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Thermal Change for Heat Exchanger Cooling System of PCM

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Abstract

A current study is presented in which a real-size room at daytime is cooled by a heat charge and discharge in a latent-heat-storage unit of phase changed materials. The unit is designed as a shell-and-tube heat exchanger. The tubes are vertical and filled with a phase-change material. The PCM melts and the room air is cooled down to a comfortable level, which lasts as long as the PCM is melting. While the ambient temperature changes from 35◦C to 45◦C, the comfortable time and variations are analytically calculated. Some cases are considered: an insulated room, a room heated by the ambient. By using the averaged values defined for the phase changed materials, the obtained analytical calculations and solutions are carried out with the available data in the literature. It is shown that the present theoretical study results related to the cooling using ambient conditions and the different dimensions will have an effect on the time of cooling. In addition, it can be seen from the results that dimension has a dominant effect on the temperature of stored energy than conditions.

Key words

Cooling Time and Heat Exchanger , Phase Change, Energy Storage

1. INTRODUCTION

Many studies about the energy problems and solutions have been reported. The problems can be solved by using thermal energy storage. Several theoretical and experimental investigations were devoted to modelling the thermal performance of the storage units and phase changed materials (PCM) [1-6], the different phase change materials, investigation of new geometries, and new concepts/ for energy storage technology. In these studies, the effects of various working parameters, the storage time, the efficiency of system and phase changed materials were investigated.

Thermal storage units are used for domestic, waste, vehicle heating or cooling in systems characterized by different temperatures in the daytime and cool nights. During the day the PCM absorbs heat in it and melts, and then it releases its latent heat to the ambient while at night/different times. So, a PCM-based unit in air conditioner decreases considerable energy consumption and solves an energy problem in its operation.

Units of similar and different structure have been investigated in the literature. Farid and Kanzawa [7] theoretically investigated a shell-and-tube heat exchanger unit based on thermal storage. The air of system was flowing in the shell across the tubes filled with PCM. The performance of the unit improves by using different PCMs in the same unit. However, Lacroix [8] investigated numerically and experimentally the PCM stored in the shell and the heat-transfer fluid circulated inside the vertical tubes in a shell-and-tube unit. Turnpenny et al. [9,10] studied a unit utilized as a ventilation cooling system in buildings and used a heat pipe embedded in a phase-change material. Mozhevelov et al. [11] studied thin vertical storage units installed parallel to walls in a room. In daytime heat was free-convected from the room air, while at night heat was released from the unit into the ambient by both free and forced convection. Mozhevelov [12] also investigated simulations of a portable storage unit with cooling elements of various shapes in a shell: vertical plates, horizontal plates, horizontal square tubes in in-line and staggered configurations, and vertical square tubes in an in-line configuration. Arye and Guedj [13] experimented with a shell-and-tube unit, in which the tubes were vertical and filled with paraffin wax. The room air convected by fans through the shell. While the ambient temperature at daytime was 30 ◦C to 35◦C and at night, 18 ◦C to 19◦C. Mosaffaa et al [14] presented a comparative study for solidification of the PCM in cylindrical shell and rectangular storages having the same volume and heat transfer surface area. Vakilaltojjar and Saman [15] investigated the effect of slab thickness on a rectangular storage unit performance by using a semi-analytical method for phase change for air conditioning applications. Akgun et al. [16] experimentally carried out PCM melting and solidification in a shell and tube heat exchanger. Hosseini et al [17] studied experimentally and numerically thermal behavior and heat transfer characteristics of Paraffin RT50 as a phase change material (PCM) during constrained melting and solidification processes inside a shell and tube heat exchanger. Hosseini et al [18] investigated the effect of inlet temperature of the heat transfer fluid on melting process in a shell and tube heat exchanger numerically and experimentally. Agyenim et al. [19-22] investigated melting and solidification on a paraffin in a shell and tube heat exchanger for various operating conditions and geometric parameters. Erek et al [23] carried out experimental and numerical investigation of thermal energy storage with a finned tube. In this type of heat exchanger, PCM fills the annular shell space around the finned tube while the heat transfer fluid flows within the tube.

Due to the relatively low thermal conductivity of PCMs, many studies have been performed to improve the heat transfer in a storage unit. Zhang and Faghri [24] investigated the heat transfer enhancement in a latent storage system using a finned tube. Sciacovelli et al [25] analysed the melting process in a single vertical shell-and-tube latent heat thermal energy storage, unit and directed at understanding the effect of nanoparticle enhancement of the system.

