Experimental and numerical investigation of heat transfer inside an air cavity with a Phase Change Material side

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Abstract

A numerical model was developed to study the coupling of heat transfer in the PCM and the air inside the enclosure. The solidification of the initially melted PCM induces a natural convection of air flow inside the cavity. We present in this work the heat transfer behavior between a chosen phase change material (PCM) incorporated in a thin right wall and a square enclosure filled with air whereas the left one is kept at ambient temperature. The solidification of the PCM generates a latent heat inducing a natural convection air flow inside the cavity and compensating the heat loss. The numerical analysis leaded to a Nusselt number correlations depending on constant Stefan number. Nusselt number is given as a function of a Rayleigh number and dimensionless temperature difference between hot and cold sides. These correlations have been established since there is a lake in the literature to evaluate the natural convection heat transfer coefficient between air and a PCM wall. The parametric domain covered the range of Rayleigh number between $3.4\ 10^5$ to $4.\ 10^7$ with Prandtl number Pr = 0.73 and Stefan number Ste=0.41.

Keywords: Energy Management; Phase Change Material; Natural Convection; Solidification Phase Front; Active Wall; Heat Storage.

NOMENCLATURE

A	Aspect ratio e/L	**Greek symbols**	
C	Specific heat of PCM	ε	Temperature jump
a	Thermal diffusivity		
C_{Pg}	Specific heat of air	ρ	Density
e	PCM thickness	ν	Cinematic viscosity
F	PCM fraction	λ	Thermal conductivity
Fo	Fourier number		
H	Latent heat	Δ	Step
g	Gravity	ψ	Stream function
Nu	Nusselt number	**Subscripts**	
PCM	Phase Change Material	C	Cold
t	Time	H	Hot
T	Temperature	g	Gas
U,V	Velocity components	l	Liquid
x,y	Cartesian coordinates	m	Melting
Ste	Stefan Number $\mathrm{Ste} = \frac{C_{pcm}\ (T_m - T_C)}{H}$		

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1. Introduction

The adaptation of large-scale energy storage solutions are a prerequisite for the development of renewable energy. Thus, to achieve an efficient energy mix, it is necessary to manage the balance between production and demand to ensure optimal service. Except in very specific cases, it is difficult to store electricity directly. It must therefore be transformed into another form of energy more easily stored. One of those ways of storing thermal energy consists in using a phase change material (PCM). The energy is then stored in the form of latent heat through the melting of these materials and then released through the solidification process (Dinçer and Rosen 2002, Zalba et al. 2003). The Thermal Energy Storage (TES) method is commonly used to save energy and to improve the comfort level in buildings. This explains the growing interest of the scientific community to elucidate both numerically and experimentally the thermal performance of a PCM layer incorporated in a part of the building envelope.

The configuration of an enclosure with one side integrating a PCM layer has been rarely addressed despite its wide application such as in optimal utilization of energy (Tan et al. 2002, Ho et al. 2005) and thermal protection (Cao and Faghri 1990). The solidification of the PCM will provide a latent heat inducing a natural convection air flow inside the cavity. In this way, the study of the influence of the PCM wall on the coupled flow and heat transfer in the cavity provides both scientific and practical interests. An issue may arise in the study of this problem. Are the existing correlations applicable to describe the heat transfer between the PCM wall and air in the cavity?

Buoyancy-driven flow in a closed cavity without PCM wall has been studied widely because of the applications in nature and engineering. Many heat transfer correlations have been proposed for imposed temperature or boundary heat flux of the most common cases of rectangular, cylindrical or other regular geometries (Davis1983, Ostrach 1988, Shilei 2014, Gracia 2015). However, there are few studies regarding the solidification/melting effect of a PCM integrated in a boundary on the convective heat fluid inside the cavity. (Zhang and Bejan 1989) reported experimentally and analytically, the melting process of enclosed paraffin heated at a constant rate from the side with an air space 30 mm left at the top of the enclosure. They predicted overall Nusselt number relationship agrees reasonably well with the existing empirical correlation. (Wang et al. 1999) investigated experimentally the melting process of polyethylene glycol in a rectangular enclosure heated from vertical wall. An air gap of 12.7 mm was maintained between the assembled unit and all sides of the container. It was shown that the temporal Nusselt number variation distinguishes three different heat transfer regimes during the melting process. (De Gracia et al. 2013) have studied the convective heat transfer between an air flow and a phase change material plate. They found that the correlation used in the literature to determine the heat transfer coefficient in the case of the PCM is no longer valid. They introduced a correction coefficient in the correlation expression to account for the presence of the PCM.

