

Design of air cooling unit with PCM

Ç Susantez¹, E Akyol¹, O Hacıhafızoğlu¹ and U Akyol²

¹ Mechanical Engineering Department, Engineering Faculty, Trakya University, Edirne, Turkey

² Mechanical Engineering Department, “Tekirdag Namik Kemal University”, Çorlu, Tekirdag, Turkey

E-mail: cigdemsusantez@trakya.edu.tr

Abstract. In this paper a heat absorption unit has been designed to cool the entering hot air. For this aim, five slabs of PCM (Phase change material) have been used inside the cooling unit. The velocity and temperature values of entering hot air respectively are 0.1 m/s and 50°C. Three different cases for PCM design were investigated: in the first case RT 44HC, RT 35HC, RT 31, RT 28HC and RT 25HC as phase change materials were used in such a way that the PCM with higher number gets contact with the hot air first. In the second and third cases only one PCM type were used for the five slabs inside the cooling unit. They are RT 44HC and RT 25HC. Analysis performed for 10h. Results show that although maximum heat absorbed for the first 8h is in the case of only RT 25HC used in the cooling system, after 8h the cooling unit with mixed PCM shows a better performance and minimum exit air temperature has been observed for this case at the end of 10h.

1. Introduction

Phase change materials have been used in many applications such as latent thermal energy storage systems, building and automotive applications, thermal protection of electronic devices and foods, heat and cold therapy applications and logistics of pharmaceuticals [1-4]. It is also possible to see PCMs in many heat exchanger configurations. It is important to select the true PCM for the heat exchanger. In this respect the properties of latent heat, phase change temperature range and thermal conductivity of PCM play an important role [5].

There are many studies on different kinds of heat exchangers taking the advantage of PCM. Raj and Velraj [6] made an experimental and numerical study on their developed modular shell and tube type heat exchanger. They concluded that air spacers between the modules increase the heat transfer by increasing the retention time of air up to the frontal velocity of 2 m/s. Sun et al. [7] experimentally investigated the thermal behaviour of PCMs in rectangular slabs inside the wind tunnel. They showed that increasing the inclination angle of slab, air inlet temperature and velocity have positive effect on the energy charging speed. Saeed et al [8] performed an experimental work on plate type heat exchanger working with water and PCM. They obtained the result that the optimum plate-plate spacing is 1 in for reducing the PCM shelf-shielding and having a lower exit water temperature. Sharma et al. [9] made a numerical study on melting fraction of fatty acids as PCMs. They showed that capric acid is the best among investigated PCMs for latent heat storage system. Lopez et al. [10] numerically modelled heat exchanger with PCM slabs and exposing to air flow for building applications. They also validated their results with experiments. Hed and Bellander [11] presented a



mathematical model for PCM air heat exchanger by taking into account the specific heat capacity of PCM as a function of temperature. They also verified their model with experiments. Dolado et al. [12] made an experimental and numerical study on PCM air heat exchanger. In their model they not only took into account temperature dependent enthalpy and thermal conductivity, but also convection inside the PCM by using effective conductivity. They concluded that an enhancement of the PCM thermal conductivity causes insignificant improvement on the thermal performance of thermal energy storage unit and 1 °C mismatching in the temperature of PCM enthalpy-temperature curve can cause relative errors up to 20 % in the power curve of thermal energy storage unit.

In this study a new design for the PCM air heat exchanger has been proposed. In the cooling unit five PCMs were used to cool the entering hot air. Three different cases were investigated. While in the first case five different types of PCMs were used, in other cases only RT 44HC and RT 25HC were used respectively.

2. Material and Method

The design of the investigated air cooling unit has been presented in figure 1 for three different investigated cases. The inlet air temperature is 50 °C. As it can be shown in figure 1, the PCM which has the highest melting temperature gets contact with the hot air first in the case of a, while in other cases only one type of PCM is used.

