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Evaluating the possibility of utilizing the waste heat of hot water before the cooling towers based on Phase change material

Reza Bemani, Zahra Hashemi* , Maysam Gholampour

Department of Mechanical Engineering, Bushehr branch, Islamic Azad University, Bushehr, Iran.

ABSTRACT

Today, efficient energy use and minimizing waste are crucial. In power plants, hot water with high thermal energy enters a cooling tower, requiring significant water consumption for cooling. In the current research, the possibility of exploiting the thermal energy of hot water between the turbine and the cooling tower has been investigated using shell and tube heat exchangers where the shell is filled with phase change materials (PCM). This research investigates the temperature reduction of the hot water temperature entering the cooling tower and possibility of using the thermal energy of the hot water entering the cooling tower. Two approaches are included: a PCM-based heat exchanger for thermal storage and a heat exchanger to reduce hot water temperature without energy storage capability. Paraffin, a phase change material, is placed between the tube and shell in a heat exchanger, where hot water flows through the tube. The enthalpy-porosity model simulates phase change. Outlet hot water temperature, volume fraction changes, and paraffin temperature in finned and non-finned tubes are compared in different conditions. Also, fines are installed on the shell side to investigate the heat transfer enhancement. Results suggest that with an insulated shell, a thermal storage system should use a heat sink (cold water) to absorb released heat during paraffin solidification. The insertion of fins increases paraffin phase change rate to about 100% which reduces required time to be completed melt from 10 to 4 hours, thus enhancing the heat transfer rate significantly. Results indicate that in a colder climate with temperatures significantly below the melting point, non-insulated heat exchangers can maintain steady performance which can cause a steady two-phase regime of PCM (steady liquid volume fraction about 0.5) and keep the shell temperature constant over time. Finally, it could reduce hot water temperature that enters cooling tower about 6 degrees.

KEYWORDS: *Phase Change Material (PCM), Enthalpy-Porosity modeling, Heat Exchanger, Water wasting, Finite Volume.*

* *Corresponding author. Address: Department of Mechanical Engineering, Bushehr branch, Islamic Azad University, Alishahr, Bushehr, Iran. Tel.: +98 7733682305.*

E-mail address: zahra.hashemi@iau.ac.ir

z.hashemi986@gmail.com

INTRODUCTION

Due to water shortages and limited water resources, special attention has been paid to reducing water consumption in recent years. Meanwhile, power plants are one of the main consumers of water resources, most of which are used for cooling systems[1]. Evaporative cooling using a cooling tower is an efficient method among the various processes. In addition, there is a great potential for increasing the efficiency of cooling towers in order to save energy and reduce water consumption [2].

In cooling towers, two factors, improving thermal properties and reducing water loss, are important for researchers. Improving the thermal characteristics increases the efficiency and further cooling of the water. On the other hand, when heat and mass are exchanged between the water and air, the amount of evaporated water is significant, which can be reduced by various methods.

One of the methods to improve thermal properties is using nanoparticles because these particles can increase heat transfer [3–6]. Nanoparticle concentration, liquid inlet mass flow rate, air inlet mass flow rate, and relative humidity are the key parameters affecting cooling towers' water loss rate and thermal characteristics. Nanofluids can significantly improve dry and wet cooling tower performance by improving thermal properties and reducing water loss. The nanofluid is synthesized by dispersing the nanoparticles in a base fluid, and its thermal conductivity and viscosity increase relative to the base fluid. Using nanofluids improves the cooling rate and increases the cooling tower's efficiency and heat transfer coefficient. On the other hand, increasing the surface tension enhances the evaporation resistance and affects the amount of water loss.

Mousavi et al. [7] investigated the effects of carbon quantum nanoparticles (CQD)on increasing heat transfer in a wet cooling tower. The results showed that using Fe_3O_4-CQD nanofluid increases the efficiency of the cooling tower by 12% and reduces water consumption by 7.5%. Using Cu-CQD leads to a 25% increase in cooling tower efficiency and a 16% decrease in water consumption.

Kisaosui et al. [8] illustrated that using the alumina/water nanofluid, the cooling tower efficiency increased by 19% and the heat transfer coefficient by 20% compared to pure water.

Askari et al. [2] investigated the thermal performance of a cooling tower using multi-walled carbon nanotubes and graphene-nanoporous nanofluids. Their results showed that using nanofluid increases the thermal conductivity, efficiency of the cooling tower, and the cooling temperature range, and also reduces water consumption in using carbon nanotubes by 10% and in the use of graphene nanoporous by 19%.

Another method that researchers have recently considered increasing the efficiency of cooling towers is using phase change materials (PCMs) or heat pipes [9–11]. Liquid-solid phase change materials are used for short-term and long-term energy systems due to their high energy density, low-density changes and relatively low price [12,13]. Fluids containing PCM have a higher sensible specific heat than single-phase fluids and are therefore suitable for transferring high amounts of heat without a significant increase in temperature.

One of the applications of PCMs in field of the thermal storage is in the residential buildings, industrial silos, and factories [14]. When we look at the role of recycling and optimizing energy consumption in energy management, their importance is multiplied [12]. These materials save about 29% and 19% of energy in the eastern-western and northern and southern, respectively [15]. The use of these materials in the external wall of an office leads to 12.8% energy savings [16]. The utilization of PCMs in heat management systems of batteries and electrical circuits has been considered in recent years [17]. Utilizing in cooling electrical circuits of a smartphone [18] temperature control of Electric chips [19] and improving the thermal performance of lithium-ion batteries of vehicles [20] are among the studies conducted in this field.

Some Researchers studied melting and solidification of phase-change material in the tube and shell heat exchangers

[21–23]. The results indicated that by increasing the heat transfer fluid temperature from 70 to 80 ℃, the time of melting decreases 37% [24].

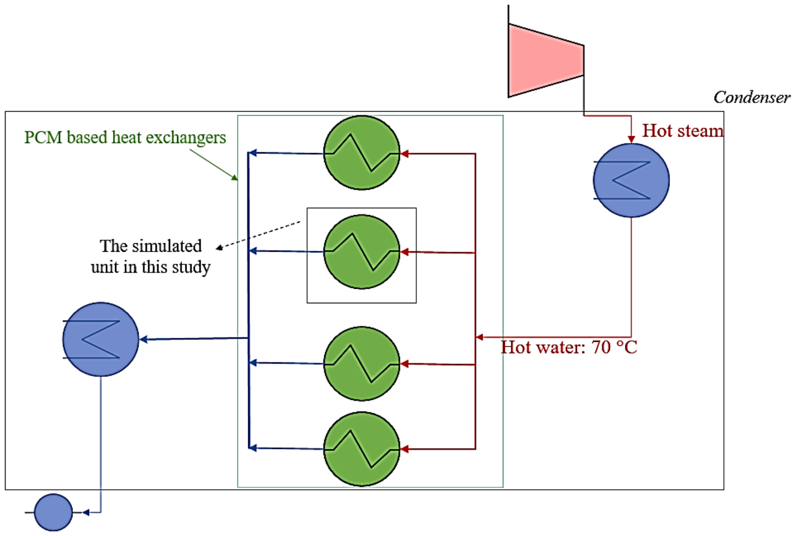
To reduce water consumption in cooling systems, Zhang et al [25] developed an air-cooling heat exchanger (ACC) using encapsulated phase change materials (EPCM). This EPCM heat exchanger improves the heat transfer coefficient and efficiency of the cooling tower and on the other hand leads to reduced pressure drop and system costs. Their results showed that the proposed EPCM heat exchanger had 37% and 42% lower water consumption compared to dry and wet cooling towers, respectively.

Many studies have been conducted on the use of PCM in cooling systems of photovoltaic panels, and the results of studies show that the use of PCM leads to a decrease in panel temperature and also increases its output power [26–29]. Abdollahi and Rahimi [30], In an experimental study, investigated the effects of nanofluids on the cooling process of a hybrid cooling tower system based on photovoltaics and phase change materials (PV/PCM). They used low-concentration of boehmite nano-powders in water as a nanofluid. PCM has been solidified during the night and early morning hours when the air temperature is lower than the freezing point of PCM. The results show that this combined method is very effective in reducing the panel temperature and leads to an increase in PV output power.

