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NUMERICAL STUDY OF USING PHASE CHANGE MATERIAL (PCM) TO ENHANCE THE PERFORMANCE OF AIR-CONDITIONING (AC) UNIT

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ABSTRACT

Split-type air conditioning system is widely used because of its simplicity and flexibility. The performance of the split type air conditioner units strongly depends on the surrounding temperature of the outdoor units. (Samuel, Nagendra, and Maiya 2013) reported that for air-cooled air-conditioner if the on-coil temperature of the condenser is raised by 1 ^{o} C, the COP of the air-conditioner drops by around 3%. (Zhao and Tan 2015) study of using shell and tube heat exchanger filed with PCM integrated with water cooled condenser. Thermal storage system is an effective way to make good use of night cold energy and helps to reduce air-conditioning power consumption in the daytime. This research proposed new technique of using PCM in enhancement the coefficient of performance (COP) of the AC unit. This technique is based on integrating a plate-based phase change material (PCM) thermal storage system with the condenser of the conventional AC to increase its cooling COP. The proposed PCM thermal storage unit uses air as media for the heat transfer fluids (HTF). A numerical model for the PCM thermal storage plates has been established, especially in consideration of staged natural convection effects on the melting process inside the PCM material. A numerical study has evaluated the effects of HTF inlet temperature and air flow rate on the PCM thermal storage system's performance.

1. INTRODUCTION

The usage of air conditioning is increasing rapidly all around the world with the increasing population and modernization of the cities. One of the novel concepts of using PCM in air conditioning purpose is called "free cooling" or night ventilation system which latent heat storage using phase change materials (PCM) can be used for free cooling. Night ventilation, one type of free cooling systems, is the use of low-temperature night air to cool down the structure of a building directly. It is a useful and low-cost approach to improve indoor thermal comfort and reduce the daytime cooling load of air conditioners in summer if there is a substantial diurnal temperature shift. In this model low air temperature is used to solidify the PCM during the night and then during the next day, the inlet air of condensing unit can be cooled down by exchanging heat with PCM. There are good analyses on PCM-based cooling for building applications (Khudhair and Farid 2004; Vakiloroaya et al. 2014; Zalba et al. 2003) investigated and discussed the different technologies and approaches, and establish their capability to improve the performance of HVAC systems to reduce energy consumption. (Khudhair and Farid 2004) summarised the investigation and analysis of thermal energy storage systems integrating PCMs for use in building applications. (Zalba et al. 2003) carried out a review of the history of thermal energy storage with solid-liquid phase change. Three aspects have been the emphasis of this review: materials, heat transfer and applications. (Giro-paloma et al. 2016) presents a review, which includes the different types of PCM that can be microencapsulated, the various shell materials used and the way to microencapsulate this PCM. (Osterman E, Tyagi VV, Butala V, Rahim NA 2012) summarised the PCM based cooling technologies for buildings, including free cooling applications, cold storage air-conditioning system with capsules packed bed (e.g., ice storage), ventilation cooling system based on PCM, and PCM based envelopes. (Pomianowski Michal, Heiselberg Per 2013) reviewed thermal energy storage technologies based on the PCM application in buildings such as PCM in construction materials (passive/active), PCM in glazing, shadings, blinds, and PCM in HVAC components/heat exchangers. Nevertheless, existing integration of PCM with the air-conditioning system either uses water as the phase change material or is for a loose coupling such as ventilation focused.