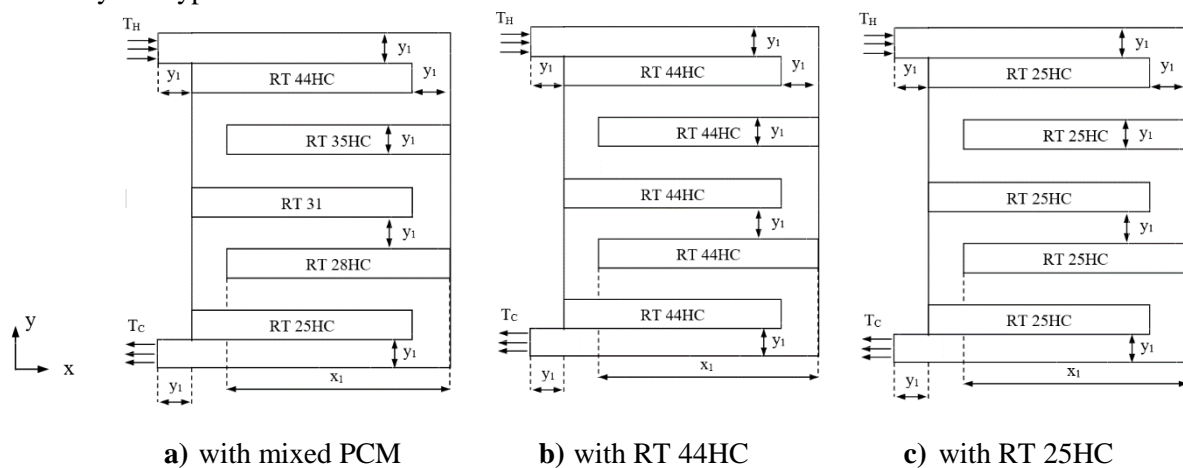


Figure 1. Cooling unit configurations.

Geometric dimensions of the cooling unit, thermophysical properties of air and PCMs are given in Table 1 and 2 respectively. Temperature dependent expressions of density, thermal conductivity and dynamic viscosity values of air have been obtained in the range of 20°C-50°C by curve fitting in Sigma Plot.

Table 1. Geometric dimensions and thermophysical properties.

x_1 (m)	0.11		
y_1 (m)	0.01		
c_p (J/kgK)	1007	[13]	
ρ (kg/m ³)	$1.2772-3.7 \cdot 10^{-3} T$ (°C)	[13]	$R^2 = 0.9991$
μ (kg/m s)	$1.7337 \cdot 10^{-5} + 4.6 \cdot 10^{-8} T$ (°C)	[13]	$R^2 = 0.9999$
k (W/m K)	$2.37 \cdot 10^{-2} + 7.3786 \cdot 10^{-5} T$ (°C)	[13]	$R^2 = 1$

The governing equations are given below.

Table 2. Thermophysical properties of PCMs.

PCM type	ρ_{PCM} (kg/m ³)	c_{PCM} (kJ/kgK)	k_{PCM} (W/mK)	L (kJ/kg)	T_s (°C)	T_l (°C)
RT 44HC [14]	750	2	0.2	220	41	44
RT 35HC [14]	825	2	0.2	210	34	36
RT 31 [14]	820	2	0.2	155	27	33
RT 28HC [14]	825	2	0.2	220	27	29
RT 25HC [14]	825	2	0.2	180	22	26

Energy equation for PCM:

$$\rho_{PCM} c_{PCM} \frac{\partial T}{\partial t} = k_{PCM} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) + S_H \quad (1)$$

$$S_H = -\rho_{PCM} L \frac{\partial \beta}{\partial T} \frac{\partial T}{\partial t} \quad (2)$$

$$\beta = \begin{cases} 0 & \text{if } T \leq T_s \\ \frac{T-T_s}{T_l-T_s} & \text{if } T_s < T < T_l \\ 1 & \text{if } T \geq T_l \end{cases} \quad (3)$$

where S_H is the source term, L is the latent heat, β is the melt fraction, T_s and T_l are solidified and liquefied temperatures respectively.

Continuity and momentum equations for air:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} = 0 \quad (4)$$

$$\rho \left(\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial P}{\partial x} + \frac{\partial}{\partial x} \left(\mu \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial u}{\partial y} \right) - \frac{2}{3} \frac{\partial}{\partial x} \left(\mu \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right) \right) \quad (5)$$

$$\rho \left(\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial P}{\partial y} + \frac{\partial}{\partial x} \left(\mu \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\mu \frac{\partial v}{\partial y} \right) - \frac{2}{3} \frac{\partial}{\partial y} \left(\mu \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right) \right) \quad (6)$$

Energy equation for air:

$$\rho c_p \left(\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = \frac{\partial}{\partial x} \left(k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(k \frac{\partial T}{\partial y} \right) \quad (7)$$

Initial and boundary conditions are given bellow.