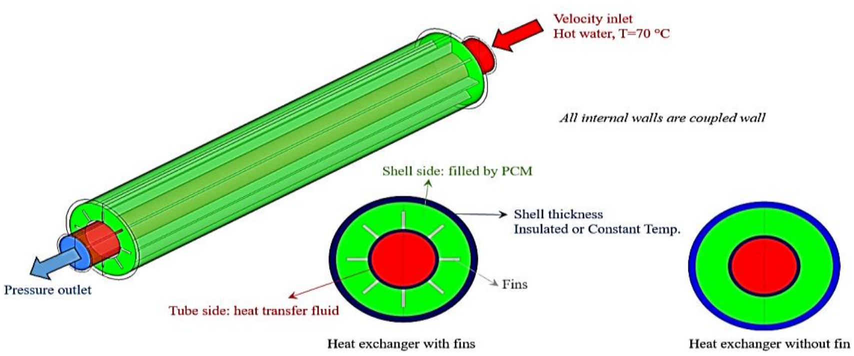
A review of previous studies shows that the use of PCMs in the efficiency of heat exchangers and their role in faster cooling of the operating fluid has yet to be studied in detail. In the literature, the use of the PCM-based heat exchanger to utilize the wasted heat of the hot water before going into the cooling tower still needs to be addressed, and using the thermal storage capacity of the PCM requires more effort to clarify that it could be a beneficial option or not. Also, the effects of the climate condition on the PCM-based heat exchangers should be answered. To cover the mentioned gap, in the present paper, the cooling operation of a working fluid in a heat exchanger is modeled by simulation of the unsteady, 3D, turbulent flow field, and heat transfer and thermal storage capability is investigated. The average temperature of the outlet hot water, the volume fraction changes, and the average temperature of paraffin are presented.

PROBLEM STATEMENT

Figure (1.a) illustrates the schematic of the studied problem. Outlet steam from the turbine, after passing through the primary condenser, enters the heat exchangers as a liquid phase (water at a temperature of 70). In the present investigation, the shell side of the heat exchanger is filled with paraffin as PCM to utilize the latent heat during the melting process. The thermophysical properties are presented in Table 1. The PCM covers the entire lateral surface of the inlet hot fluid tube and is in the solid phase at the initial moment. Melting process of PCM accrue with time as hot water passes through the tube and shell heat exchanger. Fins on the side surface of the tube can increase the conduction heat transfer, so in this research, fins are installed symmetrically on the tube, and the heat transfer is compared between two cases with and without fins. Fins are selected aluminum due to the high thermal conductivity 273 W/m-k. The shell wall is assumed to be either temperature-constant or insulated. All water properties are considered constant, and its density variation are ignored. Water enters the tube at a steady rate, and the atmospheric pressure is assumed as a boundary condition at the tube outlet. Variations of the thermophysical properties of paraffin are supposed to be constant except for density and viscosity of the liquid phase of the melted paraffine. The paraffin density is considered to change linearly with temperature due to the applied Boussinesq hypothesis. So, the effects of buoyancy on PCM should be considered on the gravity direction.



(a)



(b)

Fig. 1: Problem statement a) schematic b) geometry of the studied problem

Table 1 Thermophysical properties of paraffin [18]

property	value
Density(kg/m^3)	780
Latent heat (J/Kg)	170000
Melting/freezing temperatures (K)	318/324
Thermal conductivity coefficient (W/K)	0.2
Specific heat capacity $\left(\frac{J}{kg.K}\right)$	2000
Viscosity in molten state $\left(\frac{N}{m.s}\right)$	$\mu = Ae^{B\,T}$ A = 0.819, B = − 1.546 × 10^{-2}, 326 K ≤ T ≤ 353 K

GOVERNING EQUATIONS

The present problem contains two non-mixing operating fluids so that hot water is inside the tube and paraffin is in the shell as PCM, the momentum and continuity equations are solved separately for the two parts. On the other hand, the coupling of energy equations is performed due to the heat transfer from the wall of the hot tube to the paraffin in the shell.

- ***Equations governing the water flow inside the tube***

Assuming incompressible, 3D, turbulent, unsteady and Newtonian, the equations of continuity, momentum and energy in the pipe will be as follows:

-Continuity equation

$$\frac{\partial}{\partial x_i}(u_i) = 0 \tag{1}$$

-Momentum equation

$$\frac{1}{\rho}\frac{\partial}{\partial t}\left(\underline{u_i}\right) + u_j\frac{\partial \underline{u_i}}{\partial x_j} + \frac{1}{\rho}\frac{\partial \underline{P}}{\partial x} = \nu\left(\frac{\partial^2 \underline{u_i}}{\partial x_j \partial x_j}\right) - \frac{\partial(\underline{\acute{u}_i\acute{v}_i})}{\partial x_j} \tag{2}$$

where ν and ρ are kinematic viscosity and water density, respectively, and RANS method has been used to model the turbulent flow of water inside the tube. It should be noted that the application of buoyancy body force in this range has been abandoned due to the strong dominance of inertial forces over buoyancy forces (Richardson numbers are very small).

-Energy equation

$$\rho c_p (u_i \frac{\partial T}{\partial x_i}) = k(\frac{\partial^2 T}{\partial x_{i,i}}) \tag{3}$$

where c_p and k are special heat capacity and water conductivity, respectively. In the present paper, the thermophysical properties of water are assumed to be independent of temperature.

Equations governing the PCM

Voller et al. [31] proposed the enthalpy-porosity approach for modeling phase change within PCM-based TES systems. This approach doesn't explicitly track the melting interface [32]. Instead, it defines the liquid fraction, representing the portion of each cell in the PCM domain that is in a liquid state, using enthalpy balance calculations at each iteration of the simulation. The region where solid and liquid phases coexist (the mushy zone) is treated as a porous zone with porosity equal to the liquid fraction, incorporating momentum sink terms into the momentum equations. The motion of liquid PCM during melting is assumed to be laminar, unsteady, and incompressible, while volume changes during melting from solid to liquid are disregarded. Additionally, the motion of solid PCM during melting is neglected.

All thermophysical properties of PCM are assumed to be constant except density and viscosity. Density variations with respect to temperature are assumed to be linear with the Boussinesq approximation, which leads to apply buoyancy force on the PCM domain after sufficient melting of the solid paraffin.

The governing equations of PCM (in liquid phase) considering the above assumptions are [33]:

-Continuity equation

$$\nabla . \vec{V} = 0 \tag{4}$$

-Momentum equation

$$\rho_p (\frac{\partial \vec{V}}{\partial t} + \vec{V}.\nabla.\vec{V}) = -\nabla P + \mu_p \nabla^2 \vec{V} + \rho_p \beta_p \vec{g}\left(T - T_{ref}\right) + S \tag{5}$$

In the above equation, the third term of RHS is the buoyancy force created in the liquid after sufficient melting of the solid paraffin, this term will be dominant in the heat transfer process. On the other hand, changes in the viscosity of liquid paraffin are assumed to be a function of temperature which is a curved fit equation based on the experimental results of Hosseini *et al.* **[18]** study:

$$\mu_p = A e^{B\,T} \tag{6}$$

The momentum equation (Eq. 5) includes a source term $\vec{S}$, which arises from the decreased porosity within the mushy zone and is expressed as follows:

$$S = \frac{(1-\beta)^2}{\beta^3 + \varepsilon} A_{mush}\vec{V} \tag{7}$$

where β is the volume fraction of liquid, ε is a small number to avoid devising zero and A_{mush} is a constant parameter (10^{-4}).

- Energy equation

$$\frac{\partial(\rho H)}{\partial t} + \nabla.(\rho\vec{v}H) = \nabla.(k\nabla T) \tag{8}$$

H is the enthalpy of PCM and equals to the sum of sensible enthalpy (h) and latent heat (Δ H):

$$H = h + \Delta H \tag{9}$$

Sensible enthalpy is calculated as:

$$h = h_{ref} + \int_{T_{ref}}^{T} C_P \, dT \tag{10}$$

where h_{ref} is the reference enthalpy, T_{ref} denotes the reference temperature and C_P is the specific heat. The latent heat is estimated by $\Delta H = \beta L$, where L is latent heat and β is liquid volume fraction in PCM domain, which can vary from 0 (for solid) to 1 (for liquid) and is defined as follows:

$$\beta = \begin{cases} 0 & if\ \ T < T_{solidus} \\ \dfrac{T - T_{solidus}}{T_{liquidus} - T_{solidus}} & if\ \ T_{liquidus} < T < T_{solidus} \\ 1 & if\ T > T_{liquidus} \end{cases} \tag{11}$$

-Boundary and initial conditions:

In the first instance, the whole of PCMs is solid, water and heat exchanger are assumed to be at 25° C. Therefore:
At t = 0: u = v = w = 0 & T = 25ºC & PCM in solid state.

-Pipe boundaries:
At the inlet: u = V_0 and T = T_0, p = p_0
At the outlet: zero temperature gradient,
On the wall: u = v = w = 0, $k_w \nabla T.\vec{n} = k_p \nabla T.\vec{n}$

-Shell boundaries:
On the wall: u = v = w = 0
At the pipe-shell interface: $k_p \nabla T.\vec{n} = k_w \nabla T.\vec{n}$
At the shell outer wall: T=25ºC or $\nabla T.\vec{n}$=0

NUMERICAL SOLUTION METHOD, GRID STUDY AND VERIFICATION

Finite volume method is employed to solve the governing equations. The pressure-velocity coupling utilizes the SIMPLEC algorithm, while pressure correction equations employ the PRESTO! scheme. Second-order upwind schemes are applied to the momentum and energy equations. Under-relaxation factors for pressure, velocity, energy, and liquid fraction are specified as 0.3, 0.2, 1.0, and 0.9, respectively. Convergence criteria are set at 10^{-5} for the continuity equation and 10^{-7} for the energy equation.

