with the PCM (A_{sup}), according to Eq. (6).

$$Q_L = A_{sup}\,\rho_s\,s(t)\,L \tag{6}$$

Finally, the total heat stored in the unit, Q_{total}, is the sum of these energies involved in the solidification process according to Eq. (7).

$$Q_{total} = Q_{sens(s)} - Q_{sens(l)} + Q_L \tag{7}$$

2.3 Numerical procedure

The governing equations were discretized by the finite difference method using the Modified Variable Time Step (MVTS) approach proposed by Gupta and Kumar (1981) and described in Özişik *et al*. (2017). It is important to note that the discretization method used in time works with properties in a totally implicit way, that is, in the advanced time step. By the numerical approach, MVTS, to solve the problem of solidification process in analysis with finite differences, the "*x-t*" domain is subdivided into small Δx intervals constant in the spatial domain and variable intervals in time Δt. The approach of the variable time step requires that, at each time level t_n, the time step Δt_n is chosen so that the interface moves just at a distance Δx in the same interval. Thus, the concern is to determine this time step. The energy balance at the interface represented by Eq. (3), when being discretized through the finite difference method, results in Eq. (8) that describes the time step with the spatial domain as follows:

$$k_s \frac{T_m - T_{s,i}^{n+1}}{\Delta x} - k_l \frac{T_{l,i+2}^{n+1} - T_m}{\Delta x} = \rho_s Q_L \frac{\Delta x}{\Delta t_n} \tag{8}$$

An important numerical parameter defined for the computational cost is the convergence criterion specified as the difference between two consecutive time steps, in successive iterations, presented in the numerical method of the modified variable time step (MVTS). In the simulation, the specified convergence criterion is 10^{-5}. The choice of this convergence criterion was determined by the error value between two successive iterations in each time step, despite the totally implicit approach of the method of finite differences in temporal properties being unconditionally stable. Once the convergence criterion is defined, it determines the best computational mesh to be adopted in the problem. To choose the most appropriate mesh, some simulations were performed, using different combinations of Δx, represented by the step in the space domain along the positive direction of x. The purpose of the tests is to evaluate the refinements of the meshes to reach a condition in which the quality of the results is not compromised and represents a shorter simulation time, and consequently, a lower computational cost. The computational mesh with 100 points along the spatial domain showed better results, whereas changing parameters such as T_P and B did not result in additional computational costs. The tested and optimized numerical program was developed in the MATLAB (*Matrix Laboratory*) software according to the block diagram shown in Figure 3.
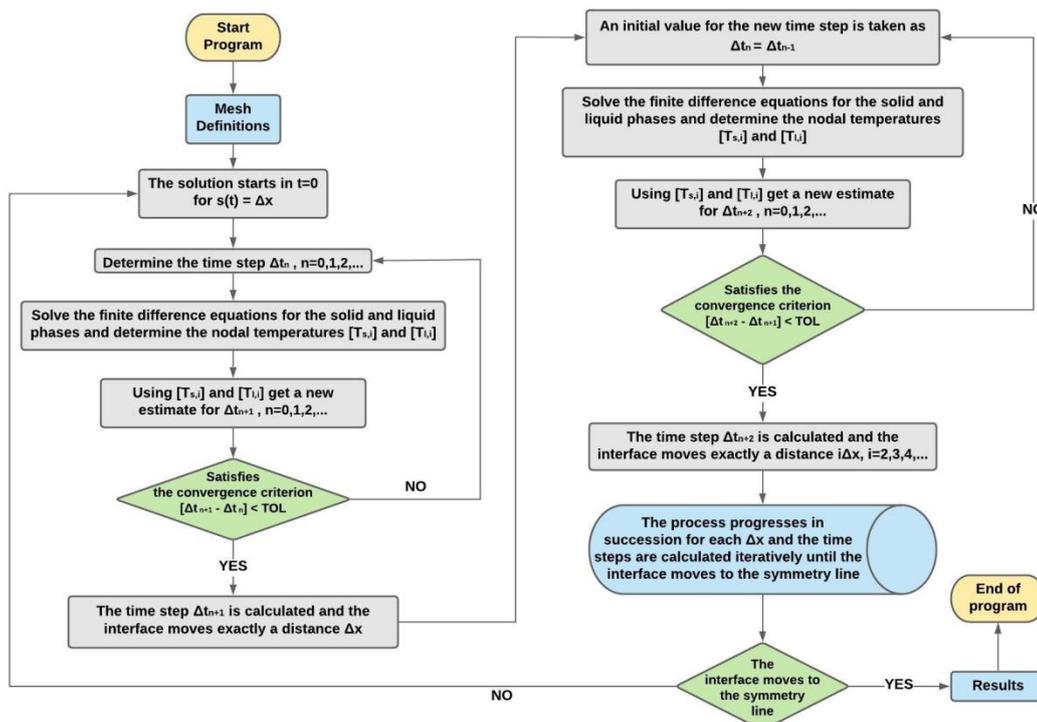