$$\left. \begin{array}{l} T = T_H = 50^\circ C \\ V = V_{in} = 0.1 m/s \end{array} \right\} \text{ for the inlet of the cooling unit} \quad (8)$$

$$\left. \begin{array}{l} P = P_o = 0 \\ \frac{\partial T}{\partial x} = 0 \end{array} \right\} \text{ for the outlet of the cooling unit} \quad (9)$$

$$\left. \begin{array}{l} u = 0 \\ v = 0 \end{array} \right\} \text{ on walls} \quad (10)$$

$$\frac{\partial T}{\partial \vec{n}} = 0 \text{ on the outer surfaces of the cooling unit} \quad (11)$$

$$\left. \begin{array}{l} \vec{n} \cdot (\vec{q}_1 - \vec{q}_2) = 0 \\ T_1 = T_2 \end{array} \right\} \text{ on contact surfaces of PCM and air} \quad (12)$$

where \vec{n} is unit normal vector, 1 and 2 represent air and PCM respectively.

$$\left. \begin{array}{l} u = 0 \\ v = 0 \\ P = 0 \\ T = T_0 = 20^\circ C \end{array} \right\} \text{ if } t=0 \quad (13)$$

In this study, as the thickness of PCM slabs are thin, the effect of natural convection in the liquid phase in PCM is neglected. The average temperature of the cooled outlet air has been calculated and

presented for all the investigated configurations of the cooling unit. Analysis has been performed in Comsol Multiphysics finite element and simulation software for 10h.

$$T_C = \frac{1}{A_{out}} \int_{A_{out}} T dA \quad (14)$$

Absorbed heat from the hot air during the process has been calculated according to equation (15).

$$\dot{Q}_{abs} = \dot{m}_{air} c_p (T_H - T_{out,ave}) = \bar{\rho} V_{in} A_c c_p (T_H - T_C) \quad (15)$$

where \dot{m}_{air} is the mass flow rate of air, $\bar{\rho}$ is the density of air at 35 °C ($\bar{\rho} = 1.145 \text{ kg/m}^3$) and A_c is the cross section area perpendicular to the air flow. Although we took temperature dependent air density expression in the analysis as it effects the exit air temperature, it was an obligation to use a constant density value for the air in the calculation of absorbed heat. The value of A_c is 0.001 m^2 as the depth perpendicular to the x-y plane was taken as 0.1 m.

3. Results and Discussion

Free triangular mesh has been used in the analysis as shown in figure 2. Six different mesh types with different number of elements were investigated in mesh dependency analysis as it can be seen on Table 3. Mesh type with number 6 has been used in the analysis. Validation of this code has also been shown in our previous study [15].

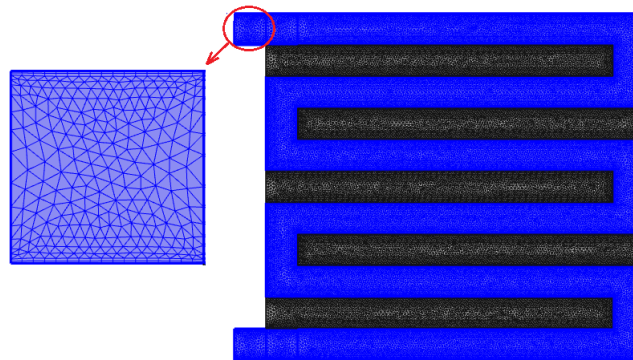


Figure 2. Geometry with mesh.

Table 3. Mesh dependency study.

Mesh type no	Number of Elements	$T_{out,ave}$ (°C) at $t = 10 \text{ h}$
1	3253	28.4629
2	5005	28.8838
3	17121	29.2522
4	51355	29.5612
5	57371	29.6260
6	78682	29.9473

Simulation results for temperature distribution have been presented for cooling unit designs with mixed PCM, RT 44HC and RT 25HC in figure 3, 4 and 5 respectively. In the case of mixed PCM, the PCM plate, which has the highest melting temperature, gets in contact with the entering hot air first. It has been shown that RT 44HC, RT 35HC, RT 31, RT 28HC and RT 25HC start to phase change after 1 hour and they are continuing to phase change after 5 h. RT 35HC and RT 31 complete phase changing process while others are still continuing absorbing heat by phase changing at the end of the investigated time of 10h.

