A THERMAL ACCUMULATOR MODEL APPLIED TO BUILDING WALLS

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Abstract. *Geographic regions with significant temperature changes throughout the day require intelligent solutions to improve the thermal comfort of buildings without increasing the energy consumption of air conditioners or heaters. One alternative is to apply phase change materials (PCMs) into the wall structure, which could then act as thermal capacitors (or temperature modulators), reducing the temperature variation inside the building. This paper presents a simplified model to predict the thermal behavior of a room with walls subjected to thermal cyclic loading in which different types of PCM are applied. The governing equations and physical assumptions that form the model are detailed and its predictions are validated through comparisons with results obtained with a commercial code.*

Keywords*: Phase change materials, Thermal accumulator model, Thermal cyclic loading*

1. INTRODUCTION

There are many cities in the world subjected to high diurnal temperature variations. These variations are more significant in desert areas, where the average annual precipitation is relatively low. In these locations temperature control is fundamental for comfort improvement inside buildings. However, the intensive use of air conditioners and heaters is always associated with an increase in energy consumption and, therefore, a more sustainable solution should be sought.

An alternative to overcome the existent mismatch between supply and demand of energy would be to store the thermal energy entering the walls during the day, keeping the internal ambience cooler, and release this energy during the night so as to warm the same ambience. Therefore, a thermal energy storage (TES) device is needed. The most common way of storing thermal energy is in the form of sensible heat. Energy is stored in a device as sensible heat by the increase in its temperature. The water is an example of energy sensible heat storage material as it modulates the temperature in places where the diurnal temperature variations are not high, absorbing energy during the day and delivering it during the night to the ambiences near it. Nevertheless, it is also possible to store thermal energy in the form of latent heat or binding energy. The former process is characterized by the phase change of a material while the latter is characterized by chemical reactions.

In this paper, the application of latent heat storage devices to building walls is modeled. In order to accomplish this task, phase change materials (PCMs) are applied as an internal layer inside the walls of the building. The phase change materials are substances with high latent heat which, changing phase at a certain temperature, are capable of storing and releasing large amounts of energy. The PCM layer inside the wall would function like a thermal capacitor, modulating the temperature inside the building. During the day, the material would suffer a phase change, becoming liquid and absorbing the energy coming from the outside. At night, the energy stored would be released to the internal ambience, keeping it warm as the temperature outside becomes low. Recently, many investigations have been carried out on the application of phase change materials in building walls to improve its thermal performance (Kuznik *et al*., 2011). In addition to thermal comfort of buildings, other common applications of PCMs include solar power plants, thermal management of electronic devices and new textiles for clothes.

Although the benefits of this type of device are more evident in desert areas, they can also be applied to other regions where the temperature variation is usually high. In Brazil, this technology could be applied in cities located far from the ocean, especially the ones located in Mato Grosso, Mato Grosso do Sul, Minas Gerais, and Goiás.

Some experimental studies have been proposed to analyze the application of PCMs to buildings. Castell *et al.* (2010) analyzed the application of macroencapsulated PCM with typical construction materials for Mediterranean construction in real climate conditions. The authors found a reduction of energy consumption of about 15% and lower temperature peaks (up to 1°C) in rooms with PCM. Kuznik *et al.* (2008) investigated the application of micro-encapsulated PCM wallboards to light weight building subjected to external temperature and radiative flux controlled to simulate a summer day. It was observed that walls with PCM were able to decrease temperature fluctuations by 4.7°C.

The aforementioned experiments give evidence that building walls are a very interesting application for PCMs as it can improve the thermal comfort inside the buildings. However, in order to optimize the walls and to analyze different configurations with low cost a simple model is necessary. Due to the non-linear nature of the problem, numerical analysis is generally required to obtain appropriate solutions for the thermal behavior of these systems. The enthalpy method is the most commonly applied solution approach to the problem (Athienitis *et al*., 1997; Zivkovic and Fujii, 2001; Zhou *et al*., 2010). In such a method there is no need for tracking the phase change boundary throughout the solution domain since the governing equation is the same for the two phases. Therefore, the interface conditions between these phases are automatically considered. In conjunction with the enthalpy method, the finite difference method (Athienitis *et al*., 1997; Zivkovic and Fujii, 2001; Zhou *et al*., 2010), the finite volume method and even the finite element method (Carbonari *et al*., 2006) are techniques usually employed. In addition to the enthalpy method, some authors have relied on a heat capacitance method, in which the energy equation is only a non-linear function of temperature in a source term (Dutil *et al*., 2011).