It can be seen from the studies above, to understand the effect of ambient and the problems faced with sizing the thermal energy storage system, a simplified heat exchanger can be a good contribution and practical both in design and also in operation. Therefore, a shell-and-tube heat exchanger based on the PCMs is presented in this paper. This uses basic mathematical calculations to estimate total time of charging and discharging processes under constant or mean entrance conditions. Furthermore, the calculations are carried out for a latent-heat-storage unit utilized for temperature moderation of an enclosed space. In the examined case, a real-size room is cooled down to a comfortable level, which continues as long as the PCM is melting. The design of the unit and its mode of operation temperature determine the thermal change of PCM and the rate of air cooling in the room.

2. HEAT EXCHANGER SYSTEM DESIGN

In this study, a shell and tube heat exchanger unit selected in order to investigate the change of storage time for a latent heat storage system. Fig. 1 illustrates a picture and a schematic diagram of the unit, which consist of airflow through tubes, and the heat exchanger section based on PCM. Because most engineering systems use cylindrical tubes and heat loss from the shell and tube system is minimal. The shell-and-tube heat exchanger as a portable cooler is operated in crossflow. The tubes are vertical and the room air is fan-driven through the shell. PCM in vertical tubes is unmixed, airflow in shell is mixed. The external dimensions of the heat exchanger section are 1.35 m length (L), 0.21 m width (W) and 0.80 m height (H) [26].

The fans are positioned vertically and the overall length of the fans is 1.55 m. Also a fan which is 0.20 m length has been located centrally and height and width are the same as in the heat exchanger. In the fans, the velocity u_∞ of air outside the tube array is changed from 1.2 m/s to 2 m/s [26].

The dimensions of tubes are diameter (d) of 0.01 m; height (H) of 0.80 m. They are vertical, thin, and circular and are made of aluminum [26]. The square in-line configuration of the tubes is $S_p = S_n = 1.5 \times d$, as shown in Fig.2, where S_n is the tube pitch normal to flow and S_p is the tube pitch parallel to flow. The location of the tubes in the exchanger are indicated in Fig. 2, detailing as $S_p = S_n = 1.5 \times d$.

The external size of the exchanger determines by reducing 25%, 50% and 75%, except for the current size of a conventional air conditioner cooler.

The dimensions of a real room has selected as the following: height 2.5 m, length 4.0 m, and width 4.0 m [27]. The initial temperatures of air in the room is $T_{r,i}$ =35 ◦C, 37 ◦C, 40 ◦C and 45 ◦C. The room temperature at any instant is uniform (the room air is mixed). Radiation inside the room is ignored.

For simplification of calculations, all the thermophysical properties are assumed independent of temperature. The PCM is homogeneous and isotropic as both a liquid and a solid. The phase change material is a paraffin wax

(RT-25 by Rubitherm). The thermophysical properties of the PCM are as follows: melting temperature Tm, 23◦C; specific enthalpy of melting hm, 206 kJ/kg; liquid density ρl, 750 kg/m3; solid density ρs, 800 kg/m3; thermal conductivity k, 0.2 W/m K; specific heat capacity cp, 2500 J/kg K [26].

In calculations, a real-size room is cooled at daytime by a unit in which the PCM melts at 23◦C, while the ambient temperature is changed from 35◦C to 45◦C. Then, the paraffin solidified at night, and thermal comfort is preserved in the room. The calculations are conducted analytically.

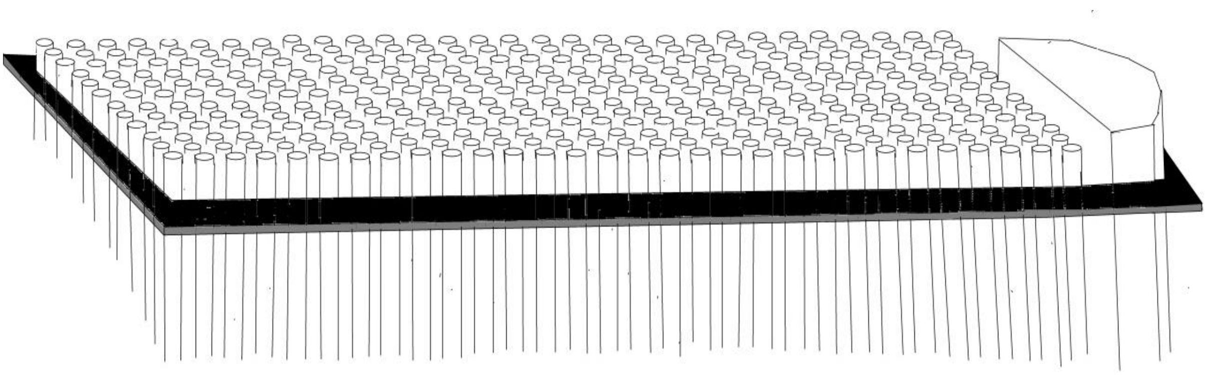


Figure.1 The structure of the heat exchanger system.

