STUDY OF HEAT TRANSFER AND PRESSURE DROP CHARACTERISTICS OF AIR HEAT EXCHANGER USING PCM FOR FREE COOLING APPLICATIONS

by

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Free cooling is the process of storing the cool energy available in the night ambient air and using it during the day. The heat exchanger used in this work is a modular type which is similar to the shell and tube heat exchanger. The shell side is filled with phase change materials and air flow is through the tubes in the module. The modules of the heat exchanger are arranged one over other with air spacers in between each module. The air space provided in between the module increases the retention time of the air for better heat transfer. Transient computational fluid dynamics modeling is carried out for single air passage in a modular heat exchanger. It shows that the phase change materials phase transition time in the module in which different shape of fins is adopted. The module with rectangular fins has 17.2% reduction in solidification compared with the plain module. Then steady state numerical analysis is accomplished to the whole module having the fin of high heat transfer, so that pressure drop, flow and thermal characteristics across the module and the air spacers are determined for various air inlet velocities of 0.4 to 1.6 m/s. To validate the computational results, experiments are carried out and the agreement was found to be good.

Key words: *phase change material, thermal energy storage, modular heat exchanger, free cooling, energy efficient phase change materials*

Introduction

In the present scenario energy crisis is the major issue which needs to be resolved. Conventional energy resources are the main cause for pollution of earth. Conventional energy resources are the main cause for pollution on earth. Most of the present technology and applications uses non- renewable energy sources thus polluting the mother earth. so this stimulates a search for pollution free renewable energy sources and energy efficient technologies. Because of severe energy and environmental issues, the energy-saving and sustainable developments of buildings are of utmost concern to the building industry. Passive cooling techniques can optimally utilize natural resources in order to reduce the energy consumption of buildings. At the same time, it can improve the buildings' thermal environment.

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Free cooling application is the passive cooling technique which is an energy efficient cooling technique which can be provided by natural cooling. Night ventilation offers the cooling of structural elements, thus providing more appropriate living conditions for the following day. Such natural cooling is especially efficient in climatic conditions, where the diurnal temperature variations of the ambient temperature are high. In passive cooling techniques thermal energy storage is used effectively to store the energy. Latent thermal energy storage (LTES) has received considerable attention due to its advantages of storing a large amount of energy during phase transition occurring at a constant temperature than the sensible heat storage. Free cooling concept is best suited for places where the diurnal temperature variation is high for at least 15 ºC.

The research studies dealing with the numerical analysis and the experimentation pertaining to the incorporation of a variety of geometric configurations of phase change materials heat exchanger and PCM encapsulation have been reported, in the recent years. In this context, it was ascertained that, the solidification time of the PCM encapsulation is much sensitive to the charging air temperature rather than its flow rate [1]. The experimental work conducted on a real-scale prototype of a PCM-air heat exchanger being empirically modeled has revealed that, for the application requiring constant temperature, the power consumption would influence the heat storage performance of such system during the charging periods [2]. For the free cooling applications in buildings, the factors related to the selection and adaptation of PCM plays a vital role in achieving good heat storage potential and energy efficiency [3].

