Numerical investigations into thermal performance of phase change emulsion in a fin-and-tube heat exchanger

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Abstract

Past investigations have identified phase change emulsions (PCMEs) as potential working fluids that could be used to reduce energy consumption in heating, ventilation and air conditioning (HVAC) systems. But how PCME behaves in a fin-and-tube heat exchanger, which is commonly found in HVAC system, is still unclear. The paper focused on theoretical studies of thermal performance of a novel PCME PCE-10 in a fin-and-tube heat exchanger. The research analyzed heat transfer and flow behavior in fin-and-tube heat exchangers and the results showed that PCE-10 has several performance advantages as a cold storage medium, which provides a basic basis for the design of a new type of practical cold storage device.

Keywords: phase change emulsion; fin-and-tube heat exchanger; heat transfer; CFD

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1. INTRODUCTION

Heating, ventilation and air conditioning (HVAC) system is one of the largest energy end-users both in buildings, which is responsible for 50% of residential buildings’ energy consumption, and in commercial buildings, such as office buildings, retail stores, warehouses, schools, hotels, etc. where the figure can be as high as 60% [1].

Currently, there is an increasing demand for cooling, which is expected to increase even further in the years to come due to climate changes. Past investigations have identified phase change emulsions (PCMEs) as potential working fluids that could be used to reduce energy consumption in HVAC systems [2, 3]. PCMEs basically contain water and phase change materials (PCMs) such as paraffin. PCMEs use the latent heat capacity of paraffin as well as the sensible heat capacity of water and that of the PCM to store thermal energy. PCMEs do not lose the fluidity during phase transition due to the carrier fluid. Hence, PCMEs can be directly used both in storage systems and in pumped systems such as cold supply network.

However, the optimal design of an integrated PCME air conditioning system requires a good understanding of flow behavior and heat transfer characteristics of PCME in heat exchangers that cannot currently be readily deduced from manufacturer's data or published data. This is because most of the published data cover straight pipes and not coiled pipes. Even though some technical data on various types of heat exchangers (e.g. air-emulsion direct-contact heat exchanger [4], double-coiled heat exchanger [5, 6], circular tube [7–9], plate heat exchange [10], helical coil pipes [11]) are available, there is limited information regarding fin-and-tube heat exchanger that is the type commonly used in air conditioning systems.

To this end, computational fluid dynamics (CFD) investigation was carried out to analyze the flow behavior and heat transfer characteristics of the developed PCE-10 sample in a fin-and-tube heat exchanger.

2. MATERIAL PREPARATION

In order to study the thermal performance, a stable and suitable paraffin/water PCME (PCE-10) was developed based on an organic paraffin material called RT10. The detailed preparation procedure can be found in [12]. The final product is shown in Figure 1. The developed PCE-10 has a phase change temperature
range of 4–11°C falling into the temperature range of chilled water air conditioning system and is an attractive candidate for cooling applications.

An analysis of the thermophysical properties revealed the particle size of the PCE-10 sample to be 3 μm with a storage heat capacity of almost twice as much as that of water and with a negligible level of sub-cooling.

The PCE-10 also achieved a good level of storage stability for 9 months and was able to withstand over 500 cycles in a pumping system without any significant sign of degradation to particle size and heat storage capacity. However, the viscosity of the PCE-10 was found to be much higher than water, which could contribute to high pressure drop in a pumping system. Its thermal conductivity was also found to be ~30% lower than the value for water and could influence heat transfer.

Listed in Table 1 are the simulation data for the PCE-10. The heat capacities were calculated by using Equations (1) and (2).

\[
C_{p,m,\text{eff}}(T) = \begin{cases} 
X_W C_{p,w} \Delta T + X_{PCM} C_{p,PCM} \Delta T & T_S < T_e \\
X_W C_{p,w} \Delta T + X_{PCM} C_{p,PCM} \Delta T + \frac{X_{PCM} \Delta h}{\Delta T} & T_e < T_S < T_c \\
X_W C_{p,w} \Delta T + X_{PCM} C_{p,PCM} \Delta T & T_c < T_s 
\end{cases}
\]

\[
C_{p,PCM} = \frac{h}{T_{MPH} - T_{MPL}}
\]