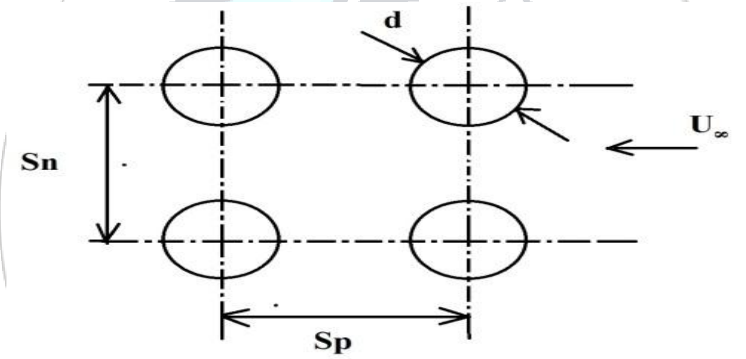


Figure.2 Square in-line configuration of tubes in the heat exchanger.

where N Total number of tubes. N_n and N_p is number of tubes in a row normal to flow and number of tubes in a row parallel to flow. The width and length of the configuration is as follows:

$$W = N_n.S_n = N_n.(1.5 \times d) \tag{1}$$

$$L = N_p.S_p = N_p .(1.5 \times d) \tag{2}$$

The total number of tubes in the exchanger is $N = N_n.\ N_p = 14 \times 90 = 1260$. The liquefied PCM mass in tubes fills each tube up to its top. The mass can be expressed [26]:

$$M_{PCM} = N.V_{tube}.\rho_l = N.\left(\left(\pi.d^2/_4\right).H\right).\rho_l \tag{3}$$

For the crossflow of air, the convection heat transfer coefficient outside the tubes (h_o) is calculated by the Grimson correlation [27]:

$$\left[h_o.d/_{k_a}\right] = C.\left(\left(U_{max}.d/_{\nu_a}\right).H\right)^n.Pr^{1/3} \tag{4}$$

Where C, n are constants of relation. For an square in-line configuration, C = 0.278 and n = 0.620 according to the pitch-to-diameter ratio $S_n/d = S_p/d = 1.5$. The properties have been evaluated as air at atmospheric pressure and room temperature of 27◦C (300 K). The air density ρ_a= 1.177 kg/m^3, specific-heat capacity $c_{p,a}$ = 1006 J/kg K, thermal conductivity k_a =0.026 W/m K, kinematic viscosity ν_a= 15.7x10^{-6} m^2/s, and Prandtl number Pr = 0.708. In an square in-line arrangement, the air velocity at the minimum frontal area U_{max} is expressed [26]:

$$U_{max} = U_{\infty}.\left({}^{S_n}/_{(S_n - d)}\right) \quad (5)$$

For $S_n/d = 1.5$, substituting above values in the equation changes the velocity versus U_{∞}. Thus, the air heat capacity rate is

$$C_a = \dot{m}_a.c_{p,a} \quad (6)$$

where the mass flow rate of air across the free cross-sectional area A_c is [26]:

$$\dot{m}_a = U_{\infty}.\rho_a.A_c \quad (7)$$

Where the area A_c for flow of air from the fans into the tube array is $A_c = W.H$. The area changes also according to reducing 25%, 50% and 75%. Mass of air in the room can be given [26]:

$$M_a = V_{room}.\rho_a \quad (8)$$

The shell dimensions of shell-and-tube heat exchanger are changed as rate of percent. Because of the thinness of the tube wall, the inside and outside surfaces are accepted as the same. The surface heat-transfer area of the tubes A can be given as [26]:

$$A = N.\pi.d.H \quad (9)$$

3. METHODS

The analysis of the PCM based heat exchanger unit is investigated for two conditions. In conditions, the exchanger operates for an insulated room and a room heated by the ambient. Each condition is to be analyzed in a way: An ideal performance of the exchanger. The wall temperature of all tubes is preserved at the constant temperature of melting Tm. This is an extreme condition. In the actual performance of the exchanger, the temperature of a tube wall varies with the melt fraction accumulated in the tube.

Both performances are compared with respect to the temperature achievable in the room, the range of operation of the ambient temperature and the design of the exchanger.

The melting of the PCM advances gradually from the first row to the last one. The wall temperature has risen above the melting point, while the tubes downstream are full of solid PCM and their wall is preserved at the melting temperature.