The objective of this paper is to present a simplified numerical model to analyze the application of PCMs to building walls. The importance of this model is justified by the fact that the commercially available models are scarce and computationally expensive. Based on the model developed, an analysis of the application of PCM to building walls subjected to temperature cyclic conditions of desert areas is conducted. Firstly, the mathematical model is presented and its main hypotheses discussed. Then, the model is numerically validated through a comparison of its predictions with results obtained with a commercial model. Finally, the paper presents an analysis of variations of thermal load resulted from the application of different PCMs to the walls of a typical room.

2. MODELING

The numerical analysis of heat transfer across a wall filled with PCM is carried out with a numerical model similar to the one described by Zivkovic and Fujii (2001). However, a fully explicit discretization has been adopted here instead in order to simplify the corrections required when the control volume is subjected to sensible and latent heat transfer during the same timestep.

2.1. Physical problem

The problem of a body subjected to heat transfer with phase change can be imagined as a region Ω in space formed by the union of two domains, namely a solid domain and a liquid domain, for the case of a solid-liquid phase change or vice-versa. The boundary conditions that define these domains can be divided in three parts, firstly the boundary dividing the solid domain from space, secondly the boundary dividing the liquid domain from space, and lastly the boundary separating the solid and liquid domains, represented by Σ in Fig. 1.

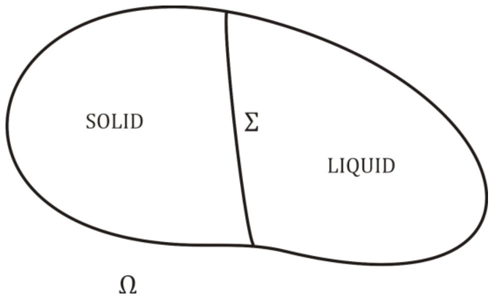

Figure 1. Domain to be modeled. Adapted from Alexiades and Solomon (1993)

An important characteristic of problems with phase change is that in addition to the temperature, pressure and velocity fields, the position of the phase change interface Σ is also not known *a priori*. This type of phenomenon is usually referred to as moving boundary problems. Besides the non-linearity associated with the interface position, this problem has many other difficulties related to the physical problem itself, characterized by heat and mass transfer, absorption or release of latent heat, sub-cooling, variation of thermo-physical properties, surface effects, among others. The interface thickness can also vary, imposing an additional difficulty for the modeling. This thickness can be a few angstroms or even some centimeters. Moreover, the interface can have a very complex structure.

With so many physical phenomena occurring simultaneously, a complete modeling of the phase change problem is a very cumbersome task. Therefore, many authors have reported different models in accordance with their respective physical problems. A very well known model is the Stefan problem, whose main assumptions are summarized in Tab. 1.

The Stefan problem is very similar to the model presented by Zivkovic and Fujii (2001), except by the fact that in the latter different density values for the two phases are considered. In their model, Zivkovic and Fujii (2001) employed the enthalpy method in which the enthalpy defines the physical state of each control volume and the interface position is not important. Such hypotheses are also adopted in the model described herein.

Table 1. Stefan problem hypotheses. Adapted from Alexiades and Solomon (1993)