Likewise, a modular heat exchanger was developed especially to cater the thermal energy storage requirements of buildings located in climatic conditions that has lower diurnal temperature variations. The experimental and numerical analysis performed in this study infers that, with the application of night ventilation concept, the modular heat exchanger integrated with the PCM showed improved thermal storage performance even at varying diurnal temperatures [4]. Interestingly, with the implementation of free cooling technique using the latent functionally heat storage materials in buildings, the size of the mechanical ventilation systems can be effectively reduced [5]. For buildings located in northern Europe, the application of free cooling or night ventilation which includes the heat storage material as part of the constructional design has showed huge potential for achieving enhanced energy savings and energy efficiency in buildings [6]. The significance and the application of a variety of PCM for different thermal energy storage systems has been collectively reviewed and reported [7]. The design of a real-time free cooling system was performed and the evaluation of which has demonstrated its economic viability for practical applications [8]. The study on the indoor and outdoor temperature variations of a building located in Israel was analyzed [9]. It was suggested that, using the developed design tool, the change in the indoor temperature with respect to the outdoor temperature variations could be well established, by virtue of the combined effects produced by the building's thermal mass and the night ventilation scheme. The numerical analysis on a thermal energy storage system comprising of multiple PCM was studied for night ventilation application in buildings [10]. In the spectrum of computational fluid dynamic analysis, predicting the characteristics and performance of wide range of heat exchangers for various application is more efficient. The heat transfer and pressure drop characteristics are influenced only by the shape and due to the symmetry of the heat exchanger only a partial section can be examined. [11,12]. The one dimensional numerical analysis performed on a real-time PCM thermal energy storage system using simulation models infers that, the thermal storage performance of the system being predicted has deviated by 12% with respect to the experimental measurements [13]. From the perspective of operating conditions, the PCM possessing wider phase transition temperature range can be considered to be efficient, when incorporated into the TES systems which is being operated at different climatic conditions [14]. Initially the thickness of the solidified layer is small and hence the thermal resistance is small,leading to a high solidification rate. As time passes, the thickness of solidified PCM increases, causing an increase in the thermal resistance and a reduced solidification rate [15]. It has been found that four times of air flow rate will be required as compare to required air flow rate during the day time so that it can be charged in limited night time of eight hours and it can provide comfort temperature for about eight hours during day time[16]. In the present work heat transfer and pressure drop studies have been carried out on air heat exchanger using PCM for free cooling in a modular heat exchanger. This study predicts the best heat transfer geometry of the modular heat exchanger and to have the minimum pressure drop for the heat transfer fluid flowing through the heat exchanger. PCM temperature gradually increases with time and remains constant during the phase change and continues to increase after the phase change before it attains charging a temperature [17]. To extent the cooling period the choice of material is important as the melting time will depend on not only the amount, but also the type of PCM used [18].

System description and methodology

Free cooling concept is best suited for places where the diurnal temperature variation is high for at least 15 ºC, so for this type of application the heat transfer area should be more, so to adapt for this condition free cooling system with modular heat exchanger is introduced [4]. The PCM are selected on the basis of high latent heat. In the present work, one module with PCM and two air spacers is considered for analysis. The system considered for free cooling application is shown in fig.1 the modules are arranged one over the other with air gaps in between. Here the room air is connected to the system, during night time dampers 1 and 4 opens and 3 and 2 are closed, so the cold ambient air enters through the PCM module the PCM gets solidified and the ambient air passes away. During day time dampers 1 and 4 are closed and 2 and 3 are opened so the room air circulates through the PCM module gets cooled and the cold air is again circulated to the room.

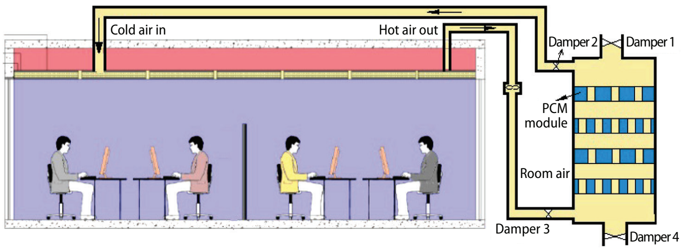

Figure 1. Working operation of a free cooling system

The heat transfer and pressure drop characteristics are studied in modular heat exchanger with different pattern of fins. The 3-D model considered in the present work is a modular heat exchanger which consists of PCM module and air flow path with two air spacers in top and bottom as shown in fig. 2. The dimensions of the PCM module are shown in table 1.

The mass of the phase change material will be high in rectangular fin compared to circular fin. The volume of air flow will be high in module with no fin and the volume will be low for rectangular fin. But the difference in mass of air flow rate is very less.

Table 1. Dimensions of the module

S.NO	Dimensional parameters	Dimensions in [mm]
1	Number of holes per module	19
2	Number of fins in a air flow path	4
3	Length of the fin	20
4	Breadth of the fin	10

The module contains holes for air flow and except air flow hole area , all the spaces are stacked with PCM. The module maintains