3.1. Room perfectly insulated

For ideal performance**, t**he melting-point temperature is preserved throughout the whole operation at the wall of all the tubes in the exchanger: $T_w = T_m$. The lowest temperature achievable in the room could be $T_{r,min} = T_m$. In such a case, the heat absorbed from the room by the exchanger and the PCM mass to be melted would have to be [26]:

$$Q_a = \left(M.c_p\right)_a.\left(T_{r,o} - T_m\right) \quad (10)$$

$$\Delta Q_{PCM} = Q_a = (\Delta M.\Delta h)_{PCM} \quad (11)$$

At such a specific melt fraction most of the tubes remain fully solid and at a temperature T_m. The time required to reach any comfortable temperature is assessed from the heat balance over the air [26-29]:

$$-\left(M.c_p\right)_a {}^{dT}/_{dt} = \left(\dot{m}.c_p\right)_a (T_r - T_m) \quad (12)$$

$$\Delta t = -\left({}^{M}/_{\dot{m}}\right)_a \ln\left[(T_r - T_m)/\left(T_{r,o} - T_m\right)\right] \quad (13)$$

3.2. Room heated by the environment

Heat is transferred from the surroundings to the room through the walls and ceiling. The floor is adiabatic. An overall heat-transfer coefficient (U_r) is assigned to both the walls and the ceiling and 1.2 W/m^2K. The ambient room temperature during the whole operation of the exchanger cooling is changed from 35◦C to 45 ◦C. For ideal performance, the heat balance over the room is [26]:

$$\left(M.c_p\right)_a {}^{dT}/_{dt} = (A.U)_r(T_{\infty} - T_r) - \left(\dot{m}.c_p\right)_a (T_r - T_m) \quad (15)$$

where the inlet temperature of air to the exchanger is the room temperature, $T_{in} = T_r$ and its outlet temperature is $T_{out} = T_m$ in the very long heat exchanger. At steady state, the temperature could be given as [26]:

$$(A.U)_r(T_\infty - T_r)_{ss} = (\dot{m}.c_p)_a(T_r - T_m)_{ss} \tag{16}$$

$$T_{r,ss} = \left[\left((A.U)_r/(\dot{m}.c_p)_a\right).T_\infty + T_m\right]/\left[1 + (A.U)_r/(\dot{m}.c_p)_a\right] \tag{17}$$

$$\mathrm{NTU} = {(A.h_o)}/{C_a} \tag{18}$$

where Ar = 4 × (4 × 2.5) + 4 × 4 = 56 m^2. At this temperature the rate of heat gains through the walls and ceiling amounts to [26]:

$$q_{amb} = (A.U)_r(T_\infty - T_{r,ss}) = (\dot{m}.c_p)_a(T_r - T_m)_{ss} \tag{19}$$

The duration of the exchanger operation in this mode can be estimated from the heat capacity of the PCM mass [26]:

$$\Delta t = {Q_{PCM}}/{q_{amb}} = {(M.\Delta h)_{PCM}}/{q_{amb}} \tag{20}$$

4. RESULTS AND DISCUSSION

In this calculation the time duration of the operation is determined, considering the initial ambient temperature of the room and the dimension change of the exchanger. All the figures given above relate to an ideal operation in which the tubes are maintained at the melting temperature. For actual performance, the full analysis of an actual performance is presented in literature.

However, if the equations described in the section are adopted to the insulated room and room heated by the environment, it would be obtained the evolution of the room temperature with time and the cooling time in operation. The room temperature versus time is plotted in Fig. 3, 4 and 5. As estimated above, while the different room temperature investigates from 35 ◦C to 45◦C, the temperature decreases from 35 ◦C to 23 ◦C (or 25◦C). Thus, the ideal and actual performances are the same.