Mesh study

Proper meshing with sufficient elements is essential in the numerical solution of a physical phenomenon to achieve the necessary accuracy in calculations. On the other hand, with increasing the number of the computational grids, the computational volume and consequently, the duration of solving the governing equations increases. Figure 2 shows the changes in PCM temperature over time with a different number of meshes. As shown in this figure, by increasing the number of solution domain meshes from 635000 to 1012222, the temperature changes are very small (less than 2% in the largest difference) and the results are accurate enough. Therefore, in the continuation of this work, all the results are presented by using 635000 nodes. It should be noted that in order to obtain the results of this figure, the following assumptions have been considered.

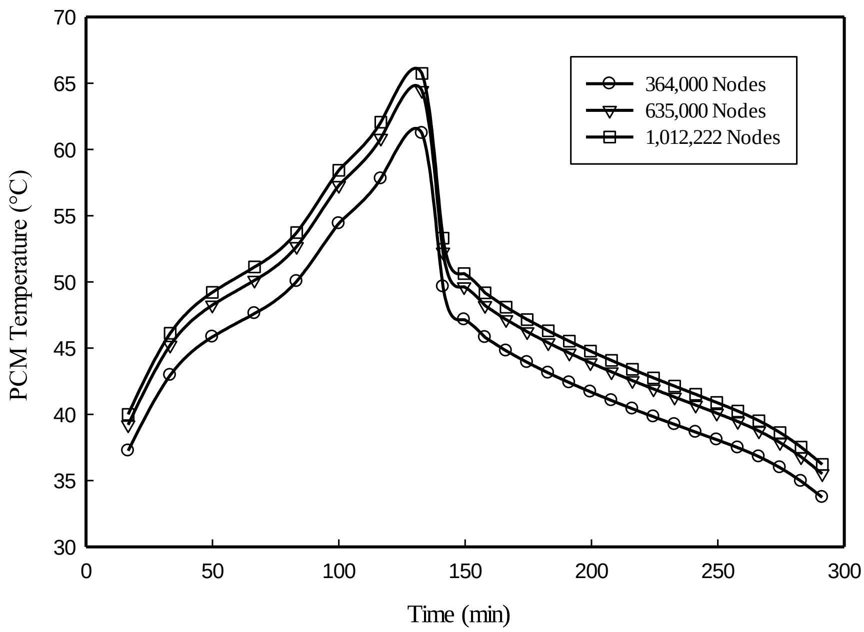


Fig. 2: Mesh independency

In the present study, the numerical solution method has been used to simulate the flow field and heat transfer, so it is necessary to evaluate the accuracy of the obtained numerical results to determine the error rate and evaluate the correctness of the numerical solution process. The most reliable results are those obtained from the laboratory, which have been published in reliable sources. The results presented in the study of Hosseini et al. [24] have been used to validate the obtained numerical results. By conducting experiments as well as numerical simulations, they investigated the behavior of phase-change materials that fill the space between two tubes (the inner tube containing hot fluid) and the outer tube. Paraffin is used as PCM in their study.

The obtained mean temperature variation of PCMs in terms of time has been compared with that of Hosseini et al. [24] in Figure 3. Comparison of the present results with the mentioned experimental results indicates that the selected numerical solution process, type of meshing and number of elements are well compatible with experimental results and the maximum error between the present numerical and experimental results is less than 8%.

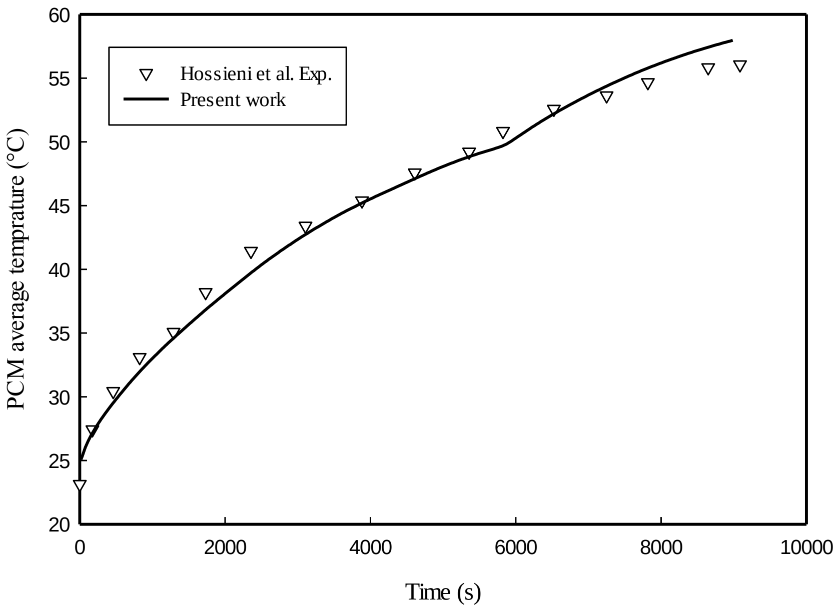


Fig. 3: Validation of used numerical solution method with Hosseini et al. [24]

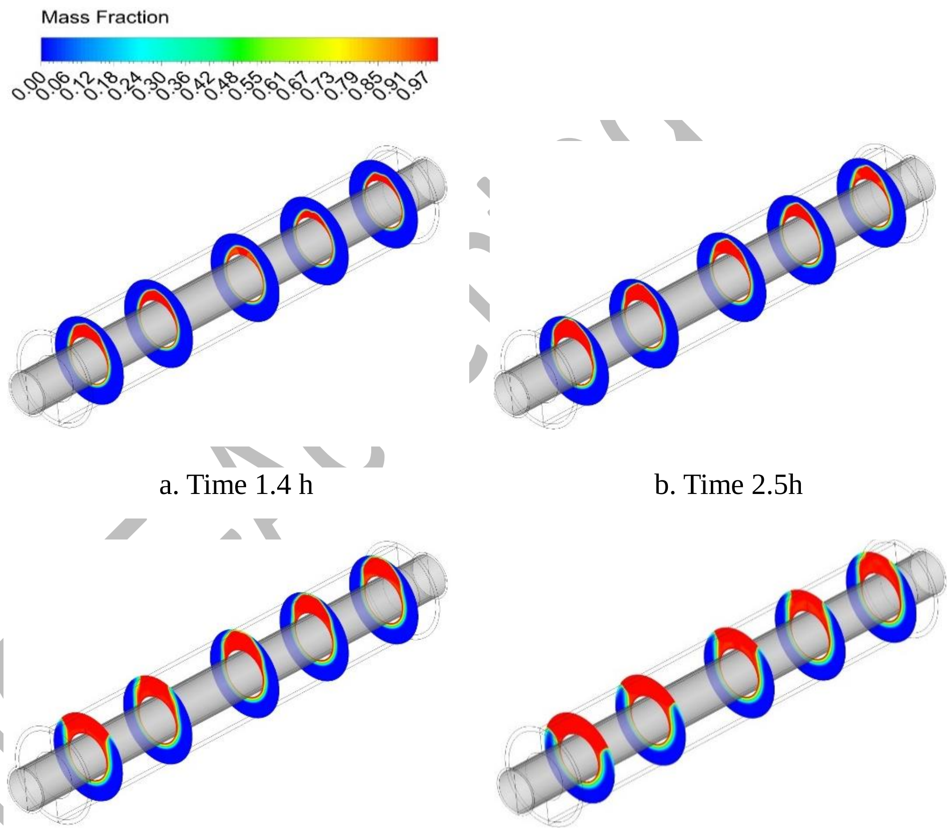
RESULTS

The present study demonstrates the heat transfer solution of a tube and shell heat exchanger containing PCM, which is used as part of a power plant. PCM is located in the space between the hot tube and the shell. PCM melts if it can absorb enough thermal energy and can solidify if it can release the absorbed heat. So, the performance of the heat exchanger, which is filled by PCM, depends on the PCM behavior. The effect of the attached fines on the tube is also investigated. In the all cases, the diameter of the tube and shell is 20 and 10 cm, respectively. In this section, firstly, the thermal storage capability in the insulated shell heat exchanger has been examined, then the fast-cooling capability of the heat exchanger is investigated.

