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TEMPERATURE PROFILE ANALYIS OF POLYURETHANE AND POLYSTYRENE BOARDS APPLIED IN COLD CHAMBERS

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Abstract. *Cooling Chambers are spaces for storage and preservation of various products and foods. The growing quest for energy costs reduction in cold rooms drives several studies, among which stands out the correct choice of building materials, the largest portion of thermal load. In the present study was rated temperature profiles in panels or boards of polyurethane (PU) and expanded polystyrene (EPS) with various thicknesses. These materials widely used in cooling Chambers. To this end, we used the explicit finite difference method using the MATLAB software for the assessment of external and internal intermediate temperatures.*

Keywords *Polyurethane plates, Polystyrene plates, temperature profiles, finite difference method.*

1. INTRODUCTION

Eurostat, 2010 comments that the cold chambers has gained increasing influence to answer global demand, given the growing quest for quality of products and their varieties. To maintain the right conditions of temperature and humidity inside a cooling chambers, it is necessary to remove the thermal loads (heat sensitive and/or latent heat) generated within internal environment. Various forms of the heat are generated inside the chamber and the greater portion corresponds to the transmission of the heat by conduction through the opaque lock. Therefore, the correct selection of construction materials and design of thermal load is important to minimize the heat generated and consequently minimize energy expenditures.

2. COLD CHAMBERS

Cold chambers are storage space for product conservation . Second (Dick, 2011) the cold chambers are classifield in the commercial refrigeration sector, which guarantees the quality of various products, keeping your organic chemical and physical characteristics for control of temperature and relative humidity. The opaque lock can be composed of various materials Fig 1, among which stand out for your high performance and cost-effective, the PU and EPS.

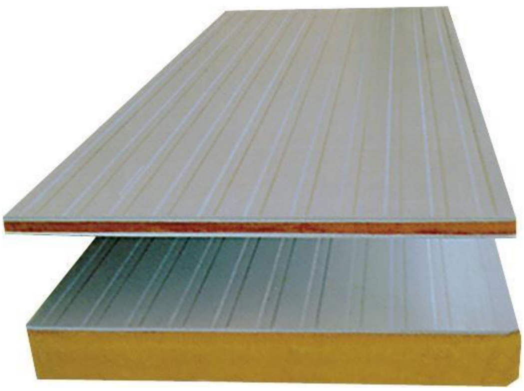

Figure 1. Polyurethane (PU) and expanded polystyrene (EPS) panels.

The cold chambers, according to its temperature, are classified in cooling and freezing chambers. Cooling Chambers are used with temperatures up to 0° C with high humidity and is aimed at conservation of fresh products, to artificial atmospheres and to control the ripening of fruits and vegetables products. The freezing chambers are used with temperatures below 0° C and are used for food preservation, among others, for a long period of time.

The search for energy costs reduction in cold chambers involves various aspects, namely, the choice of proper closure materials, design, camera placement in relation to solar radiation and the correct selection of equipment are some parameters that should be prioritized for this purpose. A study developed by (Melo, 2014) showed the characteristics of surrounding cooling chambers and concludes that a suitable building could be crucial in the final condition in terms of energy or economical. For the proper development of a cold chamber must make the correct thermal load lifting.

2.1 Thermal load calculation

When products enter an environment they are cooled with a higher temperature to the inside of the camera. This excess energy associated to other heat sources must be removed by the cooling system. The calculation of these thermal loads must be removed and both must be done thorough survey of the sources originate from these additions.

The determination of thermal load ensures the correct selection of equipment and also serves as a parameter to obtain construction materials. If the undersizing of thermal load the equipment does not take account of the cooling power required and the appropriate conditions for the preservation of products will not be affected, under penalty of loss of products and other damage. On the other hand the super thermal load requires scaling the process, as well as using more expensive equipment, they will consume more energy with many losses that tend to decrease the useful life of the equipment.

A cold chamber wins heat by infiltration through the doors and vents, by heat conduction through walls, ceiling and floor, by the presence of people and equipments, by the local lighting and for the product to be stored.

The quantity of heat transmitted by conduction through walls, ceiling and floor corresponds to the largest portion of thermal load. This depends on the surface portion subject to solar effects, the type of material and thickness of the material opaque and closing of internal and external temperature difference. Special care must be taken to choose the thickness of the thermal insulation..