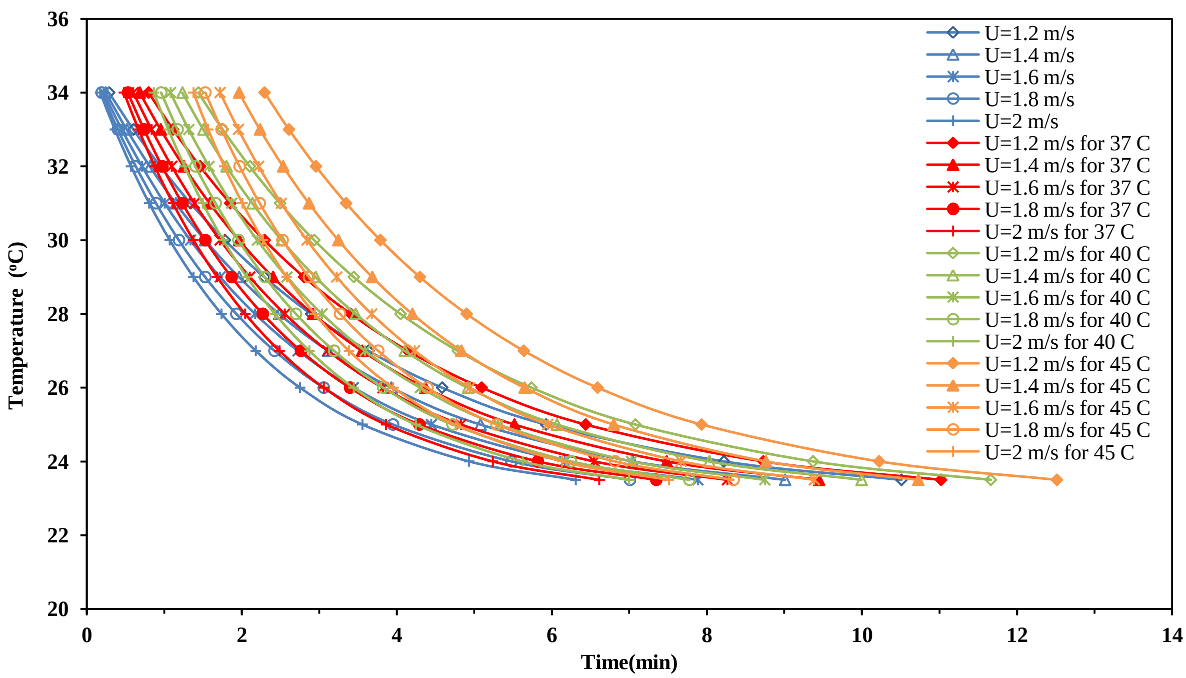
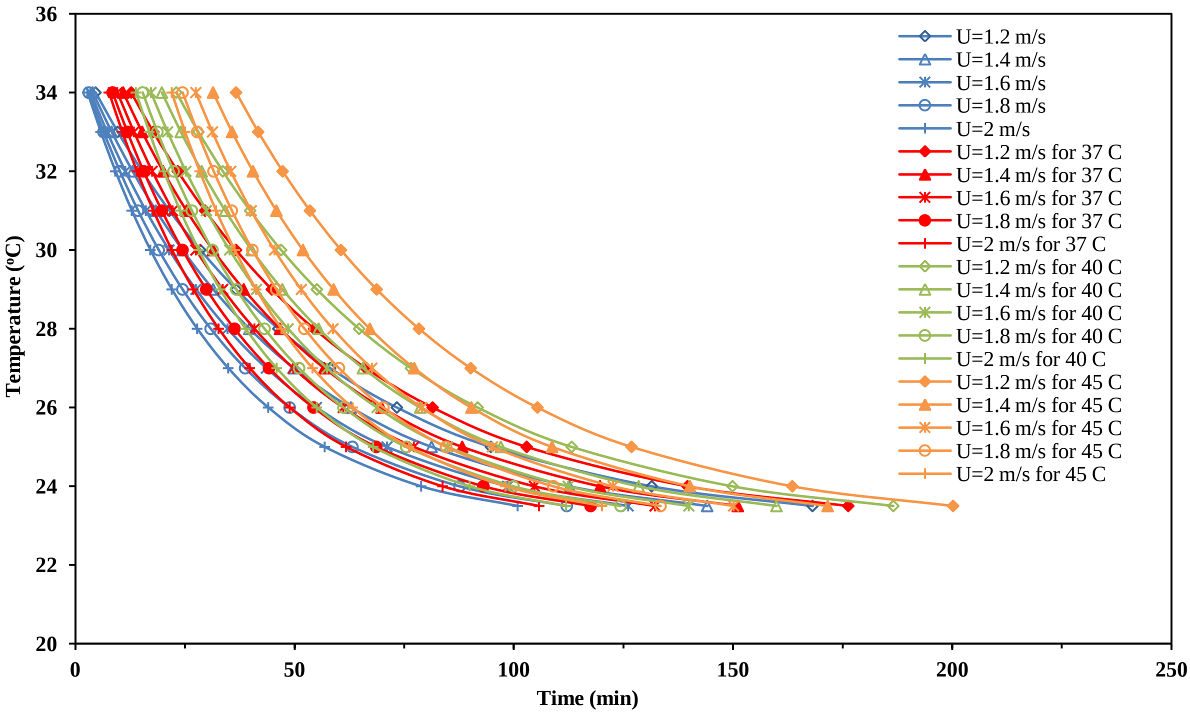
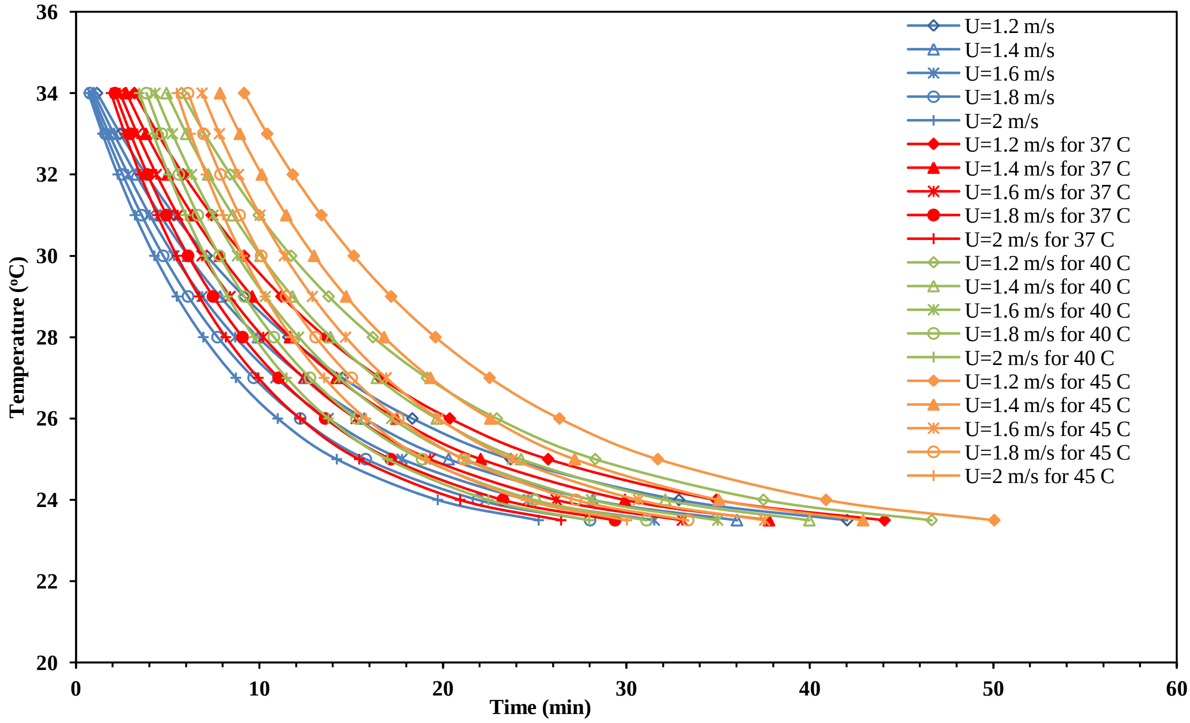