Figure 3. Block diagram of the computational algorithm.

2.4 Model validation

Figure 4 shows the comparison of the predicted interface velocity with the results of Ismail *et al.* (1999) for the case of temperature on the plate surface of -14°C and spacing between cold plates of 30 mm. As can be seen, there is a good approximation between the results, indicating that the thermal model and the numerical code are adequate. The maximum deviation between the numerical values in the present model and the experiments is approximately 16.30%.
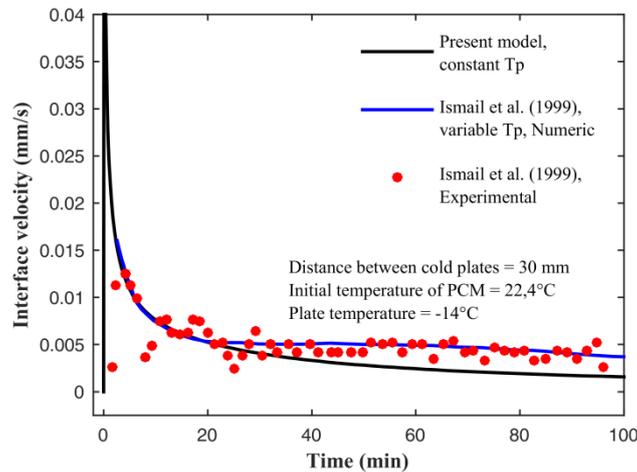

Figure 4. Comparison of the interface velocity with the numerical and experimental results of Ismail *et al.* (1999).

3. RESULTS AND DISCUSSION

In the solidification process in a storage unit with parallel plate configuration, it was considered that the phase change material, liquid water, has its initial temperature above the solidification temperature. The interface position, interface velocity, stored energy, and the time for complete solidification were simulated for the conditions described in Table 1.

Table 1. Values of plate temperature, distance from the cold plate, initial temperature PCM used in the simulations.

Parameters		Cold Plate Temperature					
		-5°C	-10°C	-15°C	-20°C	-25°C	-30°C
Distance between cold plates (2B)	5 mm to 50 mm	**Initial Temperature of the PCM: 2°C, 3°C or 4°C**					
		Interface Position (mm) **Interface Velocity (mm/min)** **Complete Solidification Time (min)** **Stored Energy (kJ)**					

Figure 5 shows the variation of the interface position with the time for the distance between cold plates of the 50 mm and six temperature values on its surface. As can be seen, the decrease in plate temperature increases the temperature gradient between the temperature of its surface and the temperature of PCM. With the value of this higher gradient, the rate of heat transfer increases and, consequently, the interface position during the process. This increase leads to a decrease in the total solidification time from 690 min at the temperature of -5°C to 150 min for the case of the surface temperature of -20°C or a reduction of about 80%, which shows the influence of the temperature on the cold surface of the flat plate in the phase change process.
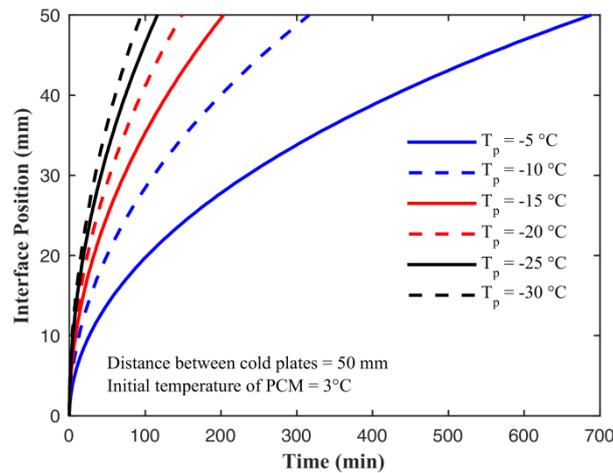
Figure 5. Effect of plate surface temperature on the interface position.