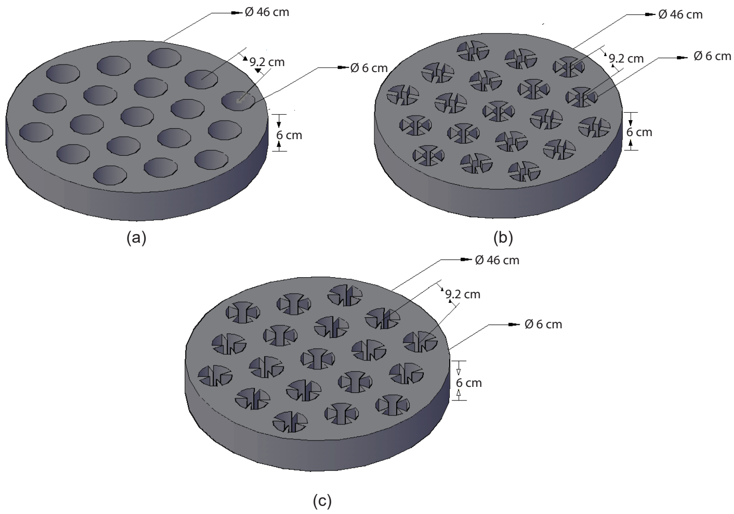

Figure 2. Modular heat exchanger; (a) without fin; (b) with triangular fin; (c) with rectangular fin

its original form during the PCM transition between solid to liquid and vice versa because the PCM is stacked inside the module made of copperand the phase of PCM doesn't affect the module.

For the transient analysis an air flow path in a module has been considered, since the air flow paths have symmetrical distance in between. It also avoids the necessity of high speed processors and reduces the time of simulation, by which the results can be considered for all the air passages in the module, so the domain considered for the analysis is shown in fig. 3.

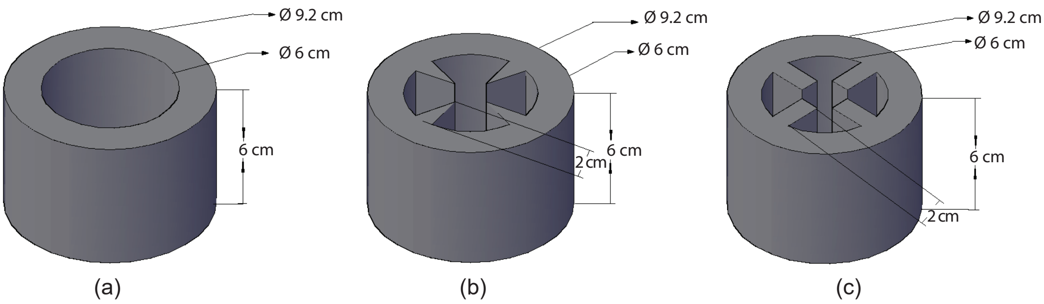

Figure 3. The 3-D single air passage in a module for transient analysis with different fins; (a) without fin; (b) triangular fin; (c) rectangular fin

The PCM used is glycerol which is filled in shell side of volume 0.0067 m^3 with mass of 8.5 kg, the properties of the PCM is shown in the table 2. For the computation analysis the following assumptions are:

- the fluid domain is considered to be incompressible as the change in density is negligi ble during the flow in the air passage,
- the density, specific heat and thermal conductivity are considered to be constant for the heat transfer fluid, and
- the density and thermal conductivity and specific heat of the PCM are considered to be constant for solid and liquid phase.

Experimentation

A modular heat exchanger is considered for the experimentation to determine the solidification time of the PCM. The dimensions of the module are same as considered for the numerical analysis. The cold air is passed through the air passage at air inlet velocity of 0.4 m/s, air inlet temperature of 280 ±1 K from a climate simulator and the PCM temperature of 306 K. The temperature inlet and outlet is measured with RTD's located evenly over the module and a 3-D ultrasonic anemometer is used to measure the air flow rate. The data from the RTD's and the anemometer is recorded by a data acquisition system. The experiment is continued till the full solidification of the PCM. The image of the module is shown in fig. 4 and the experimentation set-up is shown in fig. 5.