Figure.3 Cooling of an insulated room for different velocity and different ambient temperature: room temperature versus time for current exchanger dimensions.

Figure 3 displays the comparison between different velocity and different ambient temperature results and involves the change of the room temperature versus time. In this comparison, the inlet temperatures and the velocities for ambient are taken to be 35 °C, 37 °C, 40 °C, 45 °C and 1.2 m/s, 1.4 m/s, 1.6 m/s, 1.8 m/s, 2.0 m/s respectively. It is obvious from this figure that room temperature decreases as time go by. The amount of time for high ambient temperatures seems lower than the low temperatures. But when the velocity is decreased, the amount becomes larger than the others. This is expected due to the heat transfer factors on the current exchanger for lower ambient conditions.



(a)



(b)

Figure.4 Cooling of an insulated room for different velocity and different ambient temperature: (a) room temperature versus time for current exchanger dimensions of 25 %; (b) room temperature versus time for exchanger dimensions of 50 %.

Figure 4, 5 shows the change of temperatures of thermal energy stored as a function of time for different heat exchanger dimensions (25%, 50% and %75). The time decreases increasing ambient velocity, decreasing ambient temperatures. In addition, it can be seen from the figures that dimension has a dominant effect on the temperature of stored energy than conditions. If the dimension is decreased 50% for the heat exchanger having same condition parameters, the amount of time is decreased from 100.8 min to 25.2 min, approximately. The effect of different dimension of 75% on the time is given on Figure 5. These results show that the increase in the ambient temperature increase the time rate, as expected. This rate also increases with the decreasing ambient velocity. The change of temperature in stored energy is dramatically decreased after the dimensions is passed from current to 75%.