(Lipnicki and Weigand 2012) studied experimentally and theoretically the natural convection and ice solidification in an annular enclosure. They found that the influence of the contact layer between the frozen layer and the cold surface is of significant importance for the solidification process. To predict the heat transfer coefficient, they used a conventional correlation given by (Vahl Davis and Thomas 1970). A previous work has been conducted to highlight the thermal performance of a Keeping Warm System (KWS) incorporating a PCM wall (Ait lahbib and Chehouani 2015). The studied configuration demonstrated the effect of the PCM discharge where the solidification plays a great role on the heat transfer in the KWS cavity.

As latent thermal energy storage represents smart efficient thermal energy storage, such technique can be used in many domestic and industrial sectors depending on PCM thermophysical properties such as latent heat, melting point and conductivity. During the development of useful applied PCMs, many different groups of phase change materials have been studied, including inorganic compounds (salt and salt hydrates), organic compounds such as paraffin’s, fatty acids and even polymeric materials. In comparison to inorganic PCMs, organic substances could serve as important heat storage media because of their several advantages including their ability to melt congruently, their self nucleation and the fact they don’t have a problem of separation on melting. A relevant aspect is the useful life of this category of PCM and the number of cycles they can withstand without any degrading of their properties. (Hadjieva et al. 1992) used three paraffin mixtures and confirmed that there was no effect of the cycles on the properties of paraffin. (Gibbs and Hsnain 1995) also

verified that neither the cycles nor contact with metals degrade the thermal behavior of paraffin thus they have excellent thermal stability.
The aim of this work is to propose useful heat transfer correlation in a widely used configuration of air cavity incorporating organic paraffin side. In this way both experimental and numerical studies have been carried out for the given configuration. The developed numerical model treats the coupling of the natural convection inside an air square enclosure with only conduction in the PCM side aiming to provide a comprehensive evaluation of the effectiveness of PCMs wall in industrial applications. Nusselt number was calculated as a function of the Rayleigh number, the temperature difference between both sides of the cavity in PCM during the solidification process.

2. Experimental setup and methodology

Experimental setup consists mainly of an enclosure cavity, electrical resistance, insulation material, PCM package, thermocouples and the data acquisition system (Figure 1).

In this study, the enclosure cavity has (10x10x10) cm^3 as geometric dimensions. It was differentially heated across two vertical walls; while the remaining side walls were thermally insulated. The hot wall was heated by an electrical resistance. The cold wall consists of an aluminum plate kept at the temperature T_C by controlled circulating water from a constant temperature bath. All the external surfaces of the test apparatus were insulated with polystyrene insulation material of 4 cm thickness to reduce heat losses from the sides to the surroundings. Four K-type thermocouples with accuracy of 0,2 °C were assembled in the hot and cold walls at various locations along the length, and two others were placed in the center of the gas cavity and the layer of PCM respectively. Temperature was constantly monitored for loading data every 5s via data acquisition system (Fig. 2 and 3).

Fig. 1: Experimental setup. 1) The enclosure cavity. 2) The cold wall. 3) Temperature regulator. 4) Data acquisition, 5) Heating system, 6) Computer.

Experiment have been undertaken to evaluate the numerical model validity. In our disposal, we selected the PCM60 provided by KAPLAN ENERGY-France which is a carboxylic acid that has the properties given in Table 1. We have no data on the thermal conductivity of the solid and liquid states. Their values are measured by the method of transient hot wire. This method is the best known and most widely used for measuring the thermal conductivity of fluid and solid medium because of its rapidity and ease of implementation. The principle of this technique is to heat by Joule effect a filiform element (probe) of very small radius immersed in the fluid to be characterized and measure the temperature rise in the vicinity of the probe versus time via a sensor associated therewith. Indeed, the transfer to the sample center is assumed to be unidirectional.