Physical Factors Involved in Phase Change Processes	*Simplifying Assumptions for the Stefan Problem*	*Remarks on the Assumptions*
1. Heat transfer by conduction, convection, radiation; possible gravitational, elastic, chemical and electromagnetic effects	Heat transfer isotropically by conduction only, all other effects assumed negligible.	Most common case. Very reasonable for pure materials, small container, moderate temperature gradients.
2. Release or absorption of latent heat	Latent heat is constant; it is released or absorbed at the phase-change temperature.	Very reasonable and consistent with the rest of the assumptions.
3. Variation of phase-change temperature	Phase-change temperature is a fixed known temperature, a property of the material.	Most common case, consistent with other assumptions.
4. Nucleation difficulties, supercooling effects	Assume not present.	Reasonable in many situations.
5. Interface thickness and structure	Assume locally planar and sharp (a surface separating the phases) at the phase-change temperature.	Reasonable for many pure materials (no internal heating present).
6. Surface tension and curvature effects at the interface	Assume insignificant.	Reasonable and consistent with other assumptions.
7. Variation of thermo-physical properties	Assume constant in each phase, for simplicity.	Reasonable for most materials under moderate temperature range variations; discontinuity across the interface.
8. Density changes	Assume constant.	Assumption to avoid motion of material. Possibly the most questionable assumptions.

2.2. Mathematical model

For solid-liquid phase change variations of energy associated with swelling are small and, hence, not taken into account in the present model. Consequently, the only phenomenon related to energy variations is heat transfer and the energy conservation equation can be expressed as follows (Incropera and DeWitt, 2003):

$$\rho \cdot \frac{\partial H}{\partial t} = \mathrm{div}\left(k \cdot \nabla T\right) \tag{1}$$

where ρ is the density of the PCM, H is the total volumetric enthalpy, k is the thermal conductivity and T is the temperature.

For heat transfer problems with phase change the enthalpy can be split in two components, one related to the sensible heat transfer and other related to the latent heat transfer involved in the process. As a result, the enthalpy can be represented as:

$$H = h + L \cdot f_l \tag{2}$$

The first term in the right hand side stands for thermal energy in the form of sensible heat and can be expressed as:

$$h = \int c_p \cdot dT \tag{3}$$

where c_p is the specific heat capacity at constant pressure.

On the other hand, the second term in Eq. (2) represents the thermal energy in the form of latent heat, where L is the latent heat related to the phase change process and f_l is the liquid fraction of the material. For a phase change process at constant temperature the following conditions must be followed:

$$f_l = \begin{cases} 1 & \text{if} \quad T > T_m \\ 0 & \text{if} \quad T < T_m \end{cases} \tag{4}$$

where T_m is the phase change temperature.

Substituting Eq. (2) into Eq. (1) and by using Eq. (3), a general equation for heat transfer with phase change can be obtained:

$$\rho \cdot c_p \cdot \frac{\partial T}{\partial t} + \rho \cdot L \cdot \frac{\partial f_l}{\partial t} = \mathrm{div}(k \cdot \nabla T) \tag{5}$$

For a one-dimensional problem, this equation assumes the following form:

$$\rho \cdot c_p \cdot \frac{\partial T}{\partial t} + \rho \cdot L \cdot \frac{\partial f_l}{\partial t} = \frac{\partial}{\partial x}\left(k \cdot \frac{\partial T}{\partial t}\right) \tag{6}$$

Equation (6) can assume different forms if only sensible or latent heat transfer is involved in the process. In the former case, as $T > T_m$ or $T < T_m$, f_l is constant and equal to 1 or 0, respectively. For this situation the derivative of f_l with respect to time is null and the second term in the left hand side vanishes. In the latter case, latent heat transfer occurs only when $T = T_m$, then the first term in the left hand side vanishes. Therefore, the heat transfer with phase change problem is governed by different governing equations according to the temperature range, as depicted in Fig. 2.

Even with the described assumptions, an analytical solution for Eq. (6) is not possible due to high nonlinearities of the problem. This means that the problem must be solved numerically.