2.2 Heat Conduction

Heat conduction is the transfer of energy from the more energetic particles of a substance for the less energetic particles, due to the interactions between the particles (Çengel, 2012). Heat flow by conduction, according to Fourier's law Eq. (1) is directly proportional to the temperature gradient, the coefficient of thermal conductivity and the area.

Equations should be referred to either as “Eq. (1)” in the middle of a phrase or as “Equation (1)” in the beginning of a sentence. Matrix and vector quantities can be indicated either by brackets and braces, as in Eq. (1), or in bold style, as in Eq. (2). Symbols used in the equations must be defined immediately before or after their first appearance.

One blank line must be included above and below each equation.

$$q = -k.A.\frac{\partial T}{\partial x} \tag{1}$$

Where q is the rate heat transfer by conduction, k is the thermal conductivity of the material, A is the perpendicular area to the direction of the heat flux and $\frac{\partial T}{\partial x}$ is the temperature gradient.

For the constant thermal conductivity has the Eq. (2) summarized.

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} + \frac{e_g}{k} = \frac{1}{\alpha}\frac{\partial^2 T}{\partial t} \tag{2}$$

Where α is the thermal diffusivity of the material $\frac{k}{\rho c_p}$.

For the case of the one-dimensional heat transfer in the direction (x) in the rectangular coordinates, transient and without internal power generation Eq. (3) has:

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha}\frac{\partial^2 T}{\partial t} \tag{3}$$

2.3 Finite difference method

Second (Çengel, 2012) the finite difference numerical method is used as an alternative approach for approximation of the solution of a differential equation. Consist in to transform the resolution of a partial equation on a set of algebraic equations, replacing the derivative by their differences.

The equation (4) approximated of the derivative, in the configuration of differences, is the finite difference expression of the first order.

$$\frac{df(x)}{dx} \approx \frac{f(x+\Delta x)-f(x)}{\Delta x} \tag{4}$$

For a plane wall of thickness L, whitout internal generation as Fig. 2, that the wall is subdivided in M sections equal in thickness Δx separated by ($M+1$) points, 0,1,2… $m-1, m, m+1$,… M called nodal points Eq. (5), as show bellow.

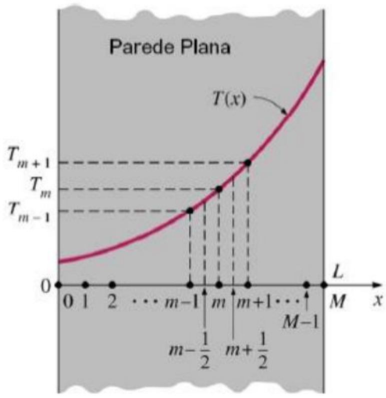

Figure 2. One-dimensional heat conduction in transient regime.

$$\Delta x = \frac{L}{M} \tag{5}$$

The equation of heat conduction in one-dimensional permanent regime has second-order derivatives of temperature in relation to the dimensional variables, and the formulation by finite difference consists in the replacement of the second order derivatives by differences.

Using the Eq. (6) the first derivative of the temperature in midpoints ($m-1/2$) and ($m+1/2$) of the section considered around the node (m) is given by Eq. (7).

$$\left.\frac{dT}{dx}\right|_{m-1/2} \cong \frac{t_m - t_{m-1}}{\Delta x} \quad \text{and} \quad \left.\frac{dT}{dx}\right|_{m+1/2} \cong \frac{t_{m+1} - t_m}{\Delta x} \tag{6}$$

The second derivative is the derivative of the firs derivative, soon:

$$\left.\frac{d^2T}{dx^2}\right|_m \cong \frac{\left.\frac{dT}{dx}\right|_{m+1/2} - \left.\frac{dT}{dx}\right|_{m-1/2}}{\Delta x} = \frac{\frac{t_{m+1}-t_m}{\Delta x} - \frac{t_m - t_{m-1}}{\Delta x}}{\Delta x} = \frac{t_{m+1} - 2t_m - t_{m-1}}{\Delta x^2} \tag{7}$$