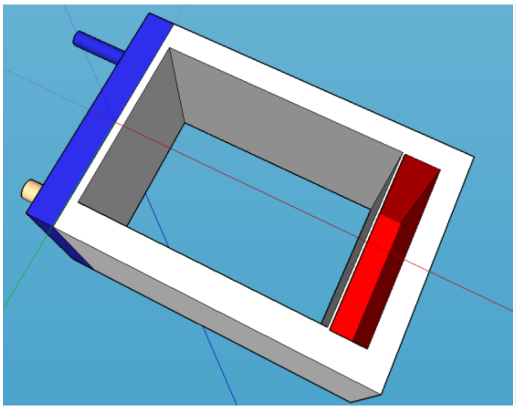

Fig. 2: Inside of the experimental square cavity.

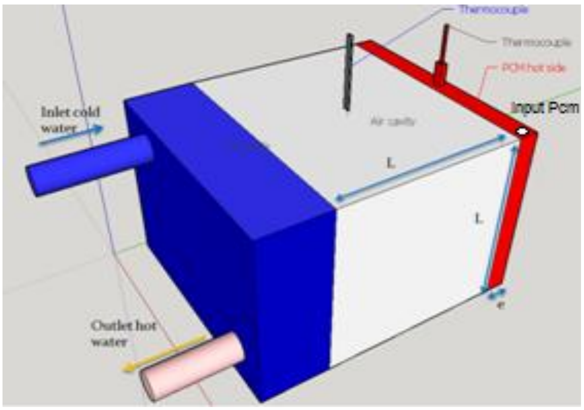

Fig. 3: Experimental square mode.

3. Model formulation and Numerical Solution

The 2D geometry used in the present study is shown in Figure 4. The thickness of PCM is equal to e and the cavity length is L. We define the aspect ratio parameter as A=e/L. The square enclosure is filled by air, cooled from the left side at T_C, insulated from the bottom and the top and heated from the right side by the PCM which is initially at T_H. The PCM is insulated from the top, bottom and right sides.

The following assumptions are made in the present study:

- The natural convection within the melt part of the PCM is negligible and can be ignored.
- All the thermophysical properties of the air are assumed constant except the density giving rise to the buoyancy forces (Boussinesq approximation). These thermophysical properties are taken at the reference temperature T_C.
- The thermo physical properties of the PCM are different for the solid and liquid phases and dependent on phase fraction.
- The PCM behaves ideally and is homogenous and isotropic.

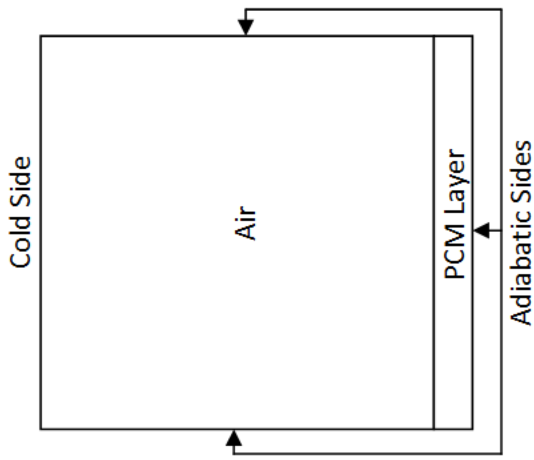

Fig. 4: Physical model

For the air inside the cavity, the equations describing the flow and heat transfer are written as follows:

- Continuity equation

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0 \quad Eq. 1$$

- Momentum equations:

$$\frac{\partial U}{\partial t} + U\frac{\partial U}{\partial x} + V\frac{\partial U}{\partial y} = -\frac{1}{\rho_g}\frac{\partial P}{\partial x} + \nu_g\left(\frac{\partial^2 U}{\partial x^2} + \frac{\partial^2 U}{\partial y^2}\right) \quad \text{Eq. 2}$$

$$\frac{\partial V}{\partial t} + U\frac{\partial V}{\partial x} + V\frac{\partial V}{\partial y} = -\frac{1}{\rho_g}\frac{\partial P}{\partial y} + \nu_g\left(\frac{\partial^2 V}{\partial x^2} + \frac{\partial^2 V}{\partial y^2}\right) - \left[1 - \beta\left(T_g - T_C\right)\right]g \quad Eq. 3$$