Thermal storage case and the effect of fins in the PCM

As mentioned, in the case of thermal storage, we seek to analyze the possibility of using the absorbed heat from the hot water for heating and preparing hot water where it is needed. This section presents the contours of the volume fraction of the liquid phase and temperature to show the physics of the flow field in thermal storage heat exchangers (insulated shell). The effect of using fins attached to the tube on the shell side on the heat transfer rate is investigated. A fin can cause hydrothermal changes in the PCM domain, thus affecting the thermal performance

of the heat exchanger. Figure 4 shows the development of the fluid region in the PCM section over time. These results are obtained by assuming the adiabatic wall of the shell. AS seen, the solid PCM changes into a liquid phase after 1.4 hours. After 1.4 hours, it can be seen that the phase change has occurred in regions near the tube. This phase change occurs at all angles around the pipe but is not uniform. Figure 4 shows that the liquid volume fractions at the beginning and end of the tube are different, and the phase changing rate is lower on the inlet side of the tube, because there is no expansion of the thermal and hydraulic boundary layer at the beginning of the pipe. The hot fluid increases the pipe wall's temperature at the pipe's end. As a result, heat is transferred to the PCM layer adjacent to the pipe, which changes to a liquid phase. It should be mentioned that in all sections shown (from the beginning to the end of the hot tube), solid paraffin is completely liquefied in the vicinity of the tube.
In contrast, heat is transferred by natural convection in the opposite direction of gravity in the fluid phase of PCM by buoyancy force. The phase change distribution expands in this direction. Thus, most of the surface that has undergone phase change will be at the end of the pipe. After 5 hours, about a third of the total solid paraffin between the tube and shell becomes liquid.



a. Time 1.4 h

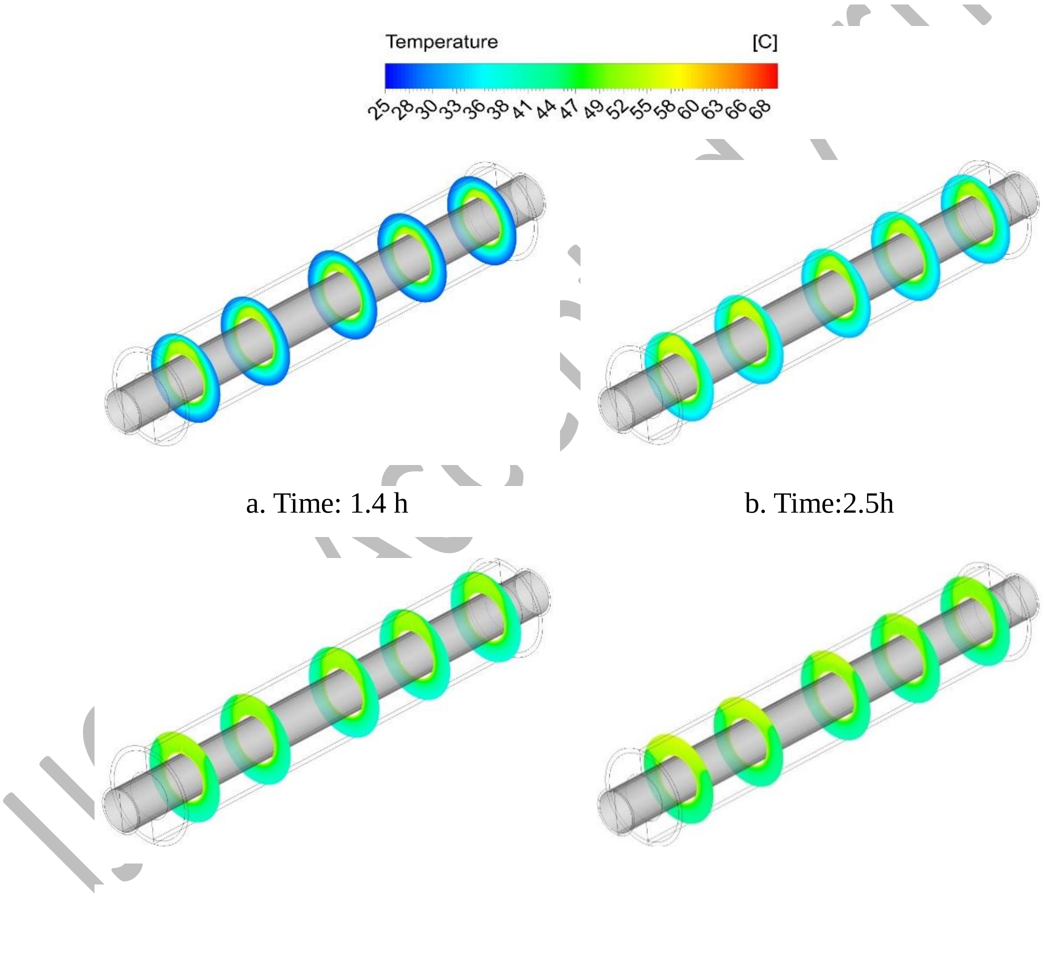
b. Time 2.5h

c. Time: 4.1 h

d. Time: 5 h

Fig. 4: Volume fraction of liquid phase of PCM with respect to time in a heat exchanger with adiabatic wall and without fin

Figure 5 illustrates the temperature variation in the case of the insulated shell. As seen, 1.4 hours after the beginning of the hot flow in the tube, isothermal surfaces with temperatures higher than paraffin melting temperature (45.09 ° C) are formed in the hot tube's vicinity. Still, areas with initial temperatures (25 °C) remained near the shell, due to insufficient thermal expansion and insufficient time for heat to transfer. It should be noted that the geometry of isothermal areas has a circular structure in 1.4 hours, which changes along the tube. Over time, the structure of isothermal lines changes for two reasons: heat transfer in solid paraffin and natural conduction in liquid paraffin. Therefore, small areas of paraffin remain at the initial temperature, and the temperature of the hot tube's upper surfaces is higher than others. Qualitatively, the most obvious effect of natural conduction heat transfer on the temperature distribution inside the paraffin occurs in the cross section at the end of the tube after 5 hours, when the temperature of the paraffin fluid near the shell is approximately equal to the temperature of the hot water fluid.

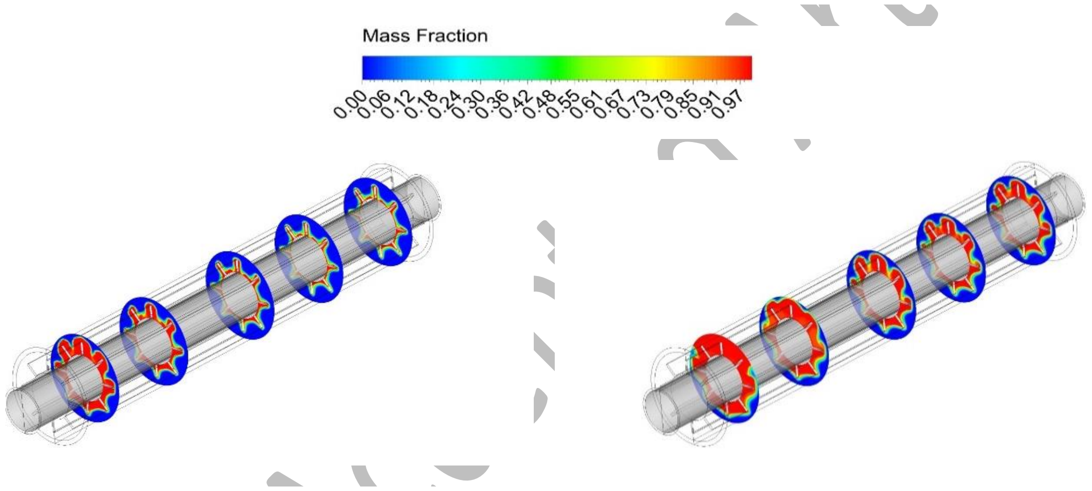


a. Time: 1.4 h b. Time:2.5h

c. Time: 4.1 h d. Time: 5 h

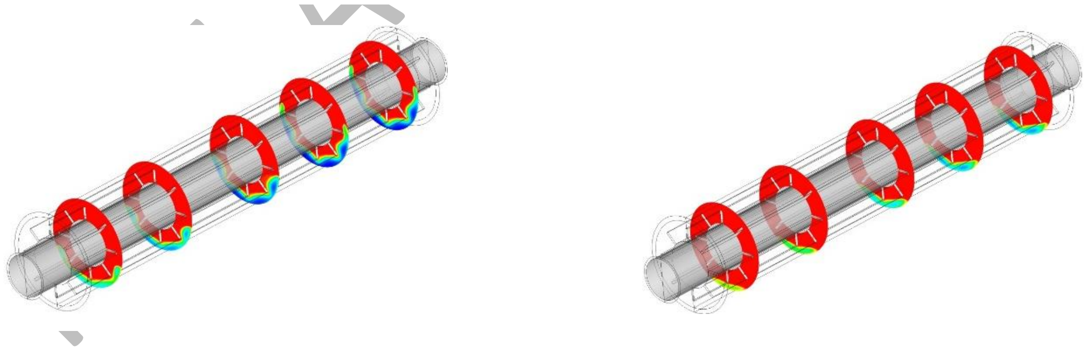
Fig. 5: Temperature distribution in PCM over time in a heat exchanger with adiabatic wall and without fin