Where the Eq. (7) is the finite difference representation of second derivative in a generic internal node m. The temperature in an internal node is expressed in terms of temperature in the node m and its neighbors. From the above we have the governing differential equation for one-dimensional heat transfer in permanent regime in a plane wall with constant thermal conductivity and internal energy generation is shown in Eq. (8).

$$\frac{d^2T}{dx^2} = \frac{\dot{g}_m}{k} \tag{8}$$

Expressing in terms of finite differences Eq. (9), comes:

$$\frac{t_{m+1} - 2t_{m-1} - t_{m-1}}{\Delta x} + \frac{\dot{g}_m}{k} = 0 \quad \text{for} \quad m = 1,2,3,...M - 1 \tag{9}$$

Where $\dot{g}_m$ is the internal heat generation rate per unit volume at node m.

For the neighbors nodes of the finite difference equations are obtained by means of the energy balance in the amount of control of this border. The energy balance Eq.(10) for the conditions of convection, radiation and heat flux combined is given by:

$$q_0 A + hA(T_\infty - T_0) + \varepsilon\sigma A(T_{surf}^4 - T_0^4) + kA\left(\frac{T_1 - T_0}{\Delta x}\right) + \frac{\dot{g}_0}{k} + A\left(\frac{\Delta x}{2}\right) \tag{10}$$

Where q_0 is the heat flux specified, h is the convection coefficient, T_∞ is the temperature of the surroundings, ε is the emissivity, σ is the Stefan Boltzman constant.

On heat conduction in one-dimensional transient regime occur variations of temperatures as a function of time and space. The finite difference solution for transient problems requires the discretization of time and space. One should adopt a step of time Δt suitable for the resolution of unknown nodal temperatures repeatedly for each Δt until the desired time is obtained solution. Taking the volume control element of an internal node m and whereas there is heat conduction of the left and right sides, the formulation for finite differences Eq. (11) for this case is shown above:

$$kA\left(\frac{T_{m-1} - T_m}{\Delta x}\right) + kA\left(\frac{T_{m+1} - T_m}{\Delta x}\right) + \dot{g}_0 A\Delta x = \rho A\Delta x c_p . \frac{t_m^{i+1} - t_m^i}{\Delta t} \tag{11}$$

Canceling the surface area A and multiplying both the terms for $\Delta x / k$ has the Eq. (12).

$$T_{m-1} - 2T_m + T_m + \frac{\dot{g}_0 \Delta x^2}{k} = \frac{\Delta x^2}{\alpha \Delta t}\left(t_m^{i+1} - t_m^i\right) \tag{12}$$

The dimensionless number knit Fourier transform is given by Eq. (13).

$$F_0 = \frac{\alpha \Delta t}{\Delta x^2} \tag{13}$$

Applying explicit formulation of finite differences and expressing the left side in step of time i and solving for new temperature t_m^{i+1} has the Eq. (14).

$$t_m^{i+1} = F_o\left(t_{m-1}^i + t_{m+1}^i\right) + \left(1 - 2F_o\right)t_m^i + F_o \frac{\dot{g}_0^1 \Delta x^2}{k} \tag{14}$$

For all internal nodes $m = 1,2,3,...(M - 1)$ in the plane wall.

The application of explicit wording for each internal node $(M - 1)$ results in $(N - 1)$ equations, and the other two equations are obtained by applying the method for both of us.

The profile of temperature and heat transfer by conduction have been assessed in PU panels and EPS for thicknesses of 50 mm and 100 mm, whereas a typical day, with hourly variations in incident radiation and outdoor temperature. For the modeling of the system and the use of the explicit finite difference numerical method, using a computational code, developed by the author in MATLAB software.

3. RESULTS AND DISCUSSIONS

The properties of the polyurethane and polystyrene panels expanded were taken from technical literature provided by manufacturers and with these data were found the thermal diffusivity. We used 10 nodes along the boards for each Panel. The nodes are listed as 0, 1, 2, 3… 10, with the node 0 on the outer surface subject to radiation and convection. The node 10 corresponds to the inner surface subject only to convection. Nodes 1 to 9 are the intermediate nodes and the formulations were given by the finite difference method, as discussed earlier.