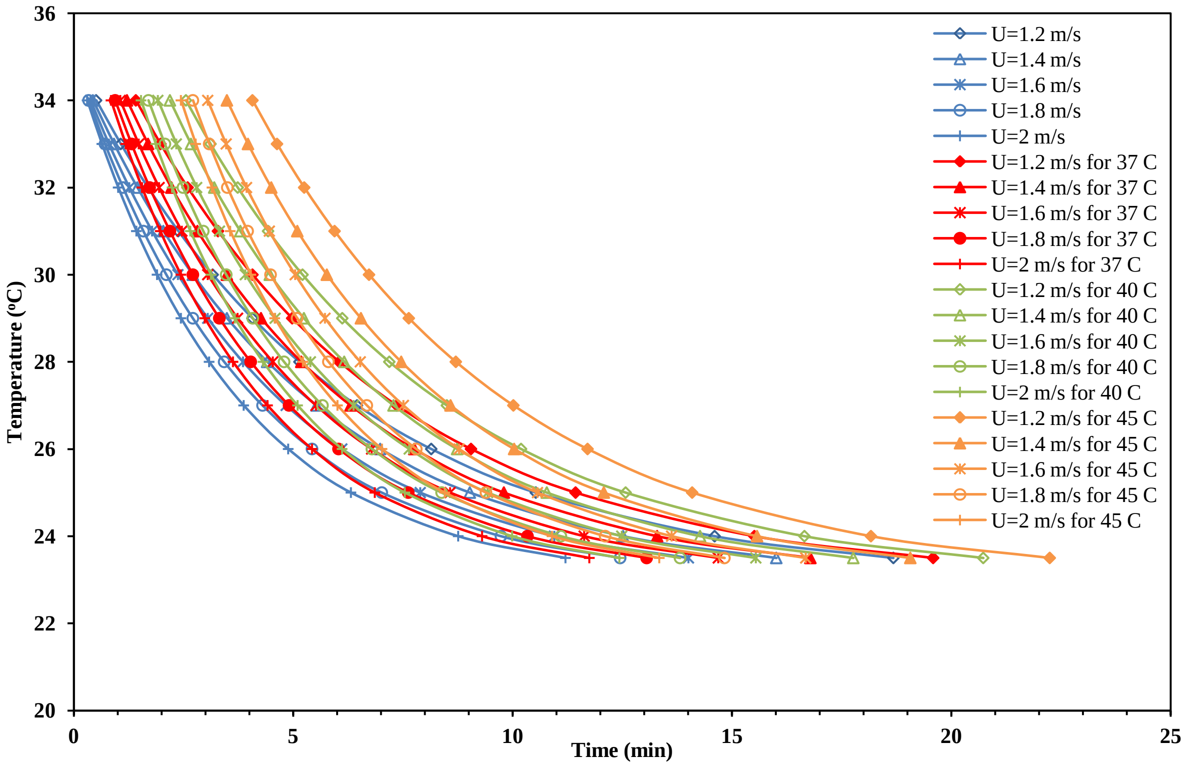
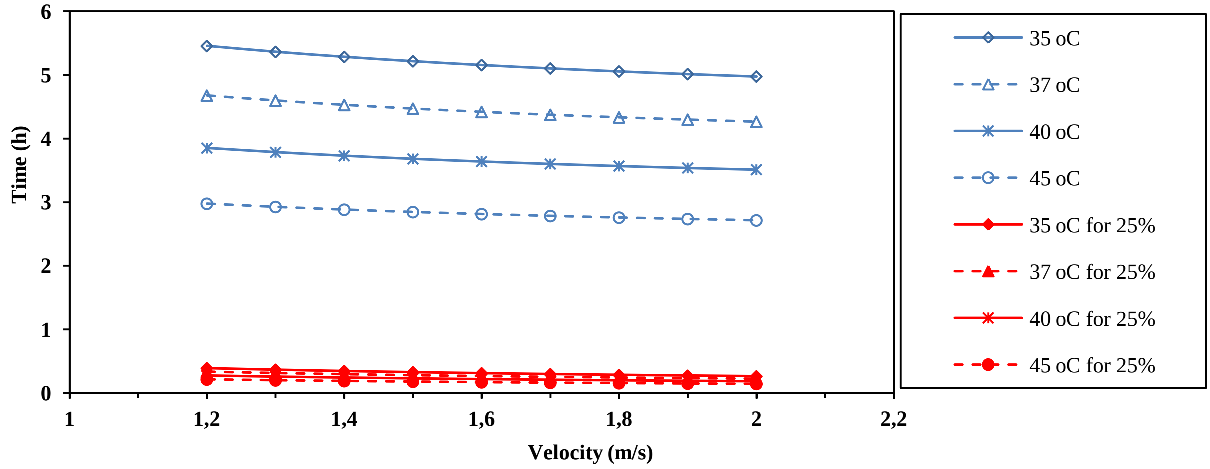