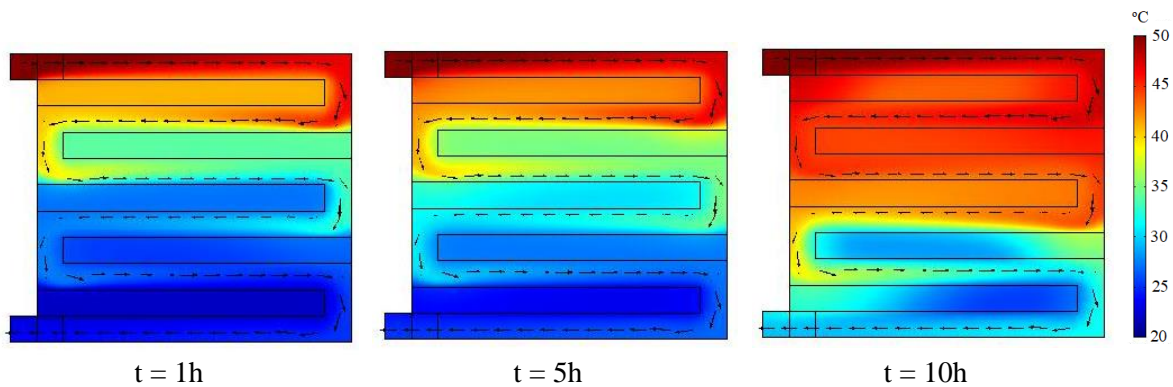


Figure 3. Temperature distribution in the cooling unit with mixed PCM.

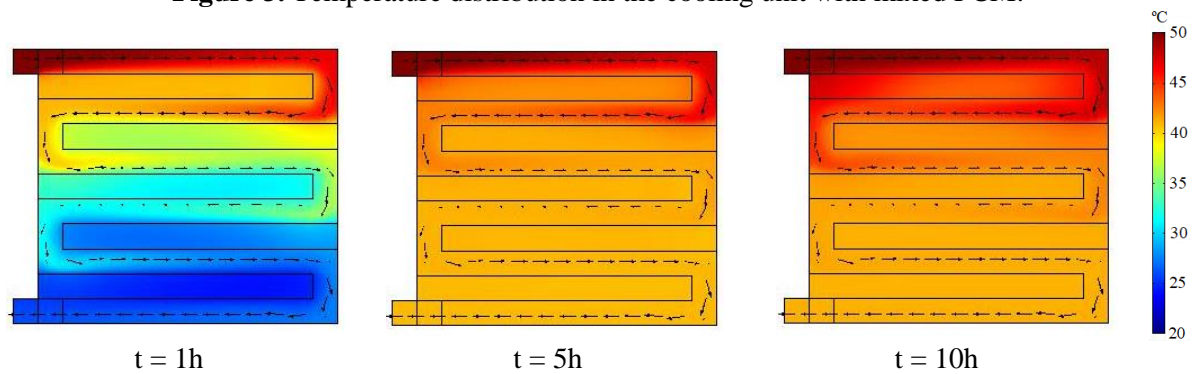


Figure 4. Temperature distribution in the cooling unit with RT 44HC.

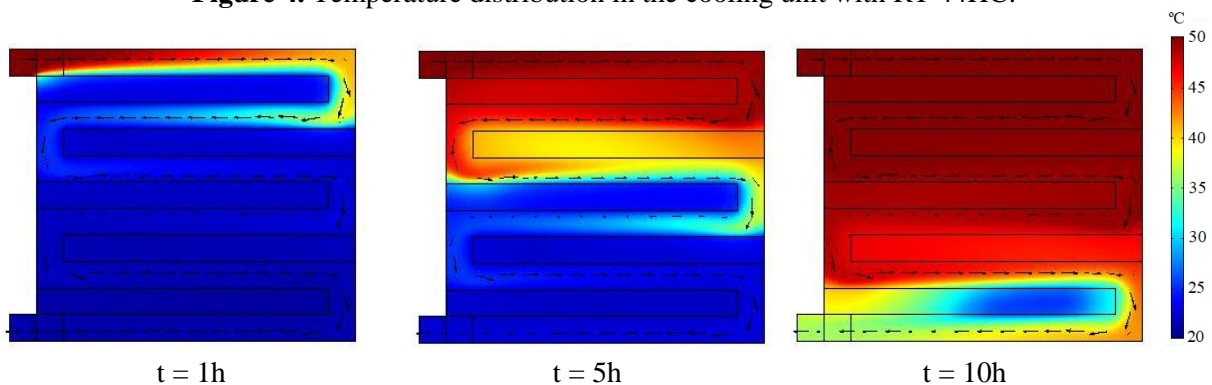


Figure 5. Temperature distribution in the cooling unit with RT 25HC.