The determination of the upper limit of the time (Δt) by the criterion of stability is required for the identification of smaller primary coefficient in the system. Boundary nodes are more restrictive than the US. For both, it was used for each situation the node 0 , as shown in the Eq. (15) and Eq. (16).

$$1-2F_o-2F_o\frac{h_e\Delta x}{k} \tag{15}$$

$$\Delta t\le\frac{\Delta x^2}{1-2F_o-2F_o\frac{h_e\Delta x}{k}} \tag{16}$$

Replacing the quantities the maximum values allowed the steps of time for proposals were obtained with these values the value of the dimensionless number of Fourier for the proposed situations.
The initial time $t=0$, the temperature of the wall varies linearly from 22.81°C to 15°C, between nodes 0 and 10. Noting that there are 10 nodal spacings of equal value the temperature change between two neighbouring nodes is given by the subtraction of the temperature of the external nodes divided by ten, obtaining this way the linear increment 0.825°C.
Nodal temperatures in $t=\Delta t=4s$ were used in all situations and were determined by the finite difference equations to the intermediate nodes and both made use of a computational algorithm in MATLAB software, where through this mechanism the hourly temperature variation in each node.

Table 1 shows the data used for the calculation in this work.

Table1. Parameters for calculation of the temperature profile.

	50 mm PU	100 mm PU	50 mm EPS	100 mm EPS
$\Delta x(m)$	0,05	0,01	0,05	0,01
$k(W/m^2.K)$	0,045	0,045	0,034	0,034
$\alpha(m^2/s)$	$2{,}65.10^{-7}$	$2{,}65.10^{-7}$	$8{,}61.10^{-7}$	$8{,}61.10^{-7}$
F_o	0,0424	0,0106	0,01377	0,0341
$\Delta t(s)$	4	4	4	4
ε	0,3	0,3	0,3	0,3
$T_i(°C)$	15	15	15	15

3.1 Hourly temperature variation for PU

Nodal temperatures in polyurethane boards with thicknesses of 50 mm and 100 mm are shown in Fig. (3) and Fig. (4), where they were checked to higher values of temperatures were obtained on the boards of thickness 50 mm.

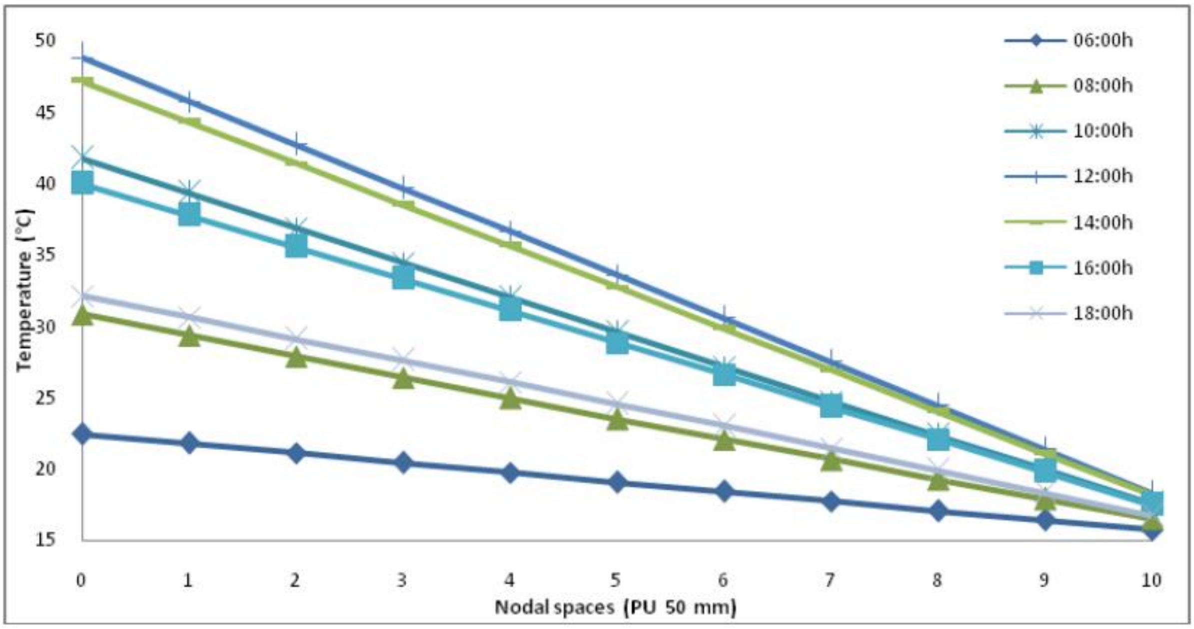

Figure 3. Variation of nodal temperature to 50 mm thick polyurethane.