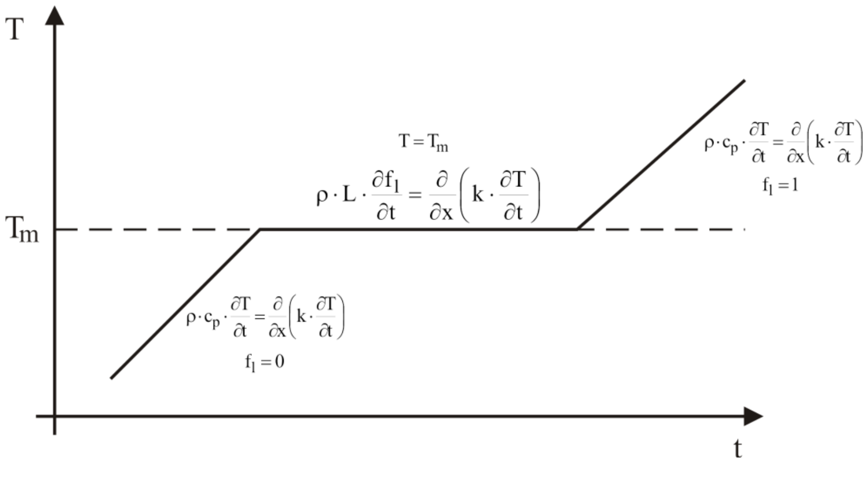

Figure 2. Mathematical conditions.

2.3. Numerical model

In the present paper, a fully explicit finite volume method is adopted to solve Eq. (6). The physical domain must be divided into n control volumes with density ρ, specific heat capacity at constant pressure c_p, thermal conductivity k, and latent heat L. For a control volume centered in P the resulting configuration is shown in Fig. 3.

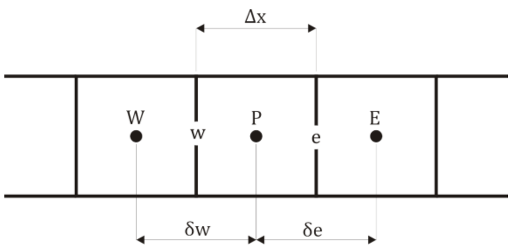

Figure 3. Discretization scheme.

As a one-dimensional formulation is employed, there are only two control volumes in the neighborhood of control volume identified by P, namely W and E, which stand for the west and east volumes. The length of the control volume P is Δx and the distances between the centers of P and W and P and E are δw or δe, respectively.

In order to apply the finite volume method, the governing equation must be integrated in time and space in each control volume. For the control volume P, the corresponding general integral equation becomes:

$$\int_{t}^{t+\Delta t}\int_{x_w}^{x_e}\left(\rho \cdot c_p \cdot \frac{\partial T}{\partial t} + \rho \cdot L \cdot \frac{\partial f_l}{\partial t}\right)\cdot dx \cdot dt = \int_{t}^{t+\Delta t}\int_{x_w}^{x_e}\frac{\partial}{\partial x}\left(k \cdot \frac{\partial T}{\partial t}\right)\cdot dx \cdot dt \tag{7}$$

If the enthalpy and the liquid fraction are admitted to be constant in each control volume:

$$\int_{t}^{t+\Delta t}\left(\rho \cdot c_p \cdot \frac{\partial T}{\partial t}\cdot \Delta x + \rho \cdot L \cdot \frac{\partial f_l}{\partial t}\cdot \Delta x\right)\cdot dt = \int_{t}^{t+\Delta t}\left\{\left(k \cdot \frac{\partial T}{\partial t}\right)\Big|_{x_e} - \left(k \cdot \frac{\partial T}{\partial t}\right)\Big|_{x_w}\right\}\cdot dt \tag{8}$$

The development of the right hand side of this equation can change in accordance with the position of the control volume since it represents the heat fluxes through the interfaces w and e. For the volume P, Eq. (8) can be expressed by using an explicit time discretization as follows:

$$\rho \cdot c_p \cdot \left(T^{t+\Delta t} - T^t\right)\cdot \Delta x + \rho \cdot L \cdot \left(f_l^{\,t+\Delta t} - f_l^{\,t}\right)\cdot \Delta x = \left[k_e \cdot \frac{\left(T_E^t - T_P^t\right)}{\delta e} - k_w \cdot \frac{\left(T_P^t - T_W^t\right)}{\delta w}\right]\cdot \Delta t \tag{9}$$

where k_w and k_e represent the thermal conductivity evaluated at the west and east interfaces, respectively.