In the case of only RT 44HC is used, only the first PCM which is close to the entrance of hot air starts to phase change after 1 hour. Only three PCM slabs close to the entrance of hot air are in the process of phase change, while others are at the temperature below solidified temperature after 5h. The first PCM which is close to the entrance of hot air almost completes phase change process and others are in the process of phase change at the end of 10h.

In the case of only RT 25HC is used, only three PCM slabs close to the entrance of hot air are in the process of phase change and others are at temperature below solidified temperature at the end of 1h. Only three PCM slabs, which are close to the exit, absorb heat by phase changing and other two PCM slabs are highly above the liquefied temperature at the end of 5h. The PCM slab close to the exit is about to complete phase change while others are highly above the liquefied temperature after 10h.

On the other hand, the average temperature of the air at the exit has presented for all the investigated configurations of the cooling unit as seen in figure 6. It has been observed that until 8h the values of exit air temperature for the cases of mixed PCM and RT 25HC are very close to each other and the minimum exit air temperature is seen for the case of mixed PCM at the end of 10h. The exit air temperature increases rapidly until solidified temperature and then stays constant around this temperature for the case of RT 44HC. In that case average value of the exit air temperature of the air is relatively higher than other cases. Because latent heat has more important effect on the cooling process than sensible heat, solidified temperature of RT 44HC is higher than that of other investigated PCMs and at the beginning of the process in the case of RT 44HC heat is absorbed from hot air in the form of sensible heat rather than latent heat. Increase on the average value of the exit air temperature has been observed after the process of phase change in the PCM slabs complete. Figure 7 shows the absorbed heat from the hot air as a function of time for all the investigated cooling unit designs. When solidified temperatures of PCMs have lower values, phase change process starts more quickly as it can be seen for the cases of mixed PCM and RT 25HC and during the phase change more heat is absorbed from the air. On the other hand, according to equation (15), absorbed heat is directly proportional to $(T_H - T_C)$. When phase change continues inside the PCM slabs in the case of RT 44HC, the exit air temperature is higher and as a result absorbed heat is lower.

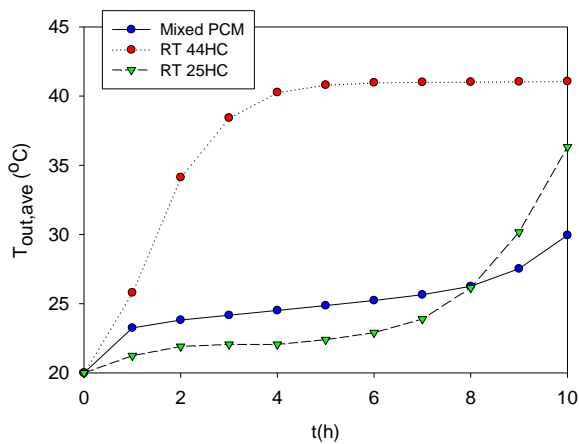


Figure 6. Average temperature of the exit air

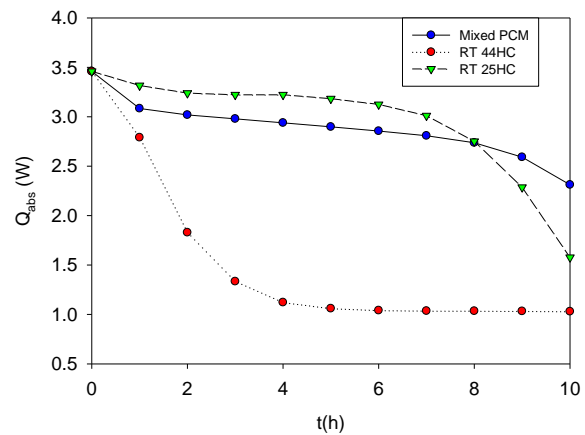


Figure 7. Absorbed heat

4. Conclusion

In this research, three different configurations of cooling unit design with PCMs have been investigated. Cooling units with mixed PCM, RT 44HC and RT 25HC were investigated. In the case of mixed PCM, increasing the absorbed heat was aimed by satisfying the PCM with the highest solidified temperature to get contact first with the hot air. It has been observed that during first 8h cooling unit performances with mixed PCM and RT 25HC are very close to each other and they are better than the case of only RT 44HC is used. The cooling unit design with mixed PCM shows a better performance by absorbing more heat after 8h.