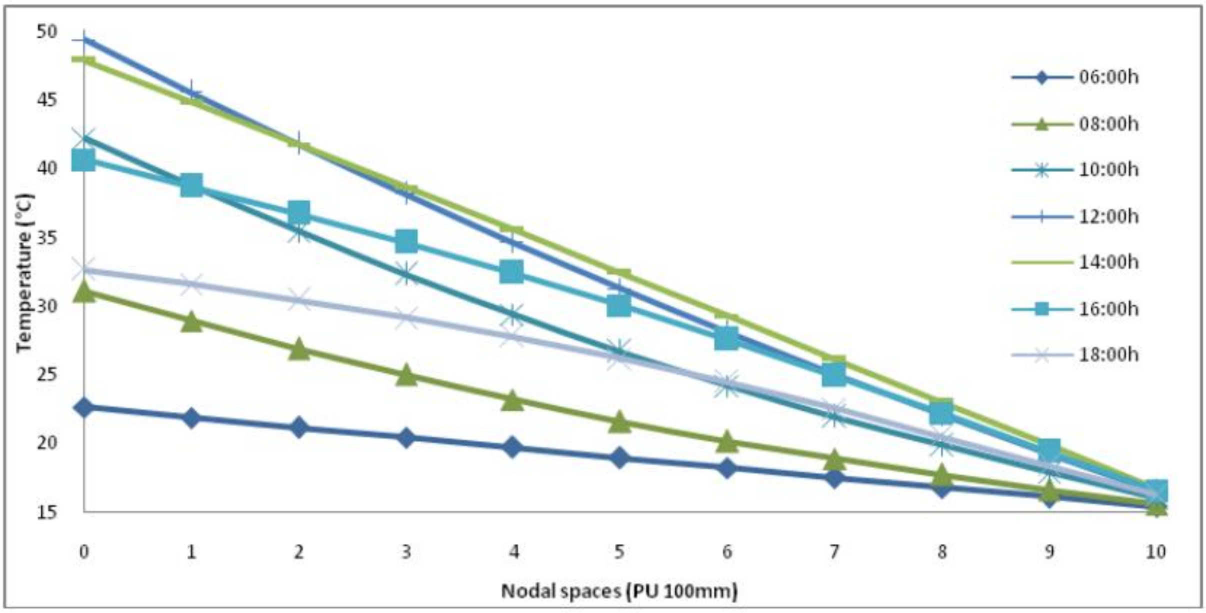

Figure 4. Variation of nodal temperature to 100 mm thick polyurethane.

The temperatures of us residents were evaluated and were obtained higher temperatures for Polyurethane plates with thickness of 50 mm. This fact has occurred largely as a function of the thickness of the material, because their properties are constant equal to each other, except the number dimensionless of Fourrier, which introduced changes to the variation of thickness of plate of PU.

3.2 Hourly temperature variation for EPS

Nodal temperatures in polystyrene plates with thicknesses of 50 mm and 100 mm are shown in Fig. (5) and Fig. (6).