### Table 1. Specification of PCE-10.

| Properties                      | Measured result |
|---------------------------------|-----------------|
| Density                         | 940 kg/m³       |
| Specific heat capacity          |                 |
| \( T < 4°C \)                   | 3800 J/kg K     |
| \( 4°C \leq T \leq 11°C \)     | 8718 J/kg K     |
| \( T > 11°C \)                  | 3800 J/kg K     |
| Thermal conductivity            | 0.40 W/mK       |
| Consistency index k             |                 |
| \( 10°C \)                      | 0.1887          |
| \( 25°C \)                      | 0.102           |
| Flow behavior index n           |                 |
| \( 10°C \)                      | 0.662           |
| \( 25°C \)                      | 0.675           |

### Table 2. Fin-and-tube coil data.

| Parameter                     |                  |
|-------------------------------|-----------------|
| Finned-tube length            | 250 mm          |
| Number of circuits            | 3               |
| Distance between rows         | 22 mm           |
| Number of tubes per circuit   | 9               |
| Distance between columns      | 25 mm           |
| Tube material                 | Copper          |
| Tube inner diameter D         | 9.3 mm          |
| Tube wall thickness           | 0.35 mm         |
| Fin material                  | Aluminum        |
| Fin thickness                 | 1 mm            |
| Face area                     | 250×250         |

### 3. MATHEMATICAL MODELING

#### 3.1. Basic assumptions

The following assumptions were considered to simplify the simulation:

- PCE-10 was treated as an incompressible, steady state, homogeneous single-phase fluid with negligible effect of viscous heating;
- fluid properties were assumed to be constant with respect to temperature, except heat capacity of material;
- phase change process was assumed to take place within a specific temperature range but volume change was neglected;
- buoyancy effects were neglected as density was assumed to be constant;
Figure 3. *Physical model of finned-tube coil.*

Figure 4. *3D model: (a) finned pipe and (b) fluid flow.*
3.2. Governing equation

3.2.1. Modeling conjugate heat transfer

The governing equations include continuity, momentum and energy equations from which all the theory and more complex models are derived. The continuity and momentum equations were used to calculate velocity vector. The energy equation was used to calculate temperature distribution and wall heat transfer coefficient. The basic equations for the conjugate heat transfer between fluid (PCE-10) and solid (finned tube) are summarized below [13].

Conservation of mass:
\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) = 0
\]  
(3)

Conservation of momentum:
\[
\frac{\partial (\rho \mathbf{v})}{\partial t} + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla P + \nabla \left[ \mu \nabla \mathbf{v} - \frac{2}{3} \mu \mathbf{v} \right] + \rho \mathbf{g} + \mathbf{F}
\]  
(4)

Conservation of energy for fluid:
\[
\frac{\partial (\rho E)}{\partial t} + \nabla \cdot (\rho E \mathbf{v}) = \nabla \cdot (k_{\text{eff}} \nabla T) + S_E
\]  
(5)

Conservation of energy for solid:
\[
0 = \nabla \cdot (k_{\text{eff}} \nabla T)
\]  
(6)

Where \(P\), \(\mu_m\) and \(\rho g\) are static pressure, molecular viscosity and gravitational body force, respectively. \(\mathbf{F}\) defines external body force and also contains other model-dependent source terms such as porous-media and user-defined sources. \(E\) equals to the sensible enthalpy for phase \(p\). \(S_E\) includes any other volumetric heat sources.

3.2.2. Modeling turbulence flow

Turbulent flows are characterized by fluctuating velocity fields. These fluctuations mix transport quantities such as momentum, energy and species concentration and cause them to fluctuate as well. Since these fluctuations can be of small scale and high frequency, they tend to be computationally too expensive to simulate directly in practical engineering calculations. Instead, the instantaneous governing equations can be time-averaged or otherwise manipulated to remove the small scales, resulting in a modified set of equations that are computationally less expensive to solve. However, the modified equations contain additional unknown variables and, therefore, turbulence models are needed to determine these variables in terms of known quantities.