The aim of using PCM is to reduce the energy consumption and the peak time problem. PCM one type of latent heat thermal energy storage system (LHTESS) free cooling applications dominantly used air as heat transfer fluid (HTF) in both charging and discharging processes as they were focused on ventilation. Therefore, the cold energy stored in PCM at night is directly discharged into the building through indoor air circulation at daytime. However, for the most intensely analysed LHTESS, plate type PCM (Aroul and Velraj 2011; Ni et al. 2016; Wang et al. 2007a, 2007b; Zhao and Tan 2015), air is used as HTF. In this research, to integrate PCM thermal storage with conventional air conditioner, air has been used as HTF for night time and daytime heat for discharging and charging process, respectively. The PCM thermal storage will act as air-conditioner's heat sink, which will help reduce the condenser side temperature and thus increase air-conditioner's COP. In the recent years, many studies had been conducted using PCM in air conditioning applications. (Wang et al. 2007a) Developed and studied use two heat exchanger of PCMs in refrigeration systems to dropping the temperature of the sub-cooled refrigerant and stabilising the superheat. Results showed that by dropping the temperature of the sub-cooled refrigerant, up to 8% energy savings could be accomplished in the UK climate conditions. By using the PCM heat exchanger as a pre-condenser in a refrigeration system, the systems COP can be enhanced by 6%. The benefit achieved by reducing the superheat is somewhat offset by its additional pressure drop occurs in PCM heat exchanger. (Wang et al. 2007b) Develoed A dynamic mathematical model for coupling the refrigeration system and PCMs heat exchanger. (Hoseini Rahdar, Emamzadeh, and Ataei 2016) studied two strategies of hybrid systems via a vapor compression A/C system. First, an ice thermal energy storage system (ITES) is used in the night time hybrid system; and subsequently, a phase change material (PCM) tank is used as a full storage system (in order) to shift (the load) from on-peak to off-peak mode.

(Samuel, Nagendra, and Maiya 2013) reported that conventional mechanical air conditioning systems are energy intensive and increase the electric grid system's burden in summer peak hours. Research shows that for air-cooled air-conditioner if the on-coil temperature of the condenser is raised by 1 ^{o} C, the COP of the air-conditioner drops by around 3% (Avara and Daneshgar 2008). Therefore reducing air-conditioner condenser's operational temperature will be highly beneficial for the air-conditioner's performance efficiency, which has attracted great efforts from researchers, such as using new types of refrigerant (Aprea and Maiorino 2007), optimizing placement of the condenser (Avara and Daneshgar 2008), investigating the effect of temperature stack (Bruelisauer et al. 2014) and roof reflectance (Wray and Akbari 2008).

The objective of the present work is to simulate plate type PCM storage numerically for PCM heat exchanger with transient heat transfer analysis for the PCM encapsulated of the module using apparent heat capacity model for phase change in Ansys 17.2 software. It is used in free cooling system with plate-type PCM. The results are validated with the experimental melting time determined for the selected air frontal velocity of 1.4, 1.2, 0.9 m/s and the comparison between the experimental and calculated values are analysed. This system has advantages of energy efficiency by decreasing air-conditioner condenser side's temperature due to cooler PCM than outdoor air for cooling periods. The effect of mass flow rate and inlet air temperature on the performance of the system was studied. With CFD analysis of such system, a graphical representation of different parameters such as melt fraction and temperature contours can be explored. This can be useful in designing free cooling system using plate-type PCM storage with vapour compression cycle. The effect of air inlet velocity and inlet fluid temperatures on the melting and solidification time and other parameters of interest are considered in detail and reported.

2. PHYSICAL MODEL

The physical model considered for the study consists of 6 PCM module placed in 0.3×0.27 m rectangular duct that allows to the air to passes between PCM plates. In this study, Rubitherm commercial available PCM (salt) SP24E was used because of its little volume expansion during melting process (lesser than 4%) and suitable temperature range for air conditioner 24 °C (Table 1). The capacity of the PCM present in one unit is 0.00133 m3 with a mass of 2 kg for a density of 1500, 1400 kg/m3 for solid and liquid phase respectively was considered in the analysis. The schematic of the heat exchanger presented in this study is shown in Fig. 1. The detailed geometry size of the heat exchanger and PCM plates are shown in this figure 3. The hot air enters the unit through an inlet aperture and then drowns over the cold storage and after that cooled air exits through the outlet aperture. The thickness of PCM plate's wall is set to 1 mm thick with a thermal conductivity of 237 (W/m K). The heat flux absorbed by PCM at each time step (j) is calculated by Eq. (1).

$$Q_j = m_j\, c_p\left(T_{i,j} - T_{o,j}\right) = \rho u_j A c_p\left(T_{i,j} - T_{o,j}\right) \qquad (1)$$

The configuration of PCM heat exchanger as shown in figure 1 are investigated in the present study has PCM plates with dimension 0.45 × 0.3 × 0.015 m in duct.