- Energy equation

$$\rho_g C_{pg}\frac{\partial T_g}{\partial t} + \rho_g C_{pg}\left(U\frac{\partial T_g}{\partial x} + V\frac{\partial T_g}{\partial y}\right) = \lambda_g\left(\frac{\partial^2 T_g}{\partial x^2} + \frac{\partial^2 T_g}{\partial y^2}\right) \quad Eq. 4$$

Using an enthalpy-porosity method with the previous assumptions, the energy equation in the PCM wall reads as:

$$\rho_{pcm} C_{pcm}\frac{\partial T_{pcm}}{\partial t} = \lambda_{pcm}\left(\frac{\partial^2 T_{pcm}}{\partial x^2} + \frac{\partial^2 T_{pcm}}{\partial y^2}\right) - \rho_{pcm} H\frac{\partial f}{\partial t} \quad Eq. 5$$

The properties of the PCM are updated following the values taken by the liquid fraction after each time step. Thus:

$$\rho_{pcm} = f\rho_l + (1-f)\rho_s \quad Eq. 6$$

$$\lambda_{pcm} = f\lambda_l + (1-f)\lambda_s \quad Eq. 7$$

For the density in the PCM layer was modeled by (Labihi et al. 2017). These authors proposed a new formula, ρ_{eff}, takes into account the PCM and the air layer resulting from volume contraction.

$$\rho_{eff} = \frac{\rho_{pcm} L^* + \rho_g \delta}{L} \quad Eq. 8$$

where ρ_{pcm} is calculated using the Eq. 6

The governing equations are completed by the following initial and boundary conditions:

- $t = 0$, all the intial values (U,V,T) in the gas are in steady state natural convection inside the closed cavity with differentially heated vertical walls. $T_{pcm} = T_H$ and $f = 1$ (the PCM is liquid).
- $x = 0, T_g = T_C, U = V = 0.$
- $x = L + e, \frac{\partial T_{pcm}}{\partial x} = 0, \frac{\partial f}{\partial x} = 0, U = V = 0.$
- $y = 0$ or $y = L, \frac{\partial T_g}{\partial y} = 0, \frac{\partial f}{\partial y} = 0, U = V = 0.$
- $x = L, T_g = T_{pcm}, U = V = 0, \lambda_g \frac{\partial T_g}{\partial x} = \lambda_{pcm}\frac{\partial T_{pcm}}{\partial x},$
- $f = \frac{T - T_s}{T_l - T_s}$ with the correction $f \begin{cases} 0 \text{ if } f < 0 \\ 1 \text{ if } f > 1 \end{cases}.$

The time discretization is performed using the simple first-order Euler implicit scheme. The sets of algebraic equations resulting from the numerical scheme were solved iteratively using the Gauss-Seidel method.

- ***Mesh grid analyses***

In order to ensure grid independent solutions, a series of trial calculations were conducted for different grid distributions with an uniform mesh cells in both PCM and Air cavities.

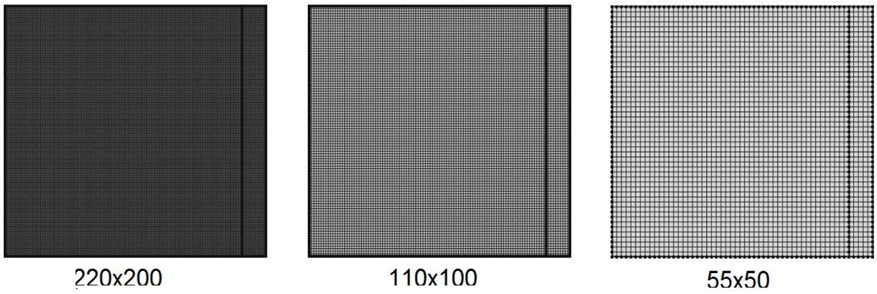

Fig. 5: Different Mesh grids sizes

To evaluate the grid size effect on numerical results, we conducted a series of trial calculations with 50X55, 100X110 and 200X220 nodes (Fig. 5) for a cavity with L=10cm and e=1cm. we calculated for each case the average Nusselt number at hot and cold side and the relative error issues between the different grid size chosen. Table 1 shows the thermophysical properties of RT55 product by RubiTherm used in this study.