Table 2. Thermo-physical properties of the PCM

Properties	Glycerol
Specific heat *Cp*, [kJkg^{-1}K^{-1}]	2.377
Latent heat of fusion, [kJ/kg]	198.5
Freezing temperature, [K]	290
Thermal conductivity, [Wm^{-1}K^{-1}]	0.286
Density of the PCM, [kgm^{-3}]	1250
Dynamic viscosity, [Pa·s]	0.934

Figure 4. Experimental photo of a air flow module

Figure 5. The PCM stacked module and experimentation set-up of air conditioning and testing system

Numerical analysis

The PCM module domain and the air flow path domain with 2 air spacers are modeled and meshed in ANSYS software, in the preprocessor the boundary condition enabled are velocity inlet and outflow. The model is analyzed in the FLUENT software which employs finite volume method for solving mass eq. (1) is used, for momentum eqs. (2), (3), and (4), and for energy eq. (5) are used. SIMPLE algorithm is taken as the solver option by adopting fully implicit method for transient simulation. For the turbulent flow region in the tube flow region k-ε turbulence model is taken in to account. The equations for the turbulence kinetic energy k and its dissipation rate ε are (6) and (7).

$$\frac{\partial \rho}{\partial t} + \mathrm{div}(\rho \mathbf{u}) = 0 \tag{1}$$

$$\rho \frac{\mathrm{D}u}{\mathrm{D}t} = \frac{\partial(-p + \tau_{xx})}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} + S_{Mx} \tag{2}$$

$$\rho \frac{\mathrm{D}v}{\mathrm{D}t} = \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial(-p + \tau_{yy})}{\partial y} + \frac{\partial \tau_{zy}}{\partial z} + S_{My} \tag{3}$$

$$\rho\frac{\mathrm{D}w}{\mathrm{D}t}=\frac{\partial\tau_{xz}}{\partial x}+\frac{\partial\tau_{yz}}{\partial y}+\frac{\partial(-p+\tau_{zz})}{\partial z}+S_{Mz} \tag{4}$$

$$\begin{aligned}\rho\frac{\mathrm{D}E}{\mathrm{D}t}=&-\mathrm{div}(p\mathbf{u})+\left[\frac{\partial(u\tau_{xx})}{\partial x}+\frac{\partial(u\tau_{yx})}{\partial y}+\frac{\partial(u\tau_{zx})}{\partial z}+\frac{\partial(v\tau_{xy})}{\partial x}+\right.\\&\left.+\frac{\partial(v\tau_{yy})}{\partial y}+\frac{\partial(v\tau_{zy})}{\partial z}+\frac{\partial(w\tau_{xz})}{\partial x}+\frac{\partial(w\tau_{yz})}{\partial y}+\frac{\partial(u\tau_{zz})}{\partial z}\right]+\\&+\mathrm{div}(k\ \mathrm{grad}\ T)+S_E\end{aligned} \tag{5}$$

$$\frac{\mathrm{D}k}{\mathrm{D}t}=\frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\frac{\mu_t}{\sigma_k}+\mu\right)\frac{\partial k}{\partial x_j}\right]+\frac{\mu_t}{\rho}\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j}-2\upsilon\left(\frac{\partial k^{1/2}}{\partial x_j}\right)^2-\varepsilon \tag{6}$$

$$\begin{aligned}\frac{\mathrm{D}\varepsilon}{\mathrm{D}t}=&\frac{1}{\rho}\frac{\partial}{\partial x_j}\left[\left(\frac{\mu_t}{\sigma_\varepsilon}+\mu\right)\frac{\partial\varepsilon}{\partial x_j}\right]+\frac{C_{1\varepsilon}\mu_t}{\rho}\frac{\varepsilon}{k}\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)\frac{\partial U_i}{\partial x_j}-\\&-C_{2\varepsilon}\frac{\varepsilon^2}{k}-2\frac{\upsilon\mu_t}{\rho}\left(\frac{\partial^2U_i}{\partial x_j\partial x_i}\right)^2\end{aligned} \tag{7}$$

PCM domain is considered to be static. During the trial of simulation inlet velocity and inlet temperature are given as input boundary conditions and the outflow weighing condition of 1 is given for exit boundary condition. The convergence criterion of $1\cdot10^{-3}$ is used for mass and momentum; $1\cdot10^{-6}$ is used for energy residuals. A transient simulation study is carried out to determine PCM solidification time for module with various fins and a module with lower PCM solidification time is considered for steady state analysis for the study of pressure drop, flow and temperature variation.