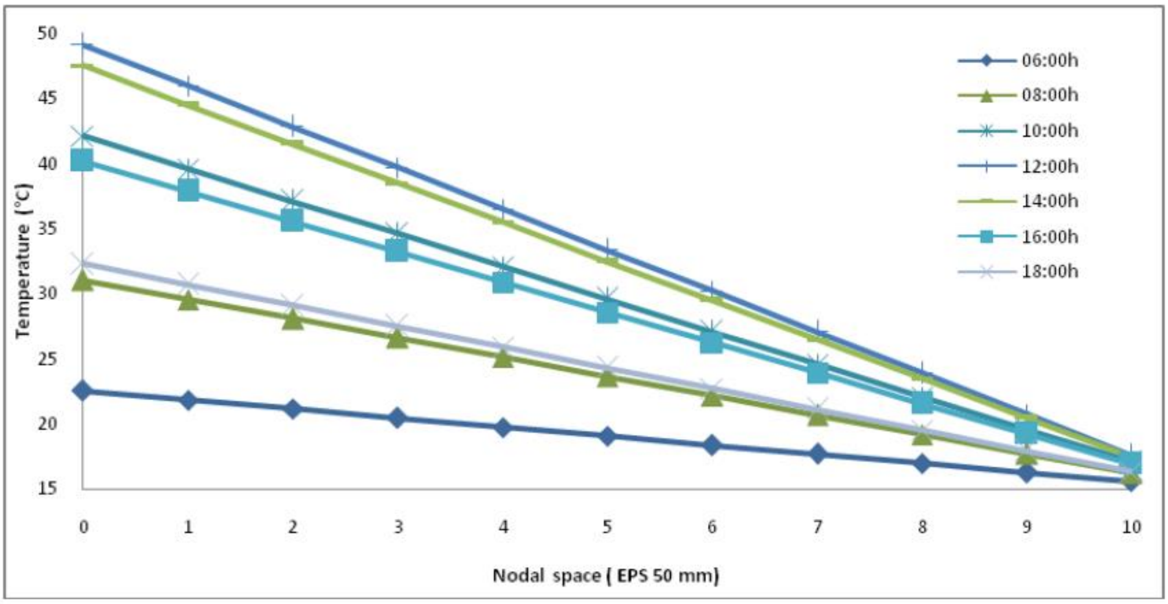

Figure 5. Variation of nodal temperature to 50 mm thick polystyrene.

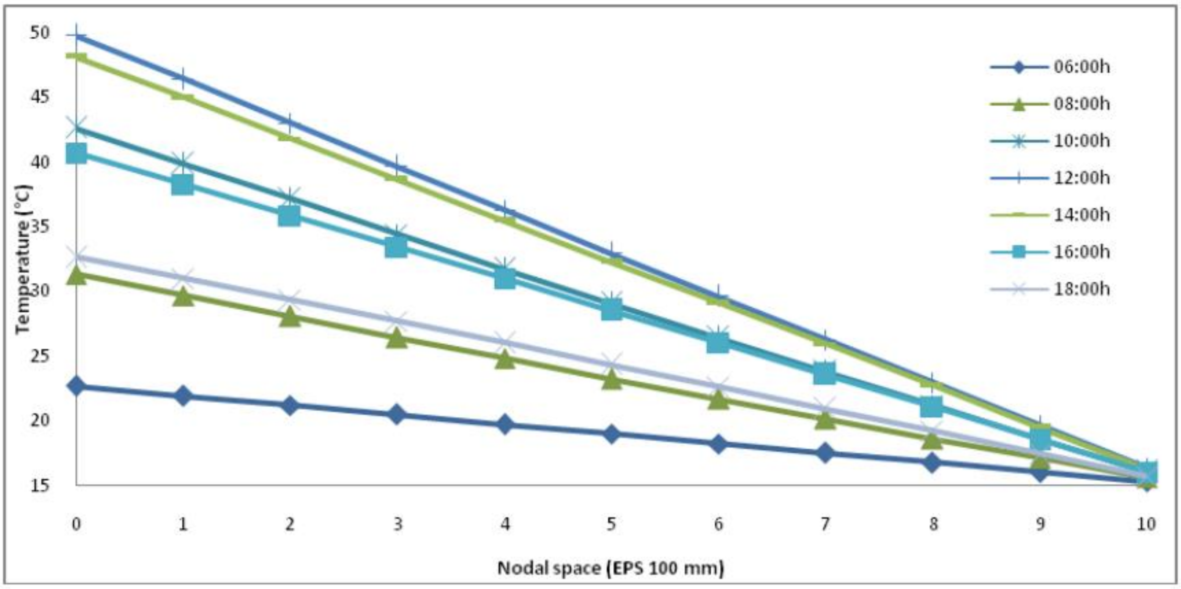

Figure 6. Variation of nodal temperature to 100 mm thick polystyrene.

Similar analysis was developed for the EPS plates where higher temperatures were obtained for the thickness of 50 mm. You can assign the result to the same factor in item previously described.