If the control volume undergoes only sensible heat transfer, Eq. (9) can be simplified to:

$$\rho \cdot c_p \cdot \left(T^{t+\Delta t} - T^t\right)\cdot \Delta x = \left[k_e \cdot \frac{\left(T_E^t - T_P^t\right)}{\delta e} - k_w \cdot \frac{\left(T_P^t - T_W^t\right)}{\delta w}\right]\cdot \Delta t \tag{10}$$

Yet, in the case of latent heat transfer, Eq. (9) assumes the following form:

$$\rho \cdot L \cdot \left(f_l^{\,t+\Delta t} - f_l^{\,t}\right)\cdot \Delta x = \left[k_e \cdot \frac{\left(T_E^t - T_P^t\right)}{\delta e} - k_w \cdot \frac{\left(T_P^t - T_W^t\right)}{\delta w}\right]\cdot \Delta t \tag{11}$$

If the control volume undergoes both the sensible and the latent heat transfer, then some corrections, described by Zivkovic and Fujii (2001) must be applied.

The application of a fully explicit discretization can be justified by the fact that it is much easier to apply the corrections to the equations when these equations are solved individually, volume by volume, than when the equations for the complete domain are solved simultaneously. In fact, wrong predictions of temperature for a certain control volume will negatively affect the results of temperature of neighbor volumes in the case of an implicit discretization. On the other hand, eventual errors in the explicit discretization are readily corrected and are not propagated throughout the solution domain.

3. NUMERICAL VERIFICATION

The verification of the model described herein was accomplished by comparing its predictions for a standard problem with the results obtained with the commercial code Fluent 12.1 (ANSYS, 2009). A small solution domain was chosen for this purpose since simulations of heat transfer with phase change are very expensive in the case of using a commercial code.

The problem is represented by a wall with thickness and height equal to5mm and 20mm, respectively, filled with PCM and subjected to cyclic convective boundary conditions. Heat transfer coefficients are considered to be constant to speed up the solidification/fusion cycle and to reduce the simulation time. Such convective boundary conditions are described in Tab. 2. Buoyancy driven flow are also taken into account so as to better represent the physical problem. The PCM domain is composed of a SP-25 A8 material (see Tab.3). The solution domain is divided into 2500 quadrilateral volumes.

For comparison purpose, results with both models are represented for the temperature and liquid fraction variations along time in the midpoint of the domain. Figure 4 and Fig. 5 show the temperature and liquid fraction comparisons, respectively. As can be seen, there is a close agreement between the results obtained with the two models, providing evidence that the model described in this paper was correctly implemented. Therefore, the model can now be applied to problems computationally more expensive.

Table 2. Convective boundary conditions for validation

Time interval (minutes)	*Heat Transfer Coefficient ($W/m.^2$°C)*	*Free Stream Temperature (°C)*
0 – 10	100	0
10 – 20	100	50