Results and discussion

Experimental results

During experimentation the air from the climate simulator is allowed to attain stability by maintaining constant heat load. Then the air at 280 ± 1 K is passed through the single air passage module with rectangular fin which contains PCM at 306 K. The results from the numerical analysis have shown a maximum deviation of 9% *i. e.*, in real time it takes around 7.3 hours to solidify. The experiment reveals that the numerical analysis gives fairly acceptable results, so this can be used to predict the thermal and flow characteristics of the modular heat exchanger and the heat transfer fluid flowing in it. This averts the expenditure involved in the experimental study of heat exchanger systems. The full solidification is determined by the isothermal process during solification. The solidification and melting curves are shown in fig. 6 and fig. 7. It shows that the rectangular fins has the better performce compared to the other configurations.

Transient analysis

Transient simulation study has been conducted to the single air passage in the modular heat exchanger. The domain considered has PCM around the air passage for the diameter of 9.2 cm. This is because of the equal distance between the air flow paths with different shape

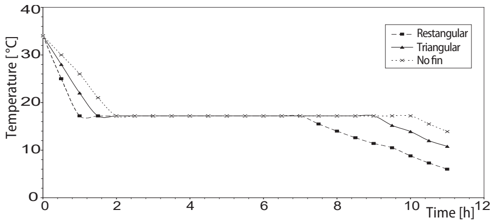

Figure 6. Solidification curves of phase change material in different fin module

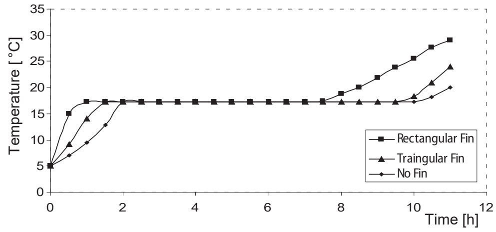

Figure 7. Melting curves of phase change material in different fin module

of fins. The given inlet conditions are air inlet 0.4 m/s, air inlet temperature 280 K and initial PCM temperature of 306 K. The analysis results for different fin profiles are shown in the fig. 8.