The values corresponding to the time for the complete phase change process at the different temperatures on the plate surface are shown in Table 2. These values confirm that the lower the plate temperature, the less time it takes to fully charge the storage tank, as shown by the behavior of the curves. It is also interesting to note that to obtain the phase change complete, the respective values of the total time at temperatures of -20°C, -25°C, and -30°C were relatively next when compared to the time with temperature on the plate of -5°C. Such observation is valid when it is proposed to evaluate the energy cost necessary to cool the working fluid that keeps the plates cold.

Table 2. Time for complete phase change at different temperature values on the cold plate surface.

Time for complete solidification	Cold Plate Temperature					
	-5°C	**-10°C**	**-15°C**	**-20°C**	**-25°C**	**-30°C**
Distance between cold plates (2B) **50 mm**	690 min	317 min	203 min	150 min	116 min	95 min

Figure 6 shows the variation of the interface velocity with the time for distance between cold plates of the 50 mm and six temperature values on its surface. As can be seen, initially, the interface velocity is high due to the small thermal resistance between the PCM and the plate surface. As the process progresses, the solidified mass layer increases and this resistance increases, causing the interface velocity to decrease. As the thickness of this layer increases, the resistance increases so much that the velocity becomes extremely small with practically no advance of the interface position.
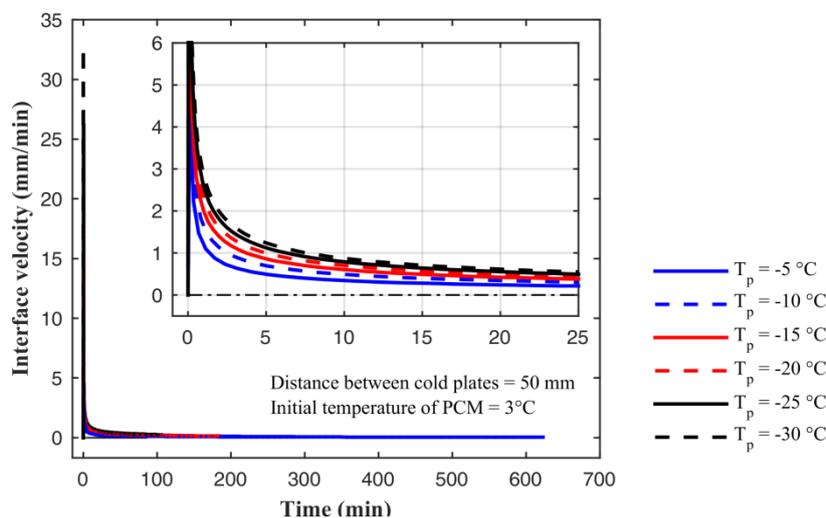

Figure 6. Effect of plate surface temperature on the interface velocity.