Figure 6 shows the volume fraction of the liquid between the tube and the shell over time to investigate the effect of fin insertion on the PCM phase change pattern. The heat-conducting fins improve heat transfer from the wall of the hot tube to solid paraffin. Thus, in 1.4 hours, solid paraffin changes phase in the vicinity of the fins as well as the areas adjacent to the hot tube. Conduction heat transfer is predominant at the beginning of the tube due to the low volume of liquid paraffin. However, at the end of the tube, by increasing the liquid volume fraction, the effect of the buoyancy force increases, and the phase change partially expands in the opposite direction of gravity. This vertical expansion of the phase change intensifies over time so that only the bottom of the paraffin remains solid at 4.1 hours at the end of the tube. Finally, after 5 hours, almost all of the PCM is melted at the end of the tube. After that, there is no latent heat to absorb thermal energy. It should be noted that the reason for melting all paraffin, in this case, is the assumption of the adiabatic wall, which prevents the environmental effects on the process of heat transfer and phase change. This situation indicates that to have a steady performance in reducing the temperature of the hot water in the tube by PCM, we need a heat sink in the shell. The heat sink can absorb heat from melted PCM and reduce the PCM temperature below the liquid point. The heat sink can be cold water, which is required to be hot in different applications.



a. Time: 1.4 h

b. Time: 2.5 h

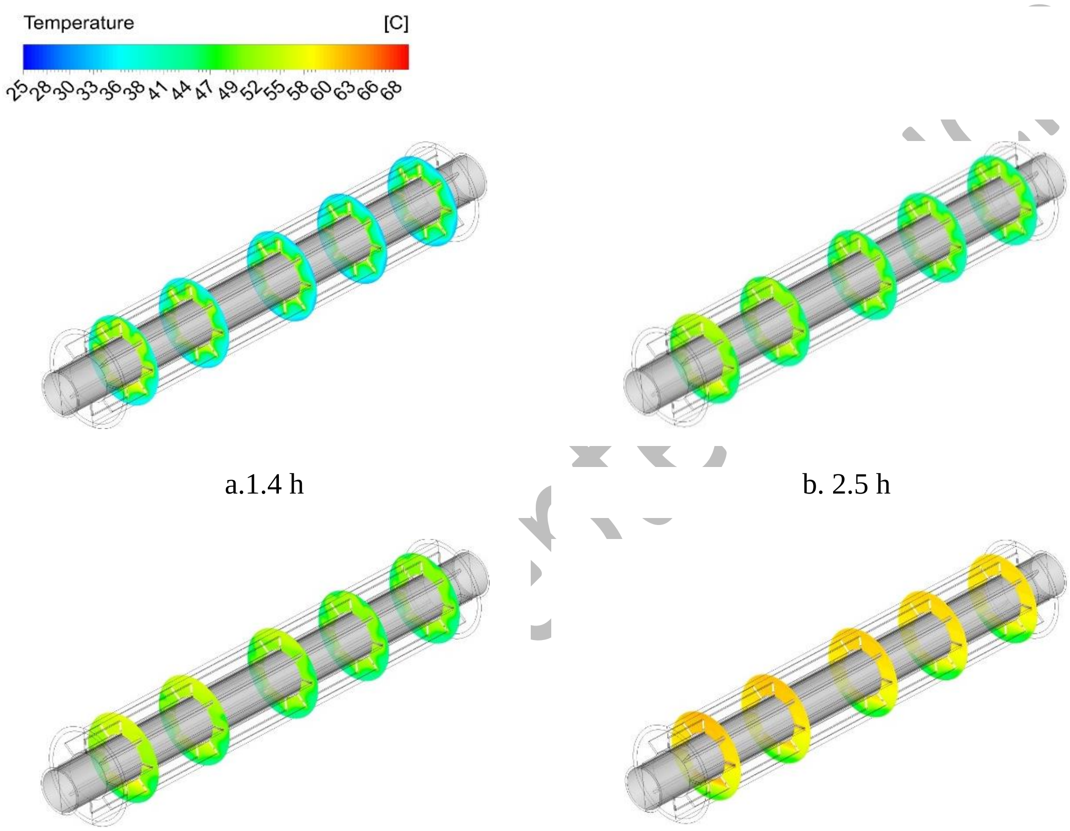


c. Time: 4.1 h

d. fins Time: 5 h

Fig. 6: Volume fraction of liquid phase of PCM with respect to time in a heat exchanger with adiabatic wall and fin

Figure 7 indicates the temperature distribution in the space between the shell and the hot tube as fins are attached to the tube on the shell side. Fins enhance the heat transfer rate from the tube to the existing paraffin because the high conductivity of the fin facilitates heat diffusion to the paraffin, increasing its temperature faster. In 1.4 hours, unlike in the non-finned state (Figure 5), the areas near the shell wall are heated from the beginning. Almost all paraffin elements have temperatures significantly higher than the initial temperature (25 ℃). The temperature in the paraffin range increases over time and by the intensification of heat transfer by natural convection. The liquid paraffin at the upper parts of the tube has a temperature of about 60 ℃ after 5 hours. Only a small part of the solid paraffin at the bottom of the tube has a temperature lower than the melting temperature. It should be noted that these results are presented by assuming the adiabatic shell wall. This assumption causes zero temperature gradients on the shell wall and prevents the environmental effects on paraffin.



a.1.4 h

b. 2.5 h

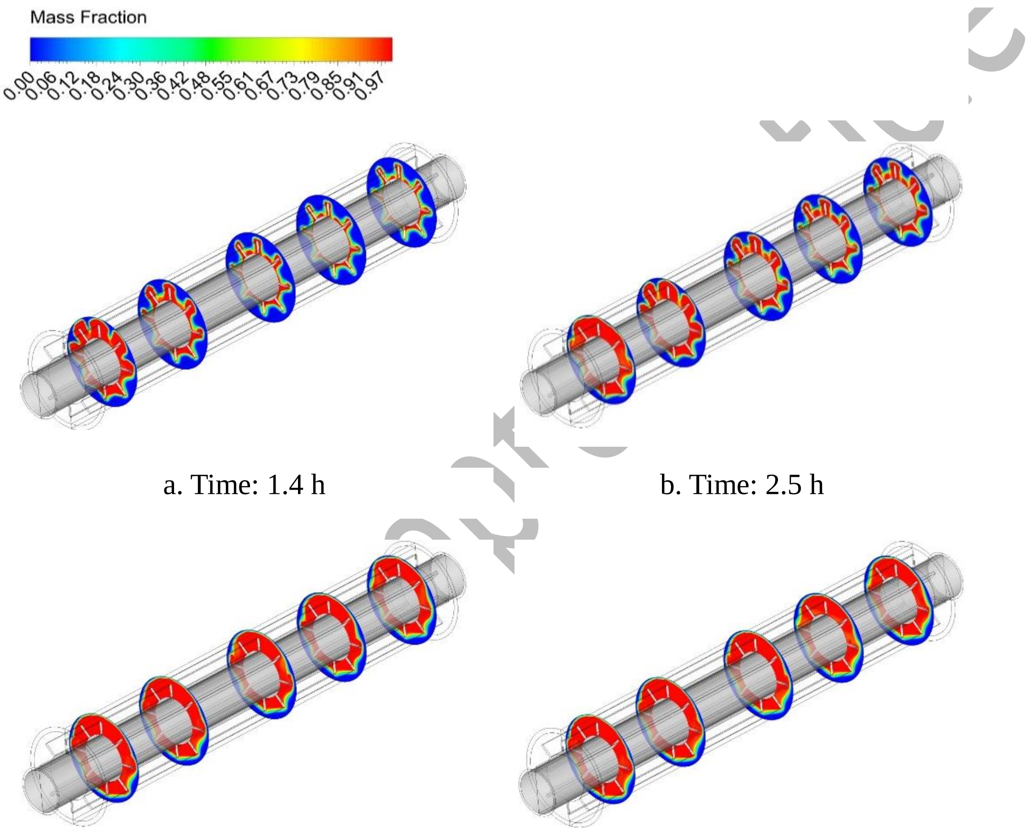
c. 4.1 h

d. 5 h

Fig. 7: Temperature distribution in PCM over time in a heat exchanger with fin and adiabatic wall

To investigate the effect of the environment and installing fins on the performance of the PCM-based heat exchanger, the obtained results as the shell wall is constant temperature (the high temperature of a cold climate 25◦ C) are presented in Figure 8. It presents paraffin's liquid volume fraction distribution in the heat exchanger. At 1.4 hours at the beginning of the tube, the conduction heat transfer mechanism is predominant. Thus, a phase change from solid to liquid occurs in paraffin by increasing thermal diffusion rate near the fins. The liquid volume fraction increases in the sections farther from the beginning of the pipe due to thermal development. As a result, a significant percentage of solid paraffin is converted to liquid in 1.4 hours. During the first 2.5 hours, heat transfer from the hot to cold layers of paraffin results in the phase change of more paraffin to liquid, and near the fin of the tube, the thickness of the liquid paraffin increases.