Table 1: Thermo-physical properties of SP24E (Rubitherm)(https://www.rubitherm.eu/ n.d.).

Melting temperature (°C)	24-25
Density (kg/m3)	1500-1400
Specific heat (J/kg)	2000
Thermal conductivity (W/m K)	0.6
Thermal expansion Co. (1/K)	0.001
Latent heat of fusion (kJ/kg)	180

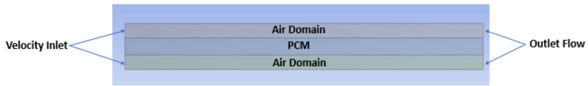

Fig 1: Physical model of the heat exchanger.

3. NUMERICAL SIMULATION

3.1 Governing equation

In the present study, both PCM and air flows are considered to be unsteady, turbulent, incompressible and two-dimensional. The viscous dissipation term is considered. The viscous incompressible flow and the temperature distribution are solved using the Navier– Stokes and thermal energy equations, respectively. Consequently, the continuity, momentum, and thermal energy equations can be expressed as follows:

Continuity:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho u_i) = 0 \qquad (2)$$

Momentum:

$$\frac{\partial(\rho u_i)}{\partial t} + \partial_j\left(\rho\, u_i\, u_j\right) = \mu \partial_{jj} u_i - \partial_i p + \rho g_i + S_i \qquad (3)$$

Energy:

$$\frac{\partial(\rho H)}{\partial t} + \partial_i(\rho\, u_i H) = \partial_i(K\, \partial_i\, T) \qquad (4)$$

In the above conservation equations, ui is the fluid velocity, ρ is the density, μ is the dynamics viscosity, P is the pressure, g is the gravitational acceleration, k is the thermal conductivity and H is the total enthalpy which is defined as sum of sensible enthalpy (h = href + ∫ Cp dT) and the latent heat content of PCM (ΔH):

$$H = h + \Delta H \qquad (5)$$

The latent heat content of PCM (ΔH) may vary between zero (solid) and L (liquid), the latent heat of the PCM. Therefore, melt fraction, β, can be defined as follows:

$$\beta = \begin{cases} \frac{\Delta H}{L} = 0 & if \;\; T < T_s \\ \frac{\Delta H}{L} = 1 & if \;\; T > T_s \\ \frac{\Delta H}{L} = \frac{T - T_s}{T_l - T_s} & if \;\; T_s < T_l < T_s \end{cases} \quad (6)$$

The latent heat content can now be written regarding the latent heat of the material, L:

$\Delta H = \beta L$ (7)

In Eq. (3), Si is Darcy's law damping terms (as source term) that are added to the momentum equation due to phase change effect on convection for PCM zone and it has zero value for air zone. It is defined as follows:

$$S_i = \frac{C(1-\beta)^2}{\beta^3}\, u_i \quad (8)$$

The coefficient C is a mushy zone constant which is fixed at a value of 105 kg/m3 s in the present numerical simulation.

$$\frac{\partial(\rho K)}{\partial t} + \frac{\partial}{\partial x_i}(\rho\, k\, u_i) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + P_k + P_b - \rho\epsilon - Y_M + S_k \quad (9)$$

$$\frac{\partial(\rho \epsilon)}{\partial t} + \frac{\partial}{\partial x_i}(\rho\, \epsilon\, u_i) = \frac{\partial}{\partial x_i}\left[\left(\mu + \frac{\mu_t}{\sigma_\epsilon}\right)\frac{\partial \epsilon}{\partial x_j}\right] + C_{1e}\frac{\epsilon}{k}(P_k + C_{3e}P_b) - C_{2e}\rho\frac{\epsilon^2}{k} + S_e \quad (10)$$