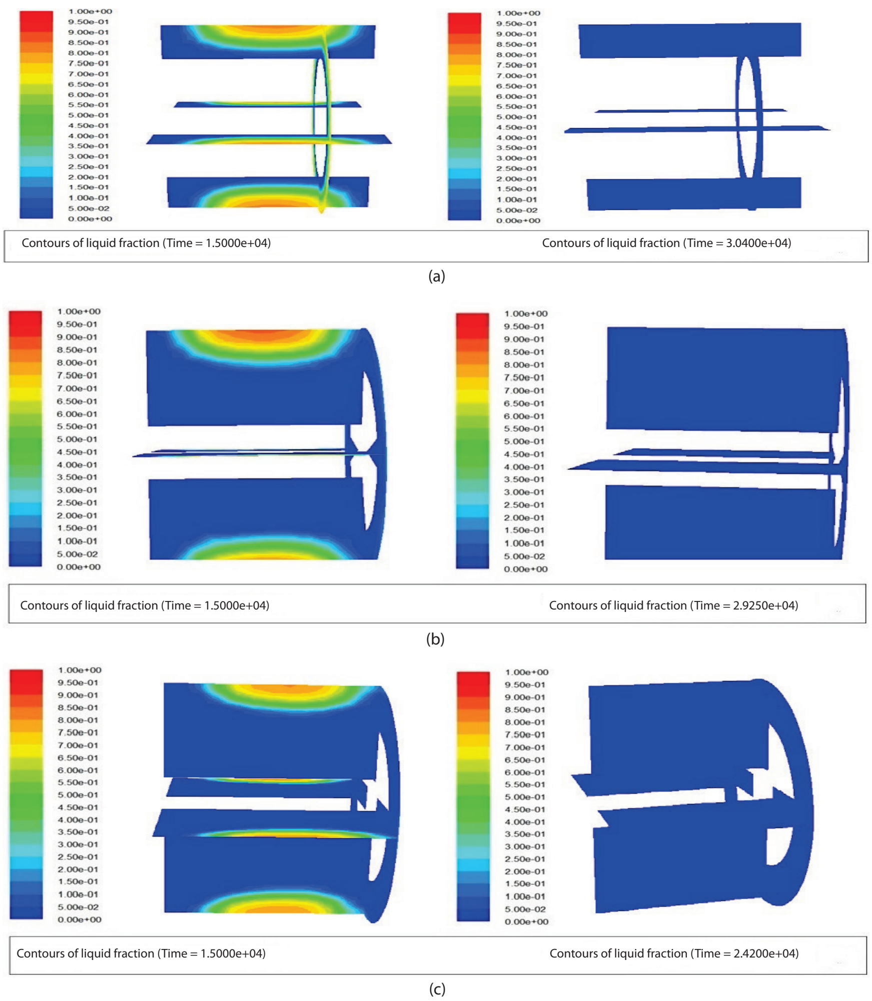

Figure 8. Liquid fraction contour for module, (a) without fin; (b) with triangular fin; (c) with rectangular fin, after 15000 seconds and full PCM solidification time, respectively

Figure 8 shows the liquid fraction contour reveals that module without fin solidifies at 30,400 seconds (8.4 hours), module with triangular fin solidifies at 29,250 seconds (8.1 hours) and the module with rectangular fin solidifies at 24,200 seconds (6.7 hours), by which it is identified that the module with rectangular fin possess higher heat transfer than the other two. This phenomenon is due to more surface area of PCM module exposed for heat transfer to the air, so this phenomenon is considered for the study of pressure drop characteristics by steady state analysis. Time variation of PCM solidification process for different profiles of modular heat exchanger at 0.4 m/s is shown in fig. 9 and time variation of PCM solidification process for different profiles of modular heat exchanger at 1.6 m/s is shown in fig. 10

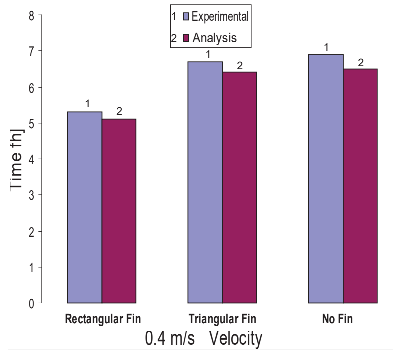

Figure 9. Time variation for 0.4 m/s

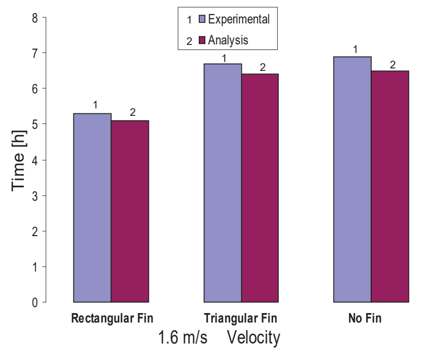

Figure 10. Time variation for 1.6 m/s

Steady-state analysis

Steady-state analysis for rectangular fin is carried out for a modular heat exchanger with two air spacers in order to determine the pressure drop, flow and thermal characteristics of the heat transfer fluid across the module and the air spacers for various frontal velocities of 0.4 to 1.6 m/s. Although the analysis is carried out for a range of frontal velocities, the results shown are only for the air inlet velocity of 0.4 and 1.6 m/s with the considered initial condition.

In chapter 5.3 module with rectangular fin's velocity ratio is asserted. The average total pressure drop in the first case is around 1.5 Pa and for velocity of 1.6 m/s is around 24.50 Pa. Figure 11 shows velocity contours and reveals that as the air flow path has varying cross sections by placing fins; air flow becomes more turbulent, so the velocity of the air increases when it passes through the tubes of the module. At the outlet of the module it is noted that the air velocity is much higher than the inlet velocity for 0.4 and 1.6 m/s and it can be observed that at the outlet non uniform mixing of air is seen for both the frontal velocities. The non-uniform mixing of air is less for 0.4 m/s and high for 1.6 m/s.