Tab. 1 Thermo-physical propreites for RT55

$\boldsymbol{T_m}$ (**K**)	324.15 - 329.15
$\boldsymbol{\rho_l}$ ($\mathbf{kg\ m^{-3}}$)	770
$\boldsymbol{\rho_s}$ ($\mathbf{kg\ m^{-3}}$)	880
$\boldsymbol{\lambda}$ ($\mathbf{Wm^{-1}K^{-1}}$)	0.2
H ($\mathbf{kJ\ kg^{-1}}$)	170
C ($\mathbf{J\ kg^{-1}K^{-1}}$)	2000

Table2 presents the different results founded of this comparison. We can note that the solution is very sensible when we use 50X55 mesh grids cells (up to 6%of difference) contrariwise, the obtained by 100X110 and 200X220 are practically the same.

The mesh 100X110 was adopted on the numerical computations in this study.

Tab. 2: Grid evaluation results

	$\overline{Nu}$ **Hot Side**	$\overline{Nu}$ **Cold Side**			**Hot Side**	**Cold Side**
55x50	13.15	13.135	Relative Error (%)	55x50 Vs 110x100	5.17	5.16
110x100	13.82	13.81		55x50 Vs 220x200	6.03	6.02
220x200	13.94	13.928		110x100 Vs 220x200	0.81	0.81

- ***Validation of the initial state***

In order to validate the initial flow and heat transfer in the cavity with PCM melted at constant T_H, we used two correlations for the average Nusselt number defined as:

$$\mathrm{Nu} = \frac{\mathrm{hL}}{\lambda_{\mathrm{g}}} \quad \mathrm{Eq.\,9}$$

Where h is the average convective heat coefficient through the hot side of the cavity.

Table 3 compares the obtained Nusselt number with those calculated using the empirical correlations established respectively by (Eckert and Carlson 1962) and (Lankhorst 1991) for $\mathrm{Ra} = 5.32\ 10^6$. Since the relative error is less than 3%, our numerical model adopted in this study could be considered valid.

Tab. 3: Validation of the numerical model.

	$\overline{Nu}$	Relative error
This study	13.82	-
(Eckert and Carlson 1962)	14.139	2.3%
(Lankhorst 1991)	13.517	2.19%

4. Results and discussion

The experimental results were presented in our previous work (Labihi et al 2017). We found that the numerical simulation results match well with the experimental measurements. For that we conclude that the numerical model be considered valid along the process.

Due to the lack of empirical information about the heat transfer coefficients in the same studied configuration, authors used to use differentially heated wall cavity literatures correlation which is not representing the real physical processes of heat exchange between PCM and air cavity. In order to obtain correlations capable of characterizing the transient heat transfer process inside such cavity a simulation runs were carried out to obtain correlations of heat transfer rate via Nusselt number, with Rayleigh number Ra and the differnce dimensionless temperature between the hot and cold sides Θ.

In this study we define Ra number by the following formula $R_a = \frac{\rho g \beta L^3 (T_H - T_c)}{\alpha \mu}$ and $\Theta = \frac{T(t) - T_c}{T_H - T_c}$
Where $T(t)$ is the average temperature at hot side in each time.

In order to get a significatif range of Ra number we have chosen to vary L and not $(T_H - T_c)$ always keeps the same aspect ratio $A = \frac{e}{L} = 0.1$. Heat transfer at the hot interface between air and PCM is thoroughly inspected in the defined range $3.4\ 10^5 < Ra \leq 4.3.10^7$ with $Ste = 0.41$, $T_H = 50$ °C and $T_C = 20$ °C.

A simulation series was carried out with L=6, 10, 14 and 18cm. Figure 6 present the liquid fraction as a function of Θ for different Ra numbers. Despite the fact that we have the same formal report we can see that the solidification process (Θ from 0.6 to 0) is not the same when we change L. whereas the curves are very close to one another for Θ greater than 0.6. That can be explaining by the fact of volume contraction due to the difference between the density liquid/solid.