3.2 Initial and boundary condition:

At the initial time stage (t = 0), the PCM is taken to be a motionless solid that is maintained at a constant temperature 24 °C melting temperature of the PCM. Also, the no-slip boundary condition is applied for the wall boundaries, i.e. ui = 0. The heat exchanger is thermally insulated. Air flow enters with a constant velocity flow (1.4, 1.2 and 0.9 m/sec) and a constant temperature (45 °C, 40 °C and 35 °C). The boundary condition for the outlet from PCM was considered as a pressure outlet with zero-gauge pressure.

3.3 Computational methodology:

The Ansys software employs the finite volume method for solving the mass, momentum and energy equations. The fully implicit approach is implemented with the PISO algorithm as the solver option for transient simulation. A second-order upwind differencing scheme is used for the approximation of the convective terms, and the k-€ model is used for the turbulence modelling of the heat transfer fluid. The conservation equations are solved by the Ansys fluent 17.2 software for the prediction of the velocity, pressure and temperature solution in the domain considered for the analysis. Also, the PRESTO scheme is adopted for the pressure correction equation. The under-relaxation factors for density, momentum, pressure correction, thermal energy, and melt fraction are 1, 0.7, 0.3, 1 and 0.9, respectively. Figure 2 (a,b) illustrate different grid densities and time step are selected and tested to certify the independence of the solution from the adopted grid density and time step based on the comparison of melt fraction. The mesh element 51742 and time step 0.1 are found sufficient for the model in the present study. The mesh size and time step are considered adequate to satisfy the convergence criteria (10^{-6}) for continuity, (10^{-8}) for momentum equations, (10^{-8}) for K-€ model and (10^{-12}) for the energy equation. A transient simulation study is conducted to analyse the PCM melting and solidification time for various heat transfer fluid inlet temperatures and velocities, considering the initial temperature of PCM as 22 ^{o} C for different frontal velocities in melting process and 26 ^{o} C for solidification process respectively.

3.4 Thermal performance of Split type air conditioner.

The performance of the air conditioning units is normally determined by the coefficient of performance (COP) which measure the cooling load effect for each unit power input to the compressor. According to the manufactures Catalogue, the COP of the air conditioner can be correlated regarding the indoor and outdoor temperature; or for a certain indoor room temperature, COP can be simply correlated in terms of the outdoor temperature as follows:

$$COP = a - b\,T_o \quad (11)$$

Where the constants a and b depend on the indoor room temperature (T_R)(Nada and Said 2017)

The electric power consumption of the AC unit (W) is defined in terms of the COP and the unit capacity (Q) as:

$$W = Q / COP \quad (12)$$

4. RESULTS AND DISCUSSION

4.1 Melt fraction and inlet velocity variation:

When using PCM thermal storage system for air-conditioning application, there are three performance parameters should be considered: (1) HTF outlet temperature in heat charging (melting) process, (2) total charging time and (3) total discharging time. The first index is directly associated with the air-conditioning system's cooling COP since lower HTF outlet temperature results in lower condensing temperature and thus will enhance the air-conditioner efficiency. The second index is related to the time duration during which the PCM thermal storage unit can function properly. The third index is important because the PCM must be completely discharged at night for a sustainable use consideration.