FLUENT provides the following choices of turbulence models that are widely used: K-epsilon model (K-\(\varepsilon\) model) and K-omega model (K-\(\omega\) model). In the derivation of the K-\(\varepsilon\) model, the assumption is that the flow is fully turbulent and the effects of molecular viscosity are negligible. The standard K-modal is therefore valid only for fully turbulent flows. The shear-stress transport (SST) k-\(\omega\) model was developed by Menter and Egorov [14] to effectively blend the robust and accurate formulation of the k-\(\omega\) model in the near-wall region with the free-stream independence of the k-\(\omega\) model in the far field. To achieve this, the K-\(\varepsilon\) model is converted into a k-\(\omega\) formulation. The SST k-\(\omega\) model is similar to the standard k-\(\omega\) model but includes the following refinements.

• The standard k-\(\omega\) model and the transformed K-\(\varepsilon\) model are both multiplied by a blending function and both models are added together. The blending function is designed to be one in the near-wall region, which activates the standard k-\(\omega\) model, and zero away from the surface, which activates the transformed K-\(\varepsilon\) model.
• The SST model incorporates a damped cross-diffusion derivative term in the \(\omega\) equation.
• Modified turbulent viscosity formulation to account for the transport effects of the principal turbulent shear stress.

These features make the SST model more accurate and reliable for a wider class of flows than the standard k-\(\omega\) model. In industry, flows inside a coiled tube are mainly turbulent or transition flow, the SST k-\(\omega\) model is selected.

Transport equation for SST k-\(\omega\) model:
\[
\frac{\partial (\rho k)}{\partial t} + \nabla \cdot (\rho k \mathbf{v}) = \nabla \cdot \left[ \left( \frac{\mu + k}{\sigma_k} \right) \nabla k \right] + G_k - Y_k + S_k
\]  
(7)
\[ \frac{\partial}{\partial t} (\rho \omega) + \nabla \cdot (\rho k \vec{v}) = \nabla \cdot \left[ \left( \mu + \frac{\mu_t}{\sigma_\omega} \right) \nabla \omega \right] + G_\omega - Y_\omega + D_\omega + S_\omega, \] (8)

where \( \sigma_k \) and \( \sigma_\omega \) are constant turbulent Prandtl numbers for k and \( \omega \), respectively. \( Y_k \) and \( Y_\omega \) represent the dissipation of k and \( \omega \) due to turbulence. \( D_\omega \) represents the cross-diffusion term.

### 3.2.3. 

**Modeling PCMEs**

An effective heat capacity approach [15], first introduced by Alisetti and Roy [16, 17], was used to model phase change behavior, since it is more easily implemented in practice. In the effective specific heat model, the PCM is assumed to be melted over a constant temperature, so that the phase change effects are

| \( h_{ai} \) (W/m²K) | PCE-10 | water |
|----------------------|--------|--------|
| 40                   | ![Temperature contour plot for outlet at \( v = 0.3 \) m/s.](image) | ![Temperature contour plot for outlet at \( v = 0.3 \) m/s.](image) |
| 50                   | ![Temperature contour plot for outlet at \( v = 0.3 \) m/s.](image) | ![Temperature contour plot for outlet at \( v = 0.3 \) m/s.](image) |
| 60                   | ![Temperature contour plot for outlet at \( v = 0.3 \) m/s.](image) | ![Temperature contour plot for outlet at \( v = 0.3 \) m/s.](image) |

\( T_\omega = 300 \) K
directly incorporated into energy equation by assuming specific heat capacity of PCM to be a function of temperature, as Figure 2 shows.

In order to use the above equation, it is necessary to define \( T_{\text{MPL}} \), onset melting temperature, and \( T_{\text{MPH}} \), end phase change temperature. Outside the phase change range, i.e. for \( T < T_{\text{MPL}} \) and \( T > T_{\text{MPH}} \), the specific heat of emulsion is the mass-averaged specific heat of its constituents \[ 18\]. Since the specific heat capacity of PCM in its solid and liquid phases are usually similar and do not vary significantly with temperature, it can be assumed as a constant. As a result, the specific heat of PCME, which was calculated using the above equation, can also be assumed to be a constant outside the phase change range.