In the vicinity of the fins, a large part of the paraffin will be in liquid form at the end of the tube. The cumulative phase change is almost equal at the beginning and the end of the heat exchanger after 5 hours because the shell wall is assumed to have a constant temperature. The heat transfer from the hot tube to the paraffin occurs until a thermal equilibrium is established between the hot tube and the shell temperature. The heat transfer rate from the hot tube to the paraffin will gradually decrease due to the limitation of hot and cold temperatures in the dissolution range after thermal equilibrium. It should be noted that due to the assumption of a constant shell temperature 25 ◦ C at all times, paraffin does not change phase in the vicinity of the shell and remains in a solid state. It reduces the effects of natural convection and limits the circulating power of liquid paraffin.



a. Time: 1.4 h

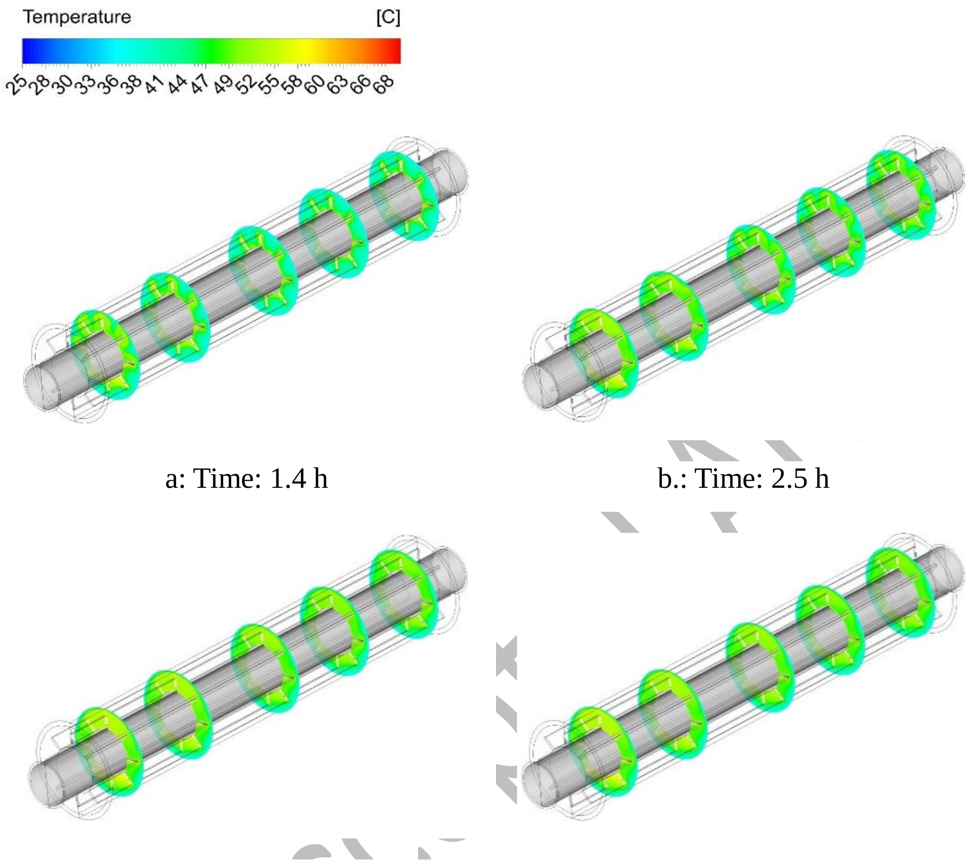
b. Time: 2.5 h

c. Time: 4.1 h

d. Time: 5 h

Fig. 8 Volume fraction of liquid phase PCM over time in the heat exchanger with fin and isothermal wall

Figure 9 presents the temperature distribution in paraffin, assuming a constant shell temperature. In the heat exchanger, the initial thermal equilibrium occurs between the hot fluid and the shell, which is filled with PCM, at 1.4 h, only a limited area near the fins experiences an increase in temperature. On the other hand, in the downstream sections of the pipe, the thermal boundary layer is altered due to two conduction and phase change mechanisms. Over time, the temperature distribution at the corresponding sections at two different times becomes almost the same. Therefore, 4.1 hours and 5 hours have almost the same structure of isothermal lines. This is also due to the limitation of temperature intervals (the constant temperature condition of the shell), which will also affect the paraffin phase change.



a: Time: 1.4 h

b.: Time: 2.5 h

c. Time: 4.1 h

d: Time: 5 h

Fig. 9 Temperature distribution in PCM over time in a heat exchanger with fin and isothermal wall

Figure 10 illustrates the variation of the liquid volume fraction in two conditions, including a hot tube with and without fines as the shell wall is insulated. In the initial time, the volume fraction equals zero, and the phase change rate is the same for both cases. The finned case experiences a more significant phase change rate after an hour than the bare tube because thermal conduction was the dominant heat transfer mechanism in the early times. In 1 hour, the total phase change from solid paraffin to liquid in finned and without fins cases is about 20% and 15%, respectively. On the other hand, the thermal conductivity mechanism intensifies the heat transfer as the liquid phase increases in the domain. Therefore, the phase change rate in the finned tube (volume fraction slope relative to time) will be much higher than in the non-finned case. Thus, the solid paraffin turns into liquid in the finned case after about 5 hours, but only half of the solid paraffin changes phase by assuming no fin. Based on Figure 10, in the case of the bare tube, there is an inflection point after 6 hours. Also, the whole of the solid paraffin melts by intensifying the natural convection heat transfer after about 10.5 hours.

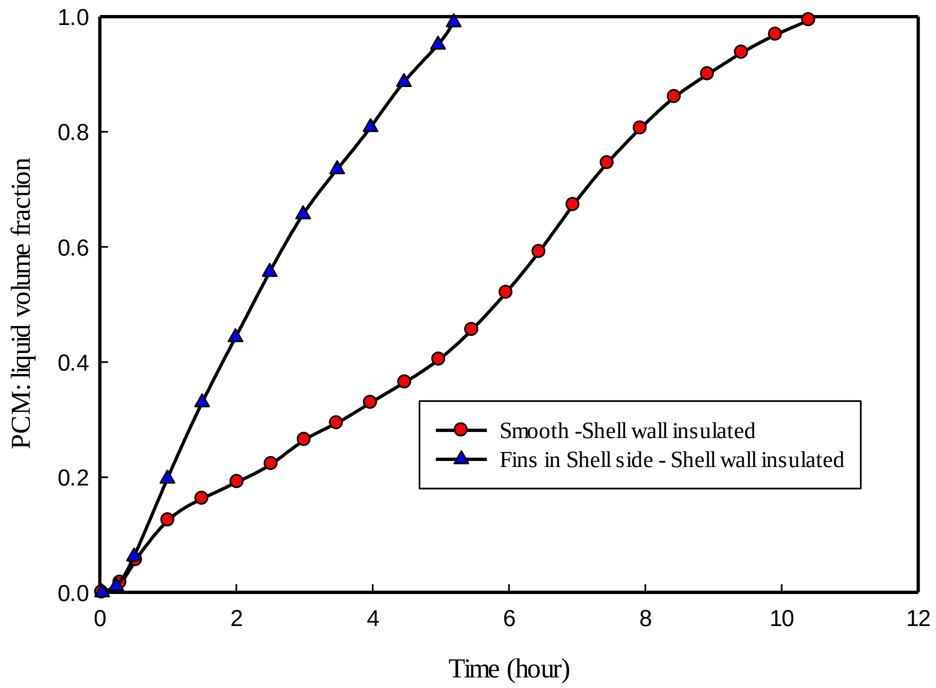


Fig. 10 Variations of PCM volume fraction versus time in finned and smooth tube cases, insulated shell wall

Figure 11 shows the average temperature changes of PCM in both cases with and without fin and assuming the adiabatic shell wall in terms of time. The PCM temperature starts at 25 °C at the initial temperature and increases with time in both cases. In the early times, the increase in temperature in the finned case had a steeper slope due to increased heat transfer by thermal conductivity. Over time, the temperature increases at a slower rate. This phenomenon is due to heat absorption caused by phase change in which the heat transfer from the hot tube to paraffin is spent at the enthalpy of phase change, and the rest leads to an increase in temperature. After 4 hours, the temperature rises with a greater slope due to the thermal development caused by the natural convection mechanism as well as the phase change phenomenon. It should be noted that in 4 hours, about 80% of solid paraffin has been melted in the finned tube case. The slope of the paraffin temperature rise is slower in the smooth tube, and it remains almost constant after 2 hours. The volume fraction of paraffin inside the heat exchanger becomes one after about 10.5 hours. Current results indicate that if the shell wall is insulated, the PCM will be melted entirely, and the hot water temperature in the tube cannot be reduced by entirely melted paraffin. As a result, a heat sink is needed to absorb the released heat during solidification and facilitate PCM solidification process. The heat sink can create a two-phase medium on the shell side to guarantee the steady state performance of the heat exchanger in cooling the hot water in the tube.