Evaluating intermediate temperatures for PU and EPS boards, it was found that the largest intermediate temperatures were obtained for the 50 mm PU boards. The lowest temperatures were recorded for EPS boards of 100 mm. You can assign these results to increase the thickness of the plates and the changes of properties between them.

3.3 Temperature analysis of internal surfaces

The temperature of the internal nodes are of important analysis, because of them depend on the heat that is transmitted to the internal environment by convection. Figure 7 shows the hourly variation of temperature of the internal nodes to PU and EPS plates.

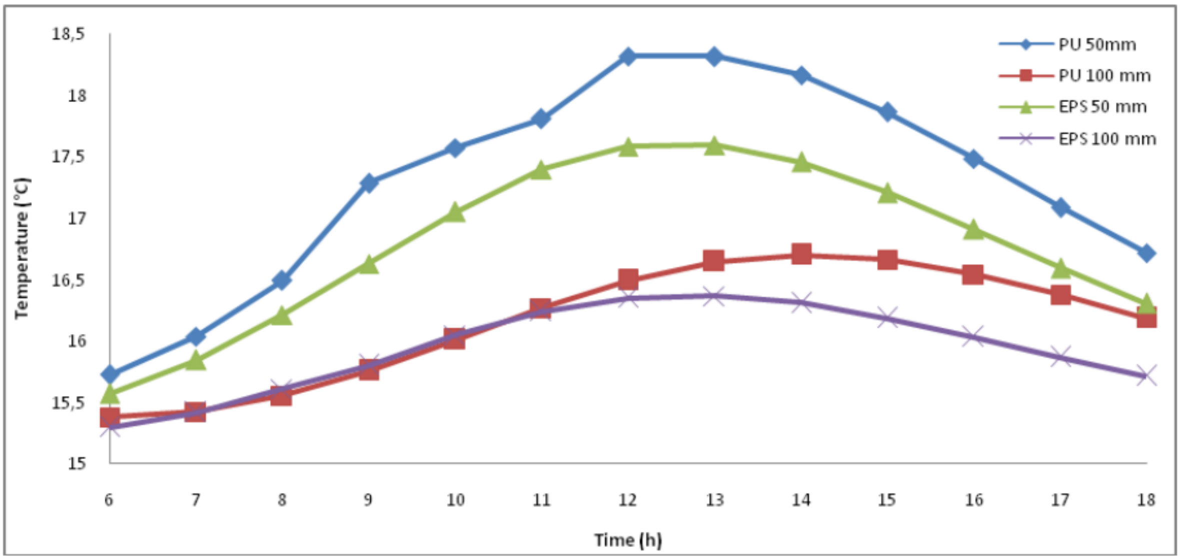

Figure 7. Temperature variation of the internal nodes to PU and EPS plates.

It was observed an increase in the internal temperature to a certain period of time depending on the variation of heat flux and temperature variation. As the radiation focuses on a wall there is an elevation in the external temperature, which is transmitted by conduction to the subsequent nodes until the last node. Using the finite difference method produced variations of temperatures of the inner surface, where it was found that the lowest temperatures were obtained respectively by 100 mm EPS, PU 100 mm, 50 mm and EPS 50 mm PU.

This fact is attributed largely to the properties of the materials are different from one another, providing more or less heat diffusion depending on the molecular structure of materials. It was noted that the thermal conductivity coefficient influence on heat transfer by conduction and the smaller the value, the lower the rate of heat transmitted. The thicknesses of the plates or boards developed important role, where it was noted that the higher the thickness smaller are the temperatures of in the inner surface.

4. CONCLUSION

The panels used in this work and for the proposed situation it was observed that the EPS 100 mm plate presented the best results, the most recommended for cold rooms in these cooling internal temperature ranges. The boards of 50 mm PU presented lower performance, which does not mean that cannot be used for these types of situations. The practice comes showing that PU panels of 50 mm can be used, but that the EPS panels of equal thickness feature better results. Although PU plates and 100 mm EPS exhibit superior performance, your use is not very common on the basis of the acquisition cost of the plates that fold, a fact that does not justify your use for medium temperature applications where temperature variations between boards do not passed of 1.5° C.

5. REFERENCES

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