Figure 12 shows the pressure distribution contour of the air in the module and the air inlet velocity of 0.4 m/s and 1.6 m/s, frontal velocity is proportional to pressure drop. When the air enters and passes over the module there is increase in pressure in the non-flow region this is due to the conversion of kinetic energy to pressure energy. The average total pressure drop in the first case is around 1.5 Pa and for velocity of 1.6 m/s is around 24.54 Pa. The variation of pressure drop in accordance with various air inlet velocities for Triangular fin as shown in fig.13 and the variation of pressure drop in accordance with various air inlet velocities for Rectangular fin as shown in fig.14

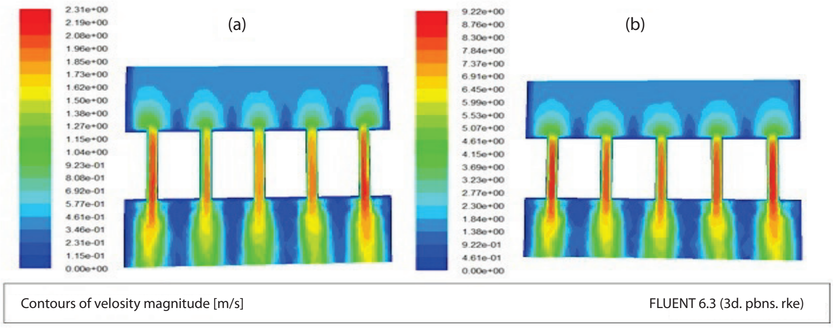

Figure 11. Velocity contour of the air domain across the module for frontal velocities; (a) 0.4 m/s, (b) 1.6 m/s

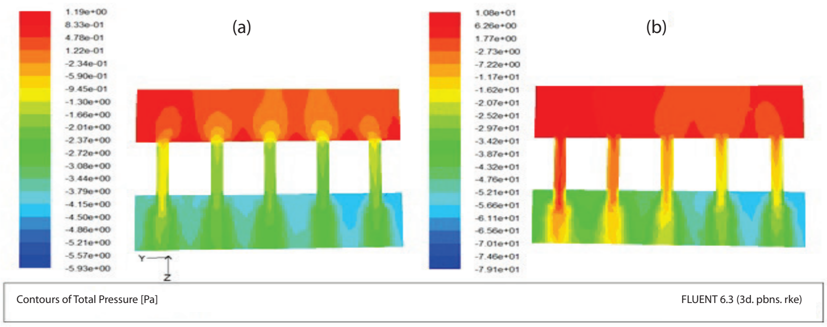

Figure 12. Pressure contours of the air domain for frontal velocities; (a) 0.4 m/s, (b) 1.6 m/s.

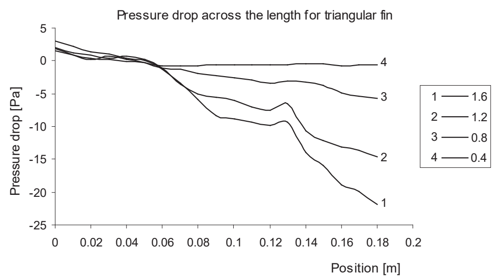

Figure 13. Pressure drop graph for various frontal velocities for triangular fin

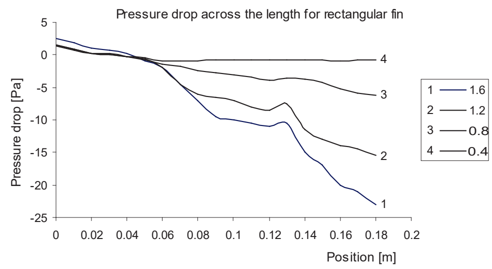

Figure 14. Pressure drop graph for various frontal velocities for rectangular fin

Thermal analysis

Chapter 5.4 shows the cooling of PCM module when air is passed through different velocities. If the air flow is laminar cooling is uniform. Figure15 shows the surface heat flux contour for the frontal velocity of 0.4 and 1.6 m/s. It is observed that at the lower as 0.4 m/s velocity, retention time is high, so the air distribution is uniform. Hence the heat flux is uniformly distributed to the wall surface of the module. But at higher velocity of 1.6 m/s the air retention