In the phase change range, the specific heat of PCE can be obtained from suitable analytical tests \[ 12\]. The specific heat is related to latent heat of PCM \( h \), through the following equation:

\[
\begin{align*}
h &= \int_{T_{\text{MPL}}}^{T_{\text{MPH}}} C_p, \text{PCM} dT. \quad (9)
\end{align*}
\]

Previous studies with phase change slurries showed that the shape of specific heat capacity-temperature curve has a little effect on heat transfer process \[ 16, 17\]. As a result, it is possible to assume the specific heat capacity of PCM remains constant during the phase change process. Its effective value in the phase change range is therefore given by Equation (2).

3.3. Geometry and meshing

Figure 3 shows the views of the fin-and-tube heat exchanger that was considered for the modeling exercise. It consists of horizontal tubes with coiled ends and thin vertical fins meant to increase the heat transfer area. The physical parameters are given in Table 2.

To analyze the problem with CFD, the geometry of the heat exchanger has been prepared in ANSYS workbench design module, as shown in Figure 4.

The geometry was meshed with automatic meshing method of defined global size, as shown in Figure 4. To account for the high variable gradients that usually exist near the pipe wall due to the no-slip condition of the wall, inflation layers were created near the wall. The number of nodes and elements for the current case were 733 087 and 2 197 511. The mean orthogonal qualities for the whole model and fluid domain were 0.679 and 0.948, respectively.

3.4. Boundary condition

The simulation was subjected to the following boundary conditions that were set according to design conditions of real air conditioning systems \[ 19, 20\]. Velocity inlet boundary condition was applied at the inlet. A 7°C fluid entered the heat exchanger at 0-1 m/s. The turbulence intensity was fixed at 5% with a hydraulic diameter pipe of 9.3 mm. The pressure at outlet was assumed to be 0.

There are two main types of thermal boundary conditions that can be applied at the wall. One is specifying a constant heat flux and the other is specifying a constant wall temperature. However, in heat exchangers, the incoming fluid exchanges heat with the pipe walls. This will result in change in temperature of the fluid as it moves down stream. This change in temperature of fluid will also result in change in temperature differential between the walls and fluid, which ultimately leads to the change in heat transfer rate. The problem is therefore neither isothermal nor iso-flux. In such cases, FLUENT provided a third option: convective heat transfer wall boundary, which is the best and most accurate approximation of real-life example. External air heat transfer coefficients were determined using following empirical correlations \[ 21\].

\[
\begin{align*}
  j &= 16.06 \frac{\text{Re} D^{-1.02 \left( \frac{\rho_f}{\rho} \right)^{0.256}} \left( \frac{A_{\text{tube}}}{A_{\text{total}}} \right)^{-0.601} \left( \frac{N_{\text{bank}}}{N_{\text{total}}} \right)^{-0.069} \left( \frac{\rho_f}{D} \right)^{0.48}}{\text{Pr}^{2/3}} \quad (10) \\
  h_{\text{a}} &= \frac{j \rho (m_{\text{air}}/\rho A_{\text{tube}}) C_p}{\text{Pr}^{2/3}} \quad (11)
\end{align*}
\]

In this case, three different air flow velocities of 0.8, 1 and 1.5 m/s were studied and the corresponding convective heat transfer coefficients were 38.7, 49.9 and 59.3 W/m²K, respectively. Therefore, the convective heat transfer coefficients of 40, 50 and 60 W/m²K were applied at the external wall surface of the pipe with free-stream air temperature of 300 K. A no-slip and no-penetrating boundary conditions were also imposed on the wall surface of the pipe.

3.5. Solution control

The pressure–velocity coupling was done using coupled scheme and the pressure discretization was achieved by the PRESTO
procedure. A second-order upwind scheme was employed to discretize the convection terms, diffusion terms and other quantities resulting from the governing equations.

3.6. Result and discussion

Figure 5 shows the total pressure drop for pure water and PCE-10 versus the flow velocity at different temperatures. It can be seen that at the same velocity, the PCE-10s produced much bigger pressure drops, i.e. 2–10 times higher than water due to high viscosity. Moreover, the pressure loss of the emulsion at 10°C (within the phase change range of 4–12°C) was higher than that at room temperature because of the existence of solid paraffin.