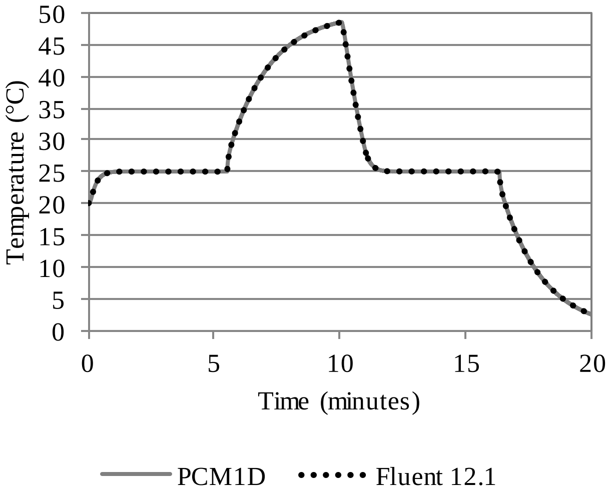

Figure 4. Midpoint temperature variation

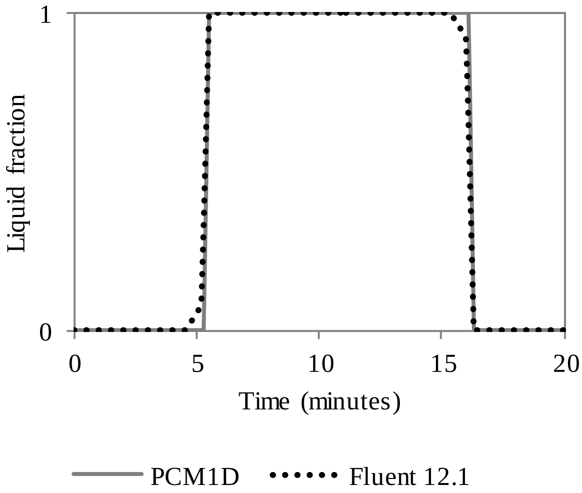

Figure 5. Midpoint liquid fraction variation

4. APPLICATION TO BUILDING WALLS

In order to analyze the potential of PCM as an alternative of reducing the thermal load of buildings located in desert areas, the corresponding cyclic variation of temperature along the day must be specified. Figure 6 presents the recorded temperature in Tucson, Arizona, during a day of May 2010. Tucson is the second biggest city near Sonoran Desert, one of the deserts in the United States with the highest temperatures and a dry arid climate. The annual precipitation is usually less than 15 inches and the average annual temperature is over 18°C.

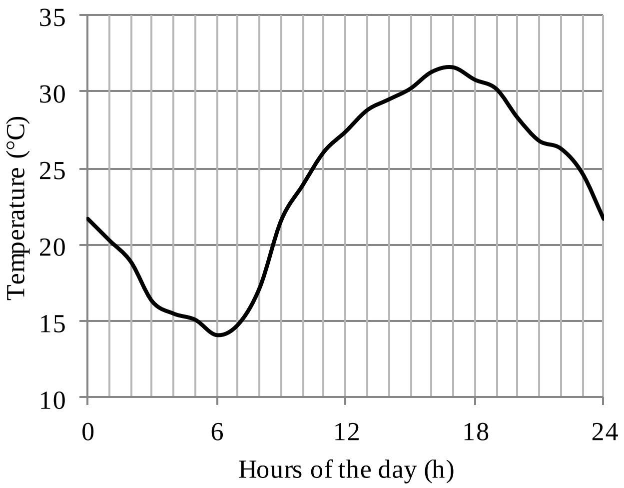

Figure 6. Temperature in Tucson, AZ in 15/05/2010. Source: National Climatic Data Center of United States.

As already discussed, the use of PCM in building walls can be especially beneficial in locations with such climatic conditions, because its high-energy storage density can reduce the mismatch between supply and demand of energy. PCMs can be enclosed in walls through different forms. In the present analysis, the wall (4m x 3m) is formed by two layers of concrete (50mm each) separated by a layer of PCM with a thickness of 5mm. Different types of PCM were considered in the analysis in order to compare their performances with a conventional wall without PCM, made of concrete only (100mm of thickness). The thermal characteristics of all materials used in this analysis are shown in Tab. 3. The material named RT 27 corresponds to a paraffin, while the other PCMs considered correspond to different blends of salt hydrates and paraffin (RUBITHERM, 2006, 2009a, 2009b).

The performance of different PCMs to reduce the thermal load was carried out with the one-dimensional numerical model just described. Results for the temperature at the internal surface are used to compare the outcome of each arrangement. The internal and external surfaces of the wall are subjected to natural convection. The air temperature inside the room is considered constant and equal to 20°C while the external temperature was described via Fig. 6. A schematic view of the problem and its associated electrical circuit analogy are shown in Fig. 7.

Table 3. Thermal characteristics of PCMs

Material	*Phase Change Temperature (°C)*	*Latent Heat (kJ/kg)*	*Density (kg/m^3)*	*Specific Heat Capacity (kJ/kg.K)*	*Thermal Conductivity (W/m.K)*
concrete	-	-	400	0.92	0.16
SP 22 A17	22	150	1430 – 1490	2.10*	0.60
SP 25 A8	25	180	1380	2.50	0.60
RT 27	27	179	750 - 870	1.80 – 2.40	0.20

(*) data based on average value for paraffin.