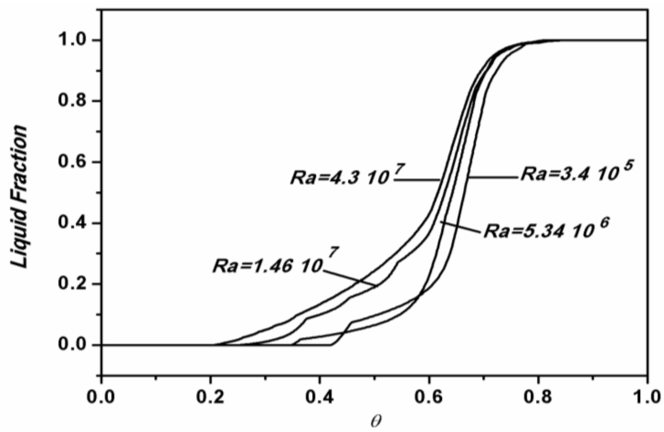

Fig. 6 Liquid fraction as function of Θ

The figure 7 shows the methodology that we use to find the correlations. to find an equation Nu as a function of Θ for different Ra number. We can see that the Nu number have a non-linear curve as function of Θ also we have some perturbation in the curves. The Nu perturbation becomes important when Ra increase.

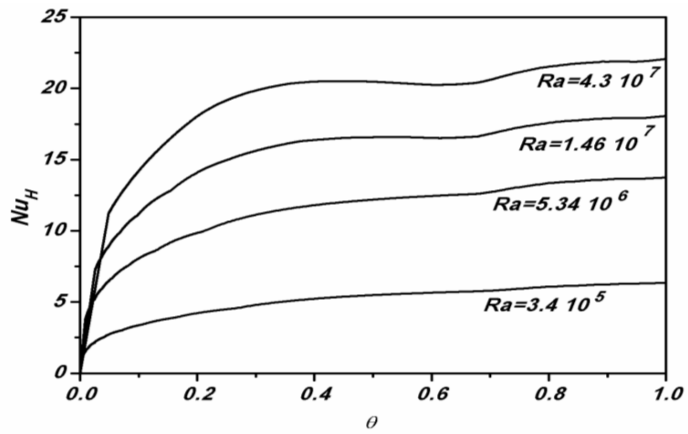

Fig. 7: Nusselt number evolution with dimensionless temperature for Ste=0.41

By using a non linear fitting for the curves issued in Figure 7.A new correlations between the dimensionless Nusselt number as a function of dimensionless Temperature Θ were made and presented in table 4.

Tab. 4: Nusselt number correlations at hot side as function of θ for different Rayleigh numbers

L (cm)	Ra	$\overline{Nu}$
6	$\mathbf{3.4\ 10^{5}}$	$\mathbf{6.7\ \theta^{0.3}}$
10	$\mathbf{5.34\ 10^{6}}$	$\mathbf{14.88\ \theta^{0.283}}$
14	$\mathbf{1.46\ 10^{7}}$	$\mathbf{19.67\ \theta^{0.254}}$
18	$\mathbf{4.3\ 10^{7}}$	$\mathbf{23.50\ \theta^{0.213}}$

5. Conclusion:

It is generally agreed that the intensive investigations undertaken in the last decades have given latent heat significant advantages especially organic PCM. We choose for this study a Rubitherm RT55 paraffin We conduct a numerical simulation where several assumptions had to be made in order to create a correlation model as close as possible to the realized experiment such as the invariability of PCM volume. Also knowledge of all the properties of the paraffin and the experimental validation would obviously provide more accurate results. Nevertheless, the coupling of convection of heat transfer in a cavity with solidifying PCM was successful and proposed correlation can be definitely used as a prediction in industrial or engineering fields. Hence, it can help for material selection step during design providing an economical advantage in saving time and money.

Based on the Numerical results and the correlated equations, the following conclusions could be drawn:

- The solidification process accurse early in time and tend to begin from the bottom to the top by following convection air loop.
- To have more Ra values we heed to vary L not the temperature difference between hot and cold side
- The contraction of volume in PCM layer affect the heat transfer processes from a Ra number to another despite the same aspect ratio in all cases.