The melt fraction of the model for different air flow rates are shown in Fig. 2. At the start, the pure heat conduction is dominant between the hot surfaces of PCM plates and cold solid PCM. The air thermal boundary layer commences in the air adjacent to the PCM plate almost before the melting surface of PCM. The PCM starts melting with the progression of time and so the amount of melted PCM increases. In this stage, natural convection is becoming important in the melted zone mainly in the layer between solid PCM and walls which enhances the melting rate at the top part of solid PCM. However, the conduction heat transfer decreases when the melted layer between hot surfaces of plates and solid PCM becomes thicker. The air thermal boundary layer is growing along the plates and subsequently, the temperature gradient between air flow and plate's surface decreases. The thermal boundary layer in the gap between the two PCM plates merged together, and as a result, the temperature of air which passes through the two PCM plates decreases more than the ones that pass between the PCM plate and the horizontal wall of the system.

As can be seen from Fig. 2, increasing the air velocity rate makes the melting faster and the difference between the amounts of solid PCM for three velocities is getting more significant with progressing of time. The variation of melt fraction versus time for different configuration with different air flows and different inlet temperature is shown in Figures. 2. The PCM melts earlier for higher inlet flow temperature and higher air flow rate. Changing the inlet flow temperature from 35 ^{o} C to 45 ^{o} C in configuration 1 decreases the full melting time by 47% whereas varying the air inlet flow rate from 0.96 to 1.44 m/sec at the same inlet temperature decreases the full melting time by 24%.

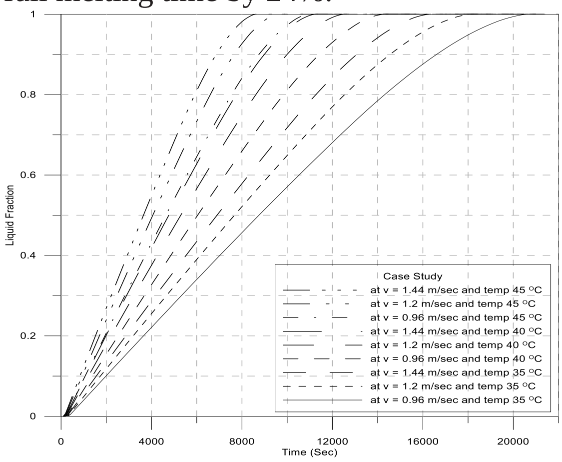

Fig 2: Melting fraction at different inlet velocity and temperature.

4.2 Outlet temperature and inlet velocity variation:

Fig. 3 illustrates the effect of air flow rate and inlet flow temperature on the average air temperature in the outlet of heat exchanger. As it can be seen, the outlet temperature increases with progressing time. This is because of the melting of PCM which decreases the heat transfer surface between the remaining cold solid PCM and hot air (Fig.2). The outlet temperature is increased by increasing the air flow rate and the inlet flow temperature. The average decreases in air temperature with an inlet flow velocity of 0.96

m/sec and 1.44 m/sec at an inlet temperature of 35 ^{o} C and an inlet flow velocity 0.96, and 1.44 m/sec at an inlet temperature 45 ^{o} C are 4 °C, 3 °C, 7 °C and 5 °C during 3.5, 2.75, 2 and 1.40 h, respectively. As a result, there are two ways in order to have a lower outlet temperature: decreasing the inlet air temperature or reducing the air flow rate. For a constant latent and specific heat of PCM, it depends on the temperature difference between the inlet air and melting temperature of PCM. In order to, decrease the outlet temperature one of the ways is to decrease the inlet temperature, however, it depends on the ambient condition and we could not control it. Lastly, another way is to use a PCM with higher latent heat of fusion and lower specific heat. Thus, the best way to lower the outlet temperature is to reduce the air flow rate.

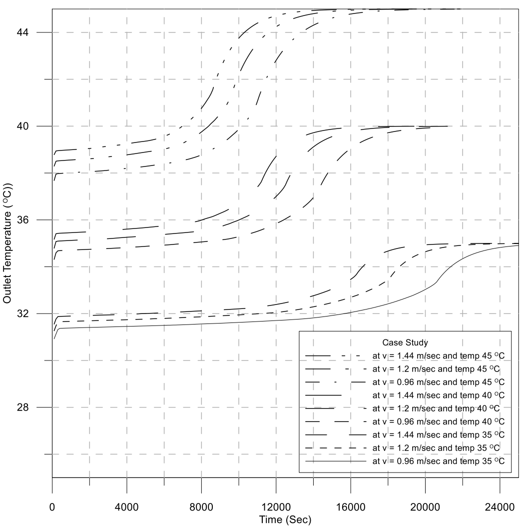

Fig 3: Outlet temperature at different inlet velocity and temperature.