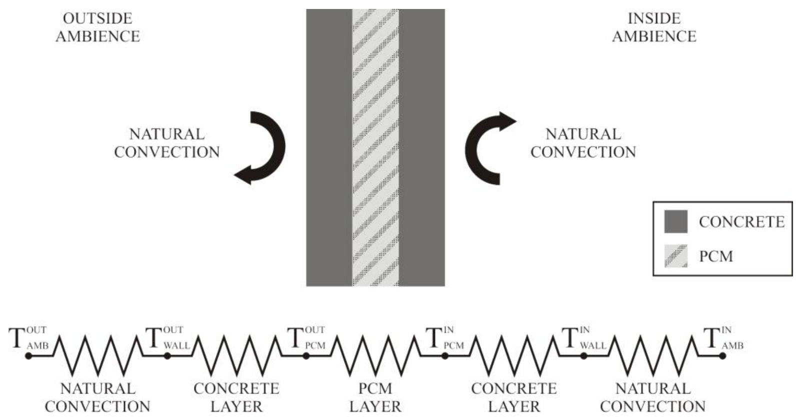

Figure 7. One-dimensional modeling of the physical problem with its electrical equivalent circuit.

5. RESULTS

Results for temperature at the internal surface are registered for all configurations during the period of one day after fully cyclic conditions are verified in the simulation. Figure 8 shows predictions for temperature at the internal surface when applying different PCMs and a plain wall of concrete. As expected, the wall made of concrete is characterized by the highest temperature variations, reaching a difference of almost 9°C between the lowest and the highest temperature. It is interesting to note that the application of RT 27 did not reduce the temperature variation significantly. This can be explained by the fact that the RT 27 layer did not reach the melting temperature. As a result, no energy was stored in the form of latent heat and the modification of the wall thermal behavior was very small.

Although the RT 27 performed poorly, the application of SP 25 A8 and SP 22 A17 contributed considerably to the reduction of temperature oscillation. The baselines at 25°C and 22°C for SP 25 A8 and SP 22 A17, respectively, confirm that these materials experienced a phase change process, increasing the thermal energy storage capacity of the wall.

The application of SP 25 A8 permitted a reduction of about 1°C in the temperature oscillation. It represents a reduction of more than 10% in the temperature range. However, if the liquid fraction variation in the outermost sub-layer of this material is considered (Fig. 9), it can be concluded that the material SP 25 A8 did not experience a complete phase change process. Therefore, the latent heat storage capability of this material has not been used completely.

It can be verified from Fig. 8 that the most useful configuration is that with a layer of SP 22 A8, for which the temperature oscillation is reduced to less than 1°C. This is consequence of the fact that the melting temperature of this material is closer to the average temperature of the cyclic outside temperature condition. Figure 9 shows that the latent heat storage capacity of this material was almost completely used as indicated by the liquid fraction in the outermost sub-layers of this PCM, varying from 0 (completely solid) to 1 (completely liquid).

If a 16 m^2 room formed by four walls subjected to the same external temperature variation is considered, and the heat fluxes through the walls are computed, it is possible to estimate the energy consumption required for the air conditioning to maintain the room temperature at 20°C. Such estimates are presented in Fig. 10 for the different configurations of walls. As can be seen, there is a great reduction of energy consumption when the walls employ PCMs, especially for the case of a 5mm layer of SP 22 A17, which is capable to reduce the consumption by 48.7%.