For Ste=0.47 we see that we can write a new correlation for Nu number as function of Θ and Ra as the following formula:

$$\boldsymbol{Nu = aRa^{n}\theta^{m}}$$

In future work we will exanimate a range of Ste values in order to write a general correlation of Nu number as a function of Ste, Ra andΘ.

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6. References:

A. Labihi, F. Aitlahbib, H. Chehouani, B. Benhamou, M. Ouikhalfan, C. Croitoru, I. Nastase, , 2017, Effect of phase change material wall on natural convection heat transfer inside an air filled enclosure, In Applied Thermal Engineering, Volume 126, Pages 305-314.

AM. Lankhorst, 1991, Laminar and turbulent natural convection in cavities: Numerical and experimental validation, PhD thesis University Delft.

Alvaro, G, Damien, D, Castell,A,and, Luisa F. C, 2013, transfer coefficient between an air flow and a phase change material Joseph Virgone, A correlation of the convective heat plate, J. Applied Thermal Engineering, Volume 51, Issues 1–2, Pages 1245-1254.

B.M. Gibbs, S.M. Hasnain, ,1995, DSC study of technical grade phase change heat storage materials for solar heating applications, Proceedings of the 1995 ASME/JSME/JSEJ International Solar Energy.

Cao,Y, and Faghri, A., 1990, Thermal protection from intense localized moving heat fluxes using phase-change material, Int. J. Heat Mass Transfer 33 .1271–1238.

Dinçer, I, Rosen M.A., 2002, Thermal Energy Storage, John Wiley & Sons, New York.

De. G,and Davis,V, 1983, Natural convection of air in a square cavity: a bench mark numerical solution, Int. J. Numer. Methods Fluids, 249-264.

E.R. Eckert, W.O. Carlson, 1961, Natural convection in a layer enclosed between two plates with different temperatures. Int. J. Heat Mass Transfer,106-120.

F. Aitlahbib, H. Chehouani, 2015, Numerical study of heat transfer inside a Keeping Warm System (KWS) incorporating phase change material. Appl Therm Eng; 75:73-85.

Gracia,A., Cèsar Fernández, Albert Castell, CarlesMateu, Luisa F. Cabeza, 2015, Control of a PCM ventilated facade using reinforcement learning techniques, Energy and Buildings, Volume 106, 1, 234-242.

Ho, C.J, Chiu, S.Y, and Lin J.F., 2005, Heat transfer characteristics of a rectangular natural circulation loop containing solid–liquid phase-change material suspensions, Int. J.Numer. Methods Heat Fluid Flow 15,441–461.

M. Hadjieva, S. Kanev, J. Argirov, 1992, Thermophysical properties of some paraffins applicable to thermal energy storage, Solar Energy Mater. Solar Cells 27, 181–187.

Ostrach,S., 1988, Natural convection in enclosure, J. Heat Transfer 110, 1175e1190.

R. W. Thomas, G. de Vahl Davis, 1970, Natural convection annular and rectangular cavities – a numerical study, in: Proc. 4th Int. Heat Transfer Conference, Paris 4, Paper N.C. 2.4.

Shilei, L,Shangbao, L, Jingyu, H, and Xiangfei, K,2014, Establishment and experimental verification of PCM room's TRNSYS heat transfer model based on latent heat utilization ratio,J. Energy and Buildings, Volume 84, Pages 287-298.

Tan, G.H, and Ho C.J, 2002, Experiments on thermal behaviors of a natural circulation loop with latent heat storage under cyclic pulsating heat load, Heat Mass Transfer 39,11–17.

Wang,Y., Amiri,A, and Vafai,K.,1999, An experimental investigation of the melting process in a rectangular enclosure,J. International Journal of Heat and Mass Transfer 42, 3659-3672.

Z. Lipnicki, B. Weigand, 2012, An experimental and theoretical study of solidification in a free-convection flow inside a vertical annular enclosure, Int. J. Heat Mass Transfer, 55, pp. 655–664.

Zhang, Z., and, Bejan A., 1989, Melting in an enclosure heated at constant rate, J.International of Heat and Mass Transfer 32, 1063e1076.

Zalba, B, Marin J.M. Cabeza L.F., Mehling H., 2003, Review on thermal energy storage with phase change: materials, heat transfer analysis and applications, J.Appl.Therm. Eng. 23,251–283.