The temperature profiles at the exits are given in Figure 6. The average outlet temperature of PCE-10 was lower than that of water because the heat capacity of PCE-10 was twice as much as that of water.

Figure 7 shows the temperature profiles for $h_{\text{air}} = 60 \text{ W/m}^2\text{K}$ and $T_{\infty} = 300 \text{ K}$. It can be seen that PCE-10 helped to reduce the wall temperature and therefore increased the temperature gradient between wall and air. However, the difference between temperature gradients for PCE-10 and water gradually reduced as the flow rate increased.

At the constant external heat transfer coefficient, heat transfer rate was proportional to the temperature gradient between wall and air; it is thus reasonable to believe that PCE-10 would help increase the heat transfer rate significantly at low flow rates but gradually lost its effectiveness at heat transfer enhancement when flow velocity was higher.

Figure 8 compares the heat transfer rates of the PCE-10 and water at different external heat transfer coefficients. It is obvious that total heat transfer rate increased with external heat transfer coefficient increasing. It can be seen that PCE-10 did help to increase the total heat transfer rate as predicted at a constant heat transfer coefficient. For example, at $h = 60 \text{ W/m}^2\text{K}$, the total heat transferred by PCE-10 was about 1.1–1.3 times the value for water at the same flow rate. The same trends were witnessed in the other two cases.

However, the heat transfer enhancement did vary at different flow velocities, as the differential heat transfer ($Q_{\text{PCE-10}} - Q_{\text{Water}}$) did not increase linearly with flow rate. There was rather a significant increase in the heat transfer rates at lower flow rates with the maximum heat transfer enhancement occurring within the range of 0.5–0.6 m/s. Similar results were also obtained for $h = 50 \text{ W/m}^2\text{K}$ and $40 \text{ W/m}^2\text{K}$ where the maximum heat transfer enhancement occurred around 0.4 m/s.

Figure 9 shows the relationship between pumping power against heat transfer rate. Pumping power is defined as

$$P = p \times V,$$

where $P = \text{pump power}$, $p = \text{pressure drop}$ and $V = \text{volume flow rate}$.

It was observed that for the same heat transfer rate, the pumping power was decreased in comparison with water. However, at lower heat transfer rate ($\leq 1.1 \text{ kW}$), the pump power consumption of PCE-10 would still be higher than water. To achieve a heat transfer rate of 1.1 kW, the required flow velocities at $h = 60$, 50 and 40 W/m²K should be 0.3, 0.3 and 0.4 m/s, respectively. Thus, in the
Figure 9. Pumping power v/s heat transfer rate.

Figure 10. Nusselt number v/s Reynolds number plot.

Figure 11. Heat transfer rate against flow velocity with different fluid thermal conductivities at 60 W/m²K and 27°C free stream temperature.

best interest of pump energy saving, the PCE-10 flow rate should not be lower than 0.3 m/s.

The fluid-side convective heat transfer coefficient, Nusselt number and generalized Reynolds number for the PCE-10 were calculated from the following equations:

\[ h = \frac{Q}{T_{\text{wall}} - T_{\text{fluid}}} \]  \hspace{1cm} (13)

\[ Nu = \frac{hD_i}{K} \]  \hspace{1cm} (14)

\[ Re = 8^{1-n} \left( \frac{3n+1}{4n} \right)^{-n} \rho \frac{u^2 - n D_i^n}{\eta} \]  \hspace{1cm} (15)

where \( Q \) = surface flux \( (\text{W/m}^2) \), \( T_{\text{wall}} \) = average temperature \( (^\circ \text{C}) \) of tube wall, \( T_{\text{fluid}} \) = average temperature \( (^\circ \text{C}) \) of fluid, \( D = \) inner diameter of pipe \( (\text{m}) \), \( K = \) thermal conductivity of fluid.