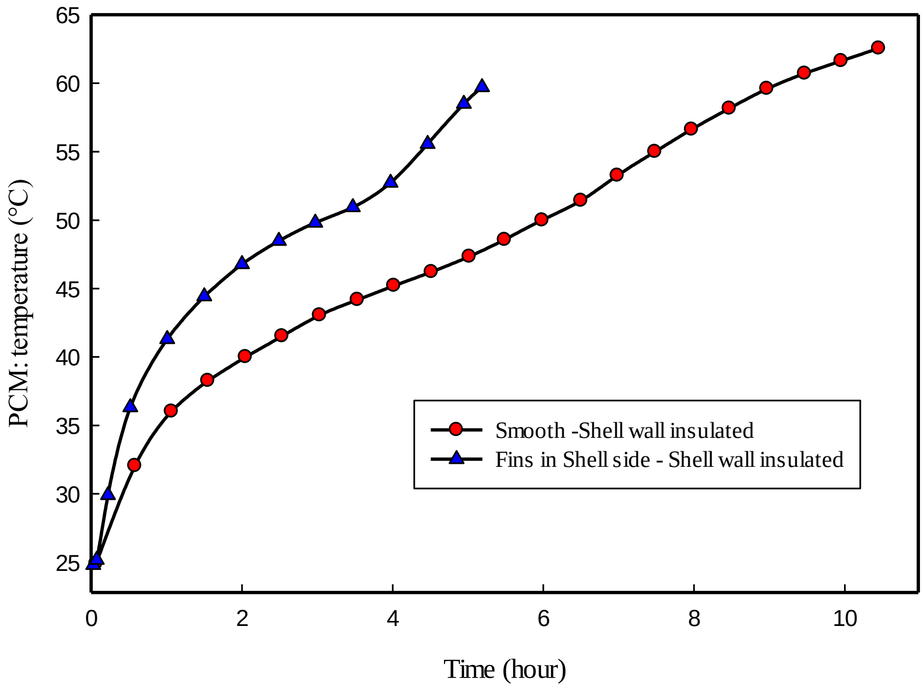


Fig. 11 Average PCM temperature versus time for finned and bare pipe, insulated shell wall

The design of optimal condensers is essential to increase thermal efficiency and reduce water consumption in the power plant. Obviously, the rate of water used for cooling will decrease as the thermal efficiency of the condenser increases. Figure 12 compares the hot water's bulk temperature variation in the heat exchanger's outlet with an insulating shell in both finned and non-finned tubes in terms of time. In the initial times, the average temperature of the water outlet from the tube increases with a high slope in both finned and smooth cases. After about 30 minutes, the slope of increasing average temperature in the finned tube decreases and converges to 63 °C. On the contrary, the average hot water temperature in the tube outlet without fin continues to increase and converges to about 65 °C. The reason for this is the improvement of heat performance due to the faster phase change of paraffin in the finned heat exchanger. Over time, more than 80% of solid paraffin changes phase to liquid, and the temperature increase in the heat exchanger is because of the created heat balance. Thus, it is observed that after 4 hours, the average outlet temperature has an insignificant increase and reaches thermal equilibrium at about 65 °C in the finned tube. The temperature rise occurs after about 6 hours in the non-finned tube. Finally, as all solid paraffin has been melted, the average temperature reaches thermal equilibrium at about 68 °C. It should be mentioned that the results of the average outlet temperature are presented up to a volume fraction equal to one in all computational space.

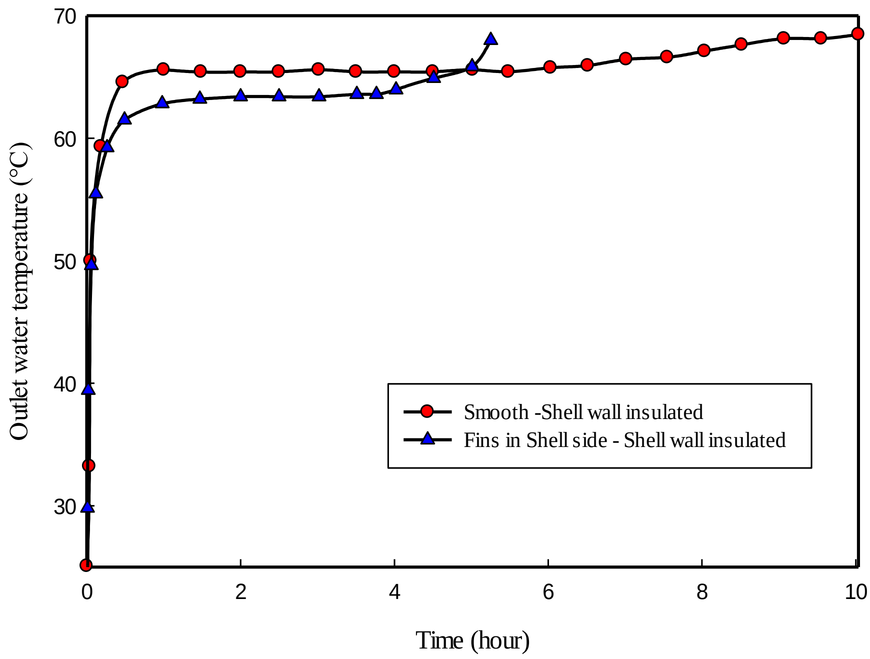


Fig. 12 Average outlet water temperature versus time for finned and smooth pipe, insulated shell wall

5.2. Heat exchanger without thermal storage capability

As mentioned, the heat conductor fins on the hot pipe improve the heat exchanger's heat performance and can reduce the hot water temperature before entering the cooling tower, thus reducing the amount of water consumed for cooling power plant systems. On the other hand, the thermal condition applied to the shell can also significantly affect the heat exchanger's heat performance. Figure 13 compares the liquid volume fraction of the PCM with two assumptions of the adiabatic and the constant temperature of the shell wall, The phase change rate was almost the same in both cases in the initial times. However, after about 30 minutes, the melting rate of solid paraffin for the initial condition of constant temperature shell increases due to the intensification of heat transfer in liquid paraffin because of free convection heat transfer in the fluid domain with hot and cold temperature differences. The rate of increasing liquid volume fraction in this heat exchanger decreases after about two hours, and eventually, no phase change is observed in PCM.

The reason for this is the assumption of a constant temperature on the shell wall, which is lower than the melting temperature of paraffin which facilitate the solidification process near the shell wall. Thus, a maximum of 56% of the solid paraffin in the finned heat exchanger with a constant temperature changes phase to liquid, and the rest remains solid. On the other hand, due to the insulation, the slope of the paraffin phase change remains almost constant over time, and the solid phase becomes completely liquid after about 5 hours.

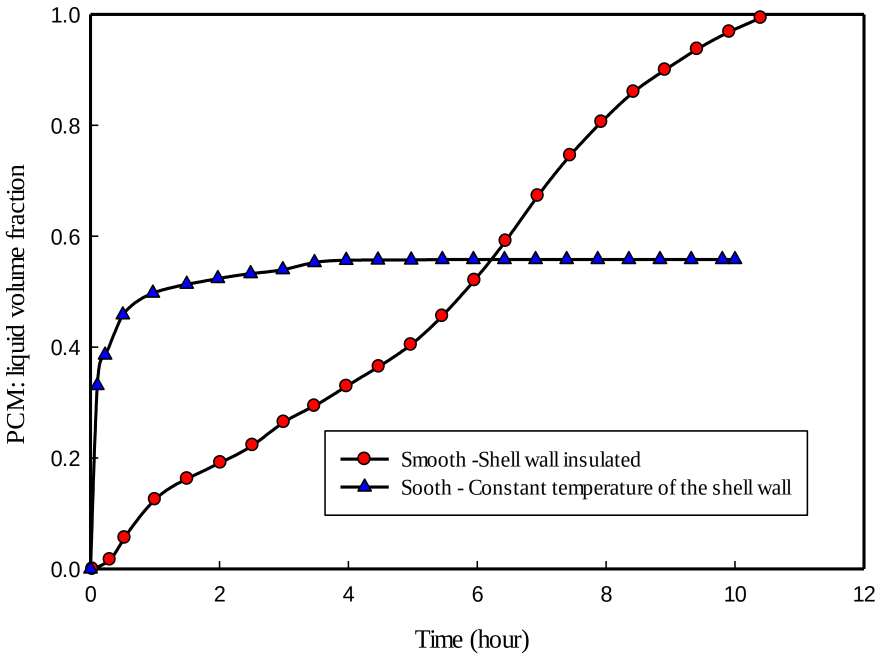


Fig. 13 volume fraction of the PCM with two assumptions of the adiabatic and the constant temperature of the shell wall

Figure 14 presents the average temperature of PCM in cases with an insulated wall shell and constant temperature. In both cases, the temperature of the solid paraffin starts to rise from the initial temperature (25°C). The average temperature of paraffin in the constant shell temperature case increases at a higher rate, but the temperature rise slope decreases after 1 hour. So over time, the average temperature remains almost constant at 47°C, due to the thermal balance between the shell's temperature and the tube (hot water). As shown in Figure 14, there is no phase changing occurred in paraffin after 4 hours because the paraffin bulk temperature is lower than its melting temperature. However, due to the unlimited temperature range in the case of the insulated tube wall, the paraffin temperature is continuously increased in the shell-adiabatic case. The average paraffin temperature reaches about 47°C after 5 hours, and the volume fraction in the paraffin field will equal 1. Based on the results, in case of non-insulated shell wall, the thermal equilibrium between the melted paraffin adjacent to the wall of the hot water tube and the solidified paraffin next to the shell has been established, and the volume fraction of the liquid phase stays the same with time.