4.3 Consumed power of AC unit

It is important to know the saved power from using this new technique with respect to conventional AC unit. Fig. 4 summarizes the average power consumed of the AC unit in kilo watts per ton refrigerant for ten working hours of the AC unit (maximum possible working hours of the AC unit during the day). The consumed power is calculated based on equation 12 and the results are presented at different inlet air velocities, inlet air temperatures. The inlet air temperature to the PCM plates is the same inlet temperature to the conventional AC condenser. The power consumed for the new technique is smaller than the conventional one for all studied input parameters. Additionally, the maximum percentage of the saved power for the new technique with relative to the power of the conventional AC at inlet velocity 0.96 m/s is about 7.2, 6.1 and 5.25% for inlet air temperature 45, 40 and 35 ^{o} C respectively. The consumed power is increased with raising the inlet air temperature however it is increased slightly with increasing the inlet air velocity. At v=0.96 m/s, decreasing the inlet air temperature to the PCM plates by about 28%, increases the consumed power by about 37% and increasing the air velocity by about 50%, decreasing the consumed power by 40% maximum. To profit of the positive effect of decreasing the CCT due to increasing the inlet air velocity to the PCM, more PCM material will be required to cover the whole working time of the AC unit.

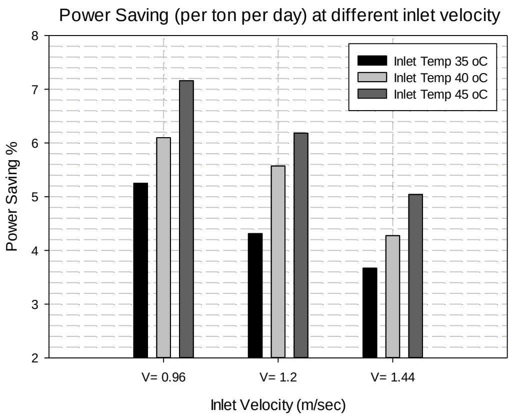

Figure 4: Saving power per unit TR per day.

3. CONCLUSIONS

In the present study, the effect of air flow rate for the different conditions of the ambient air temperature on PCM plate heat exchanger was studied. Also, the evolutions of the melt fraction of the PCM material and the temperature variation versus time were also presented numerically. Moreover, the consumed power of the AC unit was studied. The following results can be concluded from the study:

- Modelling results show that the PCM heat exchanger can reduce the ambient inlet temperature by 5 ^{o}C to 7 ^{o}C.
- The melting duration strongly depends on the fluid inlet temperature and air flow rates. The discharging time and the outlet cold

temperature from the PCM plates decrease with increase inlet air velocity and temperature.

- Using PCM plates heat exchanger enhance the performance of air conditioning units by 5 to 7 %

Nomenclature

A	area	m^2
C	mushy zone coefficient	
Cp	specific heat	J/kg.K
g	acceleration gravity	m/s^2
G	generation of turbulence kinetic energy	
H	total enthalpy	J
h	specific enthalpy	J/kg
k	thermal conductivity	W/m.K
L	latent heat	J/kg
m	mass	kg
Q	heat flux	W/m^2
p	pressure	pa
T	temperature	K
u	velocity	m/s
V	inlet velocity	m/s
Si	darcy's law damping coefficient	
W	power consumption	W

Greek symbol

ρ	density	kg/m3
μ	dynamics viscosity	m/s^2
β	melting fraction	
ΔH	latent heat content	

Abbreviations

AC	air conditioning
COP	coefficient of performance
PCM	Phase change material

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