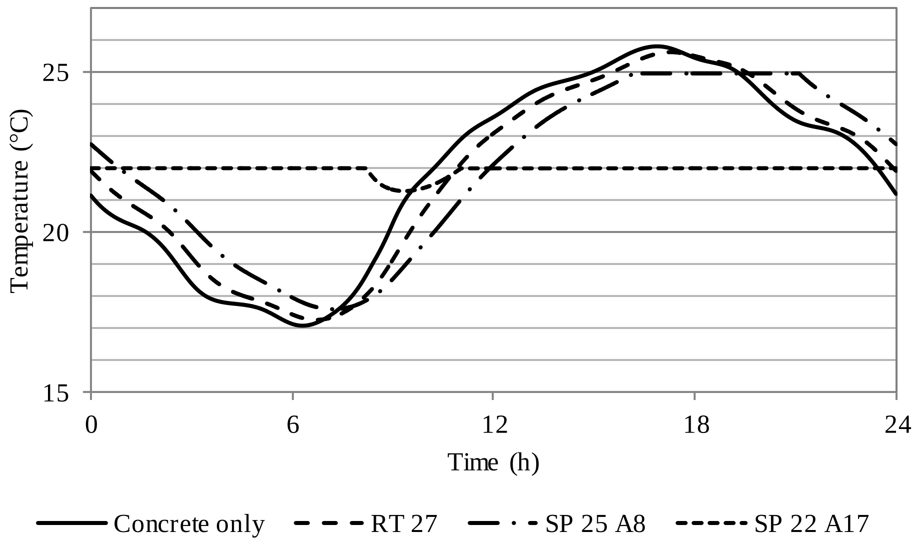

Figure 8. Temperature variation for different configurations.

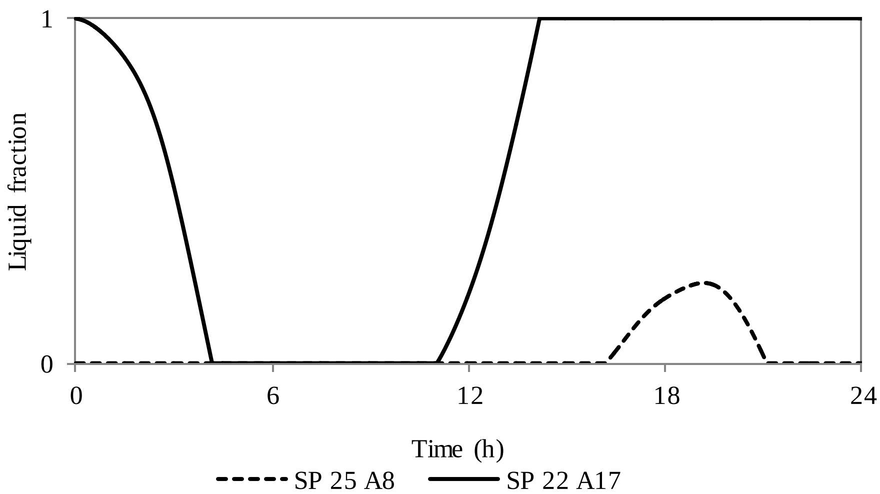

Figure 9. Liquid fraction variation in the outermost sub-layer of SP 25 A8.

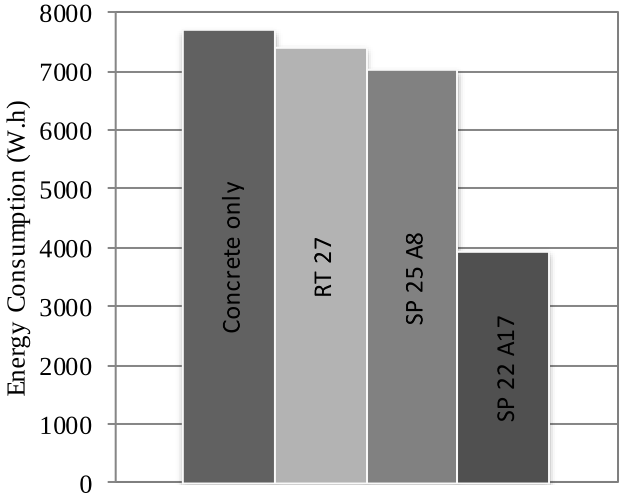

Figure 10. Energy consumption in a 16m^2 room.

6. CONCLUSION

This paper has presented a one-dimensional model developed to simulate heat transfer problems in the presence of phase change processes. The main advantage of this model is the possibility of simulating such phenomena with very low computational cost in comparison to existent commercial codes. Therefore, the application of the model is very convenient for simulation of long cycle periods. Based on the model developed, the application of phase change materials was analyzed for the purpose of reducing the thermal load of a typical room of buildings located in desert areas. It has been demonstrated that a suitable choice of the material can reduce the oscillations of wall temperature considerably. For the PCM layer composed of SP 22 A17 it was observed a significant reduction of thermal load by 48.7%. Future studies could take into account the solar thermal radiation on the external walls.

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8. RESPONSIBILITY NOTICE

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