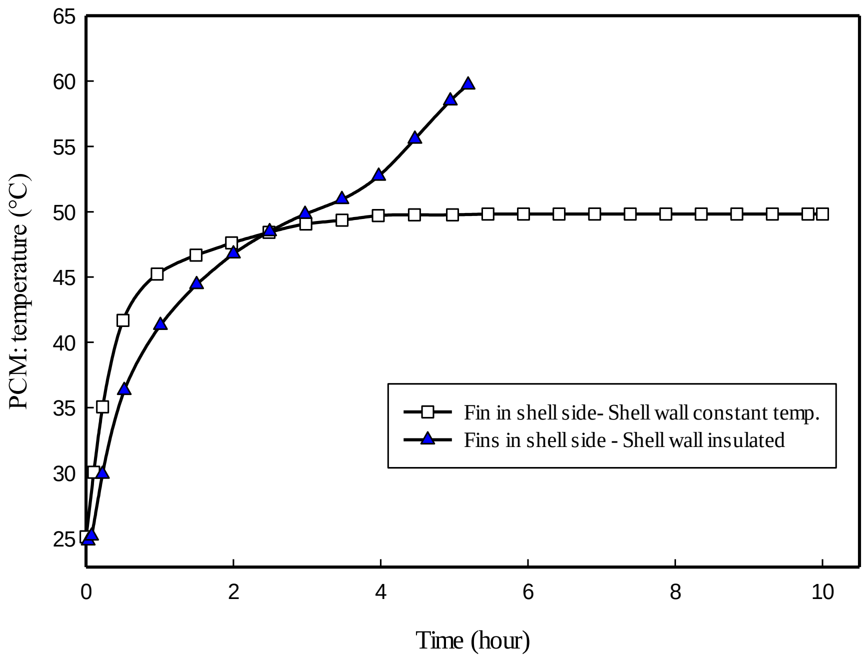


Fig. 14 PCM-temperature against time in cases of adiabatic and fixed temperature conditions in shell wall

As mentioned before, one of the current research goals is to reduce the temperature of the water entering the cooling tower. A shell and tube heat exchanger based on phase change material has been used for this purpose. The outlet temperature variation of the water in terms of time is shown in figure 4 in two cases, including the insulated shell wall and constant temperature.

Based on what was mentioned before, in the insulating shell wall, paraffin completely changes to the liquid phase by the absorption heat of the hot water. However, after 5 hours, the heat exchanger loses its efficiency by completely melting the phase change material. As shown in Figure 14, in the earlier time, the temperature at the heat exchanger outlet decreased significantly (compared to the inlet temperature, which was 70 degrees). Because in the earlier time the paraffin is in solid state and absorbed the heat of the hot water until reached to around the melting temperature. Then, the outlet water temperature remains constant for about three hours, indicating the melting process of paraffin at a constant temperature. After 4 hours, it can be seen that the outlet water temperature increases linearly because as the melting process finished and all the phase change material has turned into the liquid phase. Therefore, it is impossible to absorb the heat of the pipe's hot water in the shell, and the heat exchanger has lost its efficiency. To utilize the stored energy in the shell, using the cold-water flow in the shell part as a heat sink can be a convenient suggestion. Because in the shell side, a two-phase solid-liquid environment is permanently established by creating an area whose temperature is lower than the freezing temperature of the phase change material. By the way, it is possible to reduce the temperature of the hot water entering the cooling tower and increase the temperature of the cold water to supply the needed for consuming hot water or needed in the boiler preheating process.

A comparison of heat exchanger cooling capabilities under different conditions of the shell indicates that the heat exchanger functions well and hot water is stable at 64.5 degrees at the outlet.

Therefore, if a heat exchanger with stored thermal energy is not desired for any reason (from the thermoeconomic point of view), using it in a cold climate can permanently reduce the temperature of the inlet water entering the cooling tower. This idea, reducing the inlet temperature of the cooling tower without additional water and power, can be a good idea to reduce the water consumption of cooling towers. Of course, it is necessary to examine this proposal from a thermoeconomic point of view.

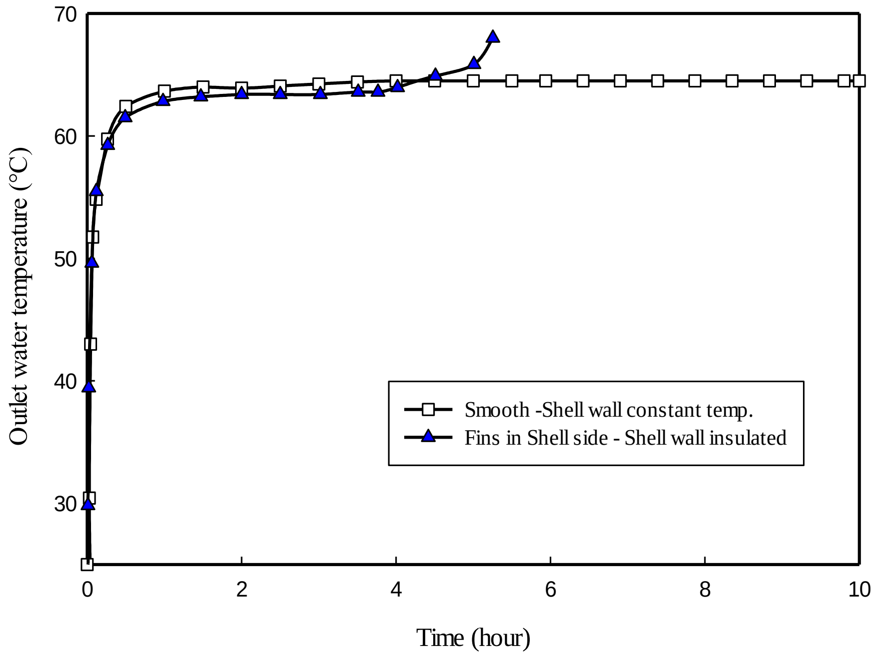


Fig. 15 Average outlet water temperature variations with respect to time

1. Conclusion

This numerical simulation investigates the possibility of reducing the inlet water temperature of a cooling tower in a power plant. The hot water temperature is reduced before entering the cooling tower by using a shell and tube heat exchanger filled with phase change materials to reduce the water consumed in the cooling tower. The behavior of the phase change material and the outlet water temperature in two conditions, including insulated shell wall and constant heat temperature, have been studied. In the insulated shell wall, thermal energy can be stored when it is absorbed in the melting process and be used to supply the required hot water. By simulating the three-dimensional, unsteady flow and the enthalpy-porosity method, the melting and solidification processes of the phase change material and the temperature changes of hot water have been studied in terms of time.

The results show that in the case of the insulated shell wall, the heat exchanger will not have a steady performance and can reduce the output water temperature only in the first 10 hours. In this situation, using a heat sink to take advantage of the stored thermal energy can be an excellent suggestion to maintain permanent performance. Conducting cold water with pipes in the shell section of the heat exchanger can absorb the stored heat and cause solidification. Therefore, the melting and solidification process on the shell side causes the heat exchanger to have a stable performance in reducing hot water temperature. In order to improve the heat transfer rate, fins have been connected to the hot water pipe, which has reduced the melting time from 10 hours to 5 hours, which shows that it has had a significant effect on the melting process of the changing material. Also, the performance of the heat exchanger has been studied without any insulation in a cold climate region. The results show that if the environmental conditions can solidify the melted paraffin, solidification and melting occur near the shell and tube, respectively. As a result, it causes the heat exchanger's stable performance in reducing the hot water temperature. The thermo-economic analysis of the present idea can examine the economic and engineering aspects of the present proposal, which are suggested for future studies. Overall, the current idea could reduce the hot water temperature before entering the cooling tower by about 6 ℃, which needs more parameter analysis to optimize the PCM-based heat exchanger characteristics regarding the environmental condition and other control parameters, which is suggested for future investigations.

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