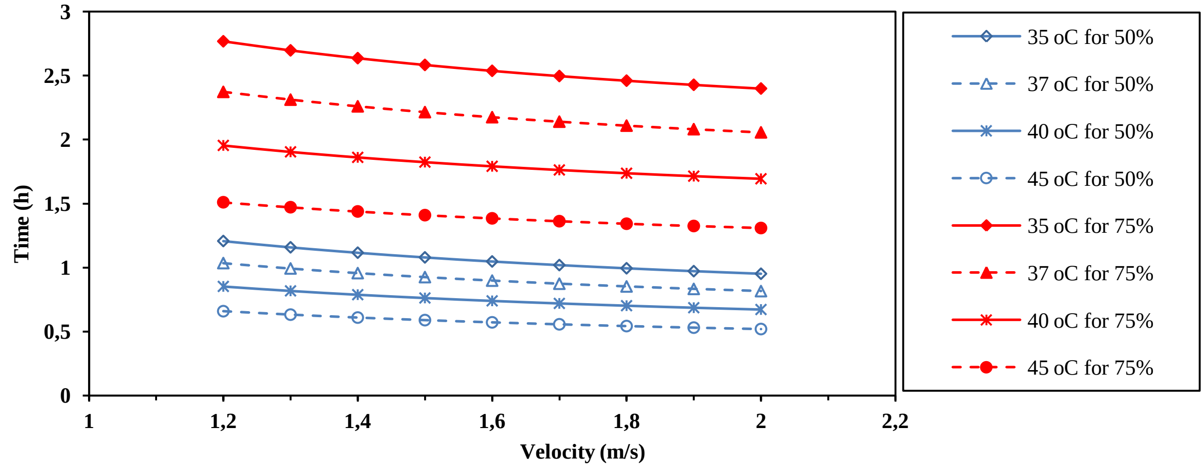
Figure.5 Cooling of an insulated room for different velocity and different ambient temperature: room temperature versus time for exchanger dimensions of 75 %.

Figure 6 shows the cooling time for the different ambient room temperatures reaching 23◦C. In the cooling time of a room heated by the environment, the variation of velocity with time is also shown in Fig. 6. These times remain at same level for different ambient room temperatures. It can be seen that the effect of velocity on the time is not very significant for the cooling time of a room heated by the environment. In the room guarded as thermal, the heat extraction decreases and the time passes as distribution temperature decreases.

It can be seen from these figures that the effects of dimensions on the decrease of time are excessive than ambient parameters. As the effect of ambient conditions, the change of time can be accelerated by decreasing the dimensions. The effect of ambient temperature on the times depends on the difference between the fluid inlet and melting temperatures, as shown in Figure. It seems that the effect of increasing ambient temperature is more significant than increasing the velocity.



(a)



(b)

Figure.6 The cooling time of a room heated by the environment for different velocity and different ambient room temperatures: time versus velocity for exchanger dimensions of (a) current, 25 %, (b) 50 % and 75 %.

4. CONCLUSIONS

In all the calculations presented herein a few simplifying assumptions were involved. The density of liquid PCM was used for both liquid and solid. The room temperature was assumed uniform, as if the air were perfectly mixed. Air properties were considered constant. Radiation in the room was ignored. These simplifications affect the closeness of the results to a real situation, but enable the comparison of solutions. The analytical solution is the simplest, while the real solution is a heavily time-consuming procedure. Two cases were analyzed: cooling of an insulated room and a room heated by the ambient. Different cases were solved analytically, for an “ideal” performance. In all cases the analytical results were close to the literature [26] results and could have been used for estimates of the room temperature.

The analytical calculations related to exactly the different dimensions. The solution was also based on different ambient room temperatures. The results obtained by this method could have been the most accurate. However, in the present study the analytical method provides the cooling times of the PCM process.

Another issue of interest is the temperature distribution of the heat transfer fluid along the heat exchanger. It has been show that, during periods of the cooling time of a room heated by the environment, the HTF temperatures are dominated by the second case, but the change of dimension in the system is very significant than HTF temperature.

Finally, in relation between the cases and the conditions, data have shown that the amount of time by the HTF in the insulated room is negligible compared to the cooling time of a room heated by the environment.

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