Nomenclature

A : Area (m^2)

A_c : Cross section area perpendicular to the air flow (m^2)

A_{out} : Cross section area perpendicular to the air flow at the outlet (m^2)

c_p : Specific heat of air (J/kgK)

c_{PCM} : Specific heat of phase change material ($J/kg \cdot K$)

k : Thermal conductivity of air (W/m K)
 k_{PCM} : Thermal conductivity of phase change material (W/m K)
 L : Latent heat (J/kg)
 \dot{m}_{air} : Mass flow rate of air (kg/s)
 \vec{n} : Unit normal vector
 P : Pressure (Pa)
 P_o : Pressure at the outlet (Pa)
 \dot{Q}_{abs} : Absorbed heat (W)
 \vec{q} : Heat flux vector (W/m²)
 S_H : Source term (W/m³)
 T : Temperature (°C)
 T_0 : Initial temperature (°C)
 T_C : Cooled air temperature (°C)
 T_H : Hot air temperature (°C)
 T_S : Solidified temperatures (°C)
 T_L : Liquefied temperatures (°C)
 t : time (s, h)
 u : Velocity in x direction (m/s)
 V_{in} : Inlet velocity (m/s)
 v : Velocity in y direction (m/s)
 x, y : Coordinates
 x_1 : Length of the PCM plates (m)
 y_1 : Width of the PCM plates (m)

Subscripts

1, 2: Represent air and PCM respectively

Greek symbols

β : Melt fraction
 μ : Dynamic viscosity of air (kg/m s)
 ρ : Density of air (kg/m³)
 $\bar{\rho}$: Density of air at the mean temperature (kg/m³)
 ρ_{PCM} : Density of phase change material (kg/m³)

References

- [1] Ezan M A, Yüksel C, Alptekin E and Yılancı A 2018 Importance of natural convection on numerical modelling of the building integrated PVP/PCM systems *Solar Energy*. **156** 616
- [2] Seddegh S, Joybari M M, Wang X and Haghghat F 2017 Experimental and numerical characterization of natural convection in a vertical Shell-and-tube latent thermal energy storage system, *Sustainable Cities and Society*. **35** 13
- [3] <https://www.rubitherm.eu/en/applications.html>
- [4] Zalba B, Marin J M, Cabeza L F and Mehling H 2003 Review on thermal energy storage with phase change: materials, heat transfer analysis and applications *Applied Thermal Engineering*. **23** 251
- [5] Lazaro A, Dolado P, Marin J M and Zalba B 2009 PCM-air heat exchanger for free-cooling applications in buildings: Empirical model and application to design, *Energy Conversion and Management*. **50** 444

- [6] Raj V A A and Velraj R 2011 Heat transfer and pressure drop studies on a PCM-heat exchanger module for free cooling applications, *International Journal of Thermal Sciences*. **50** 1573Sun X, Chu Y, Medina M A, Mo Y, Fan S and Liao S 2019 Experimental investigations on thermal behaviour of phase change material (PCM) in ventilated slabs, *Applied Thermal Engineering*. **148** 1359
- [7] Saeed R M, Schlegel J P, Sawafta R and Kalra V 2019 Plate type heat exchanger for thermal energy storage and local shifting using phase change material, *Energy Conversion and Management*. **181** 120
- [8] Sharma A, Won L D, Buddhi D and Park J U 2005 Numerical heat transfer studies of the fatty acids for different heat exchanger materials on the performance of a latent heat storage system, *Renewable Energy*. **30** 2179
- [9] Lopez J P A, Kuznik F, Baillis D and Virgone J 2013 Numerical modeling and experimental validation of a PCM to air heat exchanger, *Energy and Buildings*. **64** 415
- [10] Hed G and Bellander R, Mathematical modelling of PCM air heat exchanger 2006, *Energy and Buildings*. **38** 82
- [11] Dolado P, Lazaro A, Marin J M and Zalba B 2011 Characterization of melting and solidification in a real scale PCM-air heat exchanger: Numerical model and experimental validation, *Energy Conversion and Management*. **52** 1890
- [12] Çengel Y A and Cimbala J M 2008 *Akışkanlar Mekaniği Temelleri ve Uygulamaları* ed Engin T (İzmir: İzmir Güven Kitabevi)
- [13] <https://www.rubitherm.eu/en/index.php/productcategory/organische-pcm-rt>
- [14] Susantez Ç, Kahveci K and Akyol E, Effects of the number, thickness and non-uniform distribution of fins on the heat storage characteristics in a cylindrical thermal energy storage with phase change material 2018, *Fırat üniversitesi Mühendislik Bilimleri Dergisi*, **30** 225