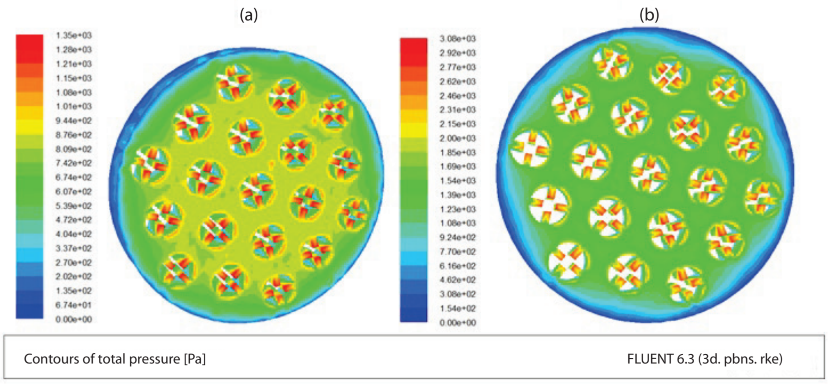

Figure 15. Heat flux contours of the air domain across the module for frontal velocities (a) 0.4 m/s (b) 1.6 m/s

time decreases and so the heat flux in the inlet wall surface is lower and heat flux in the tube side is higher comparing to the bottom and top surface of the module.The Experimintal results and comparison of solidification time of numerous fins for different inlet velocities is given below. Comparison of solidification time of different fin with different velocity is shown in fig. 16 and the results are tabulated in tab. 3.

Table 3. Solidification time of different fin with various velocity

Fin	Velocity [ms^{-1}]			
	0.4 m/s	0.8 m/s	1.2 m/s	1.6 m/s
Rectangular Fin	6.7	6.3	5.7	4.9
Traingular Fin	8.1	7.7	7	5.9
No Fin	8.4	7.9	7.1	6

Conclusions

In the present work the benefit of modular heat exchanger with rectangular fin is highlighted. The analytical results of the modular heat exchanger with rectangular fin are compared with the experimental values and a good agreement is identified. A total of 3 different patterns of modular heat exchangers are analyzed. From the analysis it is seen that the module with rectangular fin solidification time reduces by 17.2% than the other modules, which infers that the module with rectangular fin has the maximum heat transfer. When the Experimental results are compared with the analytical results there is no much deviation is found but the time taken to solidy the PCM experimentally is little bit higher. The introduction of fin causes the melting process to be quicker, so after some time the PCM will not be able to absorb the heat. In module without fin there will be some period where PCM cannot absorb heat. It depends on the capacity of the pcm.If the availability of heat is more then capacity of PCM may be increased. Mass of PCM and air flow rate plays vital role in obtaining the comfort temperatures to certain period of time during the day operation but during night time operation air flow rate andnight ambient

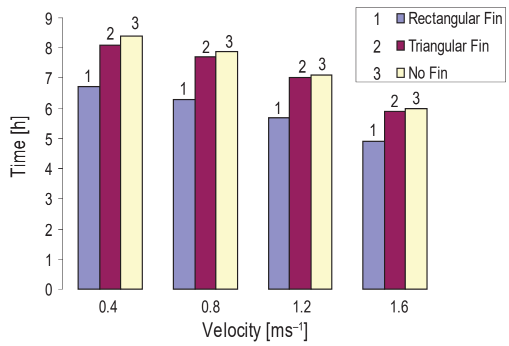

Figure 16. Comparison of solidification time of different fin with various velocity

temperature plays vital role and these will decide how much mass of PCM can be solidified in limited summer time of eight hours. The pressure drop across the PCM module increases as the frontal velocity increases, but the pressure drop is negligible since the average outlet pressure drop is around 39.5 Pa. The pressure drop characteristics are studied in comparison with velocity and the whole cooling is not only because of the pressure drop. One module won't cool a building but it can cool a prototype (fig. 5), if it is to be used for larger building then more PCM with more module should be used. The air space in between the module increases the air and PCM module contact time but the velocity increases the effect of air gap in between the module reduces. The velocity of air increases when it passes out the module, so the adjacent modules frontal velocity is higher, this phenomena may be reduced, when the air flow paths in the adjacent modules are not in the straight line.

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Nomenclature

E – energy, [J]
k – turbulence kinetic energy, [m^2s^{-2}]
p – pressure, [Pa]
S_E – source term, [–]
u – velocity, [ms^{-1}]

Greek symbols

ε – turbulent dissipation rate, [m^2s^{-3}]
μ_t – turbulent viscosity, [$kgm^{-1}s^{-1}$]
ρ – density, [kgm^{-3}]
σ_ε – model constant, [–]
σ_k – model constant, [–]
τ_{xx} – normal stress, [Nm^{-2}]
ϕ – general constant, [–]

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