Figure 7 shows the variation of the predicted time for complete solidification with the distance between cold plates for different temperatures on the surface. As can be seen, the greater distance between plates increases the thermal resistance between its cold surface and the moving interface, which also reduces the heat transfer rate and the interface velocity, resulting in a long time for the complete phase change.
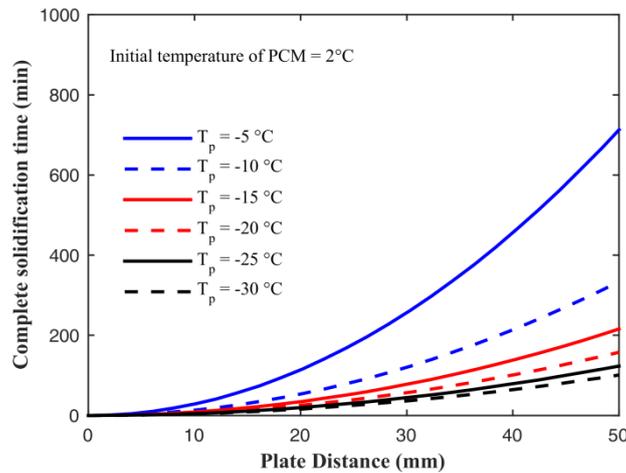

Figure 7. Variation of the complete solidification time with the cold plate distance.

Figure 8 shows the variation of the stored energy with time for temperature on the plate surface of -10°C, the distance between cold plates of 50 mm, and the initial temperature of the PCM of 4°C. During the charge of the ice bank, the solidification front moves, initially with more velocity, and then it decreases until the PCM solidifies completely. The stored energy is composed of sensible heat due to the decrease temperature of the PCM and latent heat due to the water change phase. As can be seen, the stored sensible heat is too little, and most of the total energy is due to latent heat, which characterizes these energy storage systems.
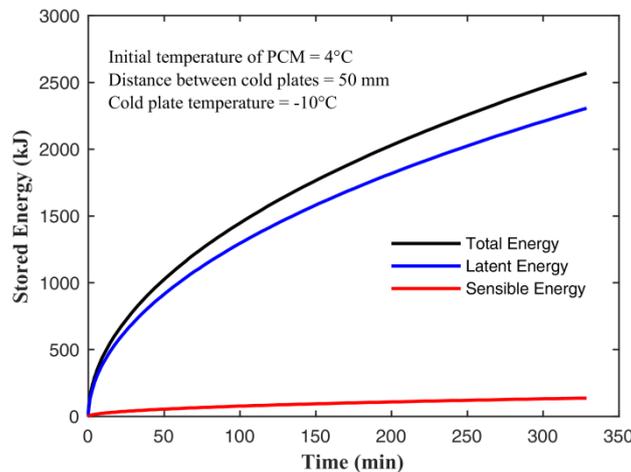

Figure 8. Variation of the stored thermal energy with the time during solidification of PCM.

4. CONCLUSIONS

In this work, a thermal model based on pure conduction was analyzed for the phase change process of water in a latent energy storage system with a parallel flat plates configuration. From the numerical results, it was possible to evaluate the solidification process through its thermal parameters such as interface position, interface speed, stored energy, and complete solidification time. As the temperature on the surface of the plate decreases, there is an increase in the heat transfer rate and a decrease in the full solidification time due to the growth in the gradient between the temperatures. For higher values of distance between cold plates, there is an increase in the time for the complete phase change due to the decrease in the interface velocity. When compared to the work of Ismail *et al*. (1999), this study presented a different solution method through the finite difference approach in the solution of the governing equations that describe the solidification process. Also, the present manuscript includes nanomaterials as PCM. Thus, the analyzed

model contributes some data that can be considered relevant in these types of flat plate latent heat storage configurations and be useful for designers in the area.

As suggestions for future work, it would be appropriate to develop a new thermal model and implement a computational code evaluating the case of flat finned plates in the study of storage units from the two-dimensional heat conduction formulation involving enthalpic methods. One of the advantages of including fins in a rectangular cavity is the improvement in heat transfer, which results in shorter times for the solidification process of the PCM. This improvement is due to the possibility of using materials with high thermal conductivity in these geometries. The major disadvantage of extending this study to two-dimensional configurations is the infeasibility of the MVTS method in calculating the exact location of the interface position in the 2D domain of the rectangular geometry. Some geometric parameters of the rectangular cavity, such as width, fin length, fin thickness and thickness of the top and bottom, are used in the phase change processes in two-dimensional configuration. The solution to the governing equations of these phenomena involves enthalpy methods and similar methods.

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