Figure 10 represents the plot of Nusselt number versus Reynolds number \( (Re) \) for the PCE-10 in a velocity range of 0-1 m/s, which fell in the laminar regime with \( Re < 2300 \). The results showed a remarkable improvement in the heat transfer performance. In comparison with water, the Nusselt number did increase by more than 10 times at the same Reynolds number condition. Therefore, the Reynolds number was established as a dominating parameter affecting the Nusselt number of the emulsion.
3.7. Factors affecting heat transfer characteristics
For a viscous fluid flow inside a pipe, fluid layer coming into contact with the wall surface comes to a complete halt and adjacent layer slow down due to viscosity effect. A thin velocity boundary layer is formed in near-wall region. Thickness of velocity boundary layer increases along the flow length until the entire pipe is filled up with boundary layer. Due to the no-slip condition of a stationary wall and the viscous damping by the fluid, there is a very thin layer of fluid at the wall in which the flow velocity can be considered as zero. The main heat transfer method in this layer is conduction. Energy exchange between wall and fluid has to pass through this stationary layer. Ignoring the effect of radiation, the total amount of heat transferred between wall and fluid stream equals to the heat conduction through the thin layer. Applying the Fourier law, the heat flux can be expressed as

\[ q = -k \frac{\partial T}{\partial n} \bigg|_{n=0}. \] (16)

So, increasing conductivity of fluid shall help to increase heat transfer. To this end, series of CFD simulations were conducted to explore the effectiveness of increasing thermal conductivity. The simulation was subjected to the following boundary conditions: \( v = 0.5 \text{ m/s}, T = 7^\circ \text{C}, T_{\infty} = 27^\circ \text{C} \) and \( h_0 = 60 \text{ W/m}^2\text{K} \). The conductivity of PCE-10 was varied from 0.44–1 W/mK, i.e. 1.1–2.5 times of original conductivity.

The total heat transfer rate and calculated heat transfer coefficient against thermal conductivity were plotted in Figure 11. As shown, heat transfer rate and heat transfer coefficient increased with fluid thermal conductivity. If the conductivity increased to 1 W/mK, the total heat transfer rate would increase by 10% and the corresponding \( h_{\text{fluid}} \) was doubled.

The regression between heat transfer rate and thermal conductivity (with \( R^2 = 0.99 \)) is

\[ Q = -319.53K^2 + 758.34K + 1321.5, \] (16)

where \( Q \) = heat transfer rate (W) and \( K \) = thermal conductivity (W/mK).

It is worth pointing out that heat transfer rate would peak around 1771 W when thermal conductivity increased to 1.2 W/mK and suffer a drop afterwards.

4. CONCLUDING REMARKS

CFD analysis for PCE-10/coiled tube system was carried out. The results of rheological performance and heat transfer parameters have been compared with water under similar operating conditions. The flow behavior was analyzed within a range of 0.1–1 m/s, in which the flow was laminar. The pressure drop was found to be 2–10 times much higher than that of water under identical conditions. The heat transfer study was performed under convective heat transfer wall boundary. It was found out that the PCE-10 did help increase the total heat transfer rate by a factor of 1.1–1.3 within flow velocity of 0.4–0.6 m/s. However, the PCE-10 gradually lost its advantages as flow velocity increasing.

The relationship between pump power and heat transfer rate was also investigated. It was observed that for the same value of heat transfer rate, the pumping power was decreased in comparison with water. However, at lower heat transfer rate (≤1.1 kW), the pump power consumption of PCE-10 would still be higher than water. Thus, in the best interest of pump energy saving and optimum heat transfer efficiency, the optimal working range should be 0.3–0.6 m/s.

The study also explored the factors could affect the heat transfer characteristic of PCE-10, such as air side conditions, flow condition and fluid properties. An increase in thermal conductivity also helped to enhance the heat transfer effect. For instance, a 10% increase in heat transfer rate could be achieved if the conductivity was increased to 0.6 W/mK. Therefore, thermal conductivity enhancement is recommended for further studies. However, in this part of the study, the simulation was only carried out in the laminar region. Further works are also recommended to numerically explore the heat transfer performance in the turbulent regime. In addition, heat transfer studies with different geometries would also be helpful to enhance the knowledge of heat transfer behavior.

CONFLICT OF INTEREST

The authors declared that they have no conflicts of interest to this work.

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