LAMINAR FLOW FORCED CONVECTION HEAT TRANSFER BEHAVIOR OF PHASE CHANGE MATERIAL FLUID IN MICROCHANNELS WITH STAGGERED PINS

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ABSTRACT
Microchannels have been extensively studied for electronic cooling applications ever since they were found to be effective in removing high heat flux from small areas. Many configurations of microchannels have been studied and compared for their effectiveness in heat removal. However, there is little data available in the literature on the use of pins in microchannels.

Staggered pins in microchannels have higher heat removal characteristics because of the continuous breaking and formation of the boundary layer, but they also exhibit higher pressure drop because pins act as flow obstructions. This paper presents numerical results of two characteristic staggered pins (square and circular) in microchannels. The heat transfer performance of a single phase fluid in microchannels with staggered pins, and the corresponding pressure drop characteristics are also presented.

An effective specific heat capacity model was used to account for the phase change process of PCM fluid. Comparison of heat transfer characteristics of single phase fluid and PCM fluid are made for two pins geometries for three different Reynolds numbers. Circular pins were found to be more effective in terms of heat transfer by exhibiting higher Nusselt number. Circular pin microchannels were also found to have lower pressure drop compared to the square pin microchannels.

Keywords:  
Microchannels, Phase Change Material (PCM) fluid, staggered square and circular pins, and aspect ratio.

INTRODUCTION
Microchannels have been studied for electronic cooling applications because of their capacity in removing high heat flux from small areas. Many configurations of microchannels have been studied and compared for their effectiveness in heat removal [1-8]. However, there is little data available on the use of pins in microchannels. Also, except for few publications, there are virtually no optimization studies in this area.

Recent studies of microchannels have also shown that a phase change material (PCM) fluid improves the heat removal rate while maintaining lower wall temperature. Experiments have shown that PCM fluids exhibit an enhanced heat capacity due to the phase change material’s latent heat of fusion [9-18]. Hao and Tao [18-19] did an extensive study and evaluated the performance of PCM particle flow in circular microchannels. They modeled the particle flow separately from the mean flow using source terms in momentum and energy equations. They also considered the particle-particle interactions and particle-depletion layer effects near the wall. Said [20] performed a computational fluid dynamics (CFD) analysis of PCM slurry flow in microchannels with thick walls taking into account conjugate heat transfer. Results from the previous efforts on PCM slurry flows have been promising with the main advantage being the lower wall temperature for same heat flux removal compared to conventional single phase fluids.

The current work focuses on use of PCM fluid in microchannels with staggered pins, both circular and square pins. Also, the heat transfer characteristics without the presence of pins (called as “no pins”) is studied and compared with square and circular pins configurations.
SPECIFIC HEAT MODEL

Simulating the complete phase change process is difficult with the currently available multi-phase models in commercial software, and even if possible, it would be computationally expensive. Two types of effective modeling approaches were found in the literature. One uses a heat source term in the energy equation. The other considers a specific heat model. While both the methods are found to be effective, the use of the specific heat model is simpler and easier to implement into a computer code. The latter approach was used in the current simulation.

The model assumes that the phase change can be approximated by change in the specific heat of the bulk fluid within the melting temperature range of the phase change material. Yamagishi et al. [10] has found that this model gives results that are comparable with experimental findings. Also, using this type of model for the specific heat is simple and straightforward to implement in commercial software like Fluent. The following equations were used to account for effective specific heat.

For \( T < T_1 \) or \( T > T_2 \) [15]:

\[
C_{p,b} = C_{p,f} \cdot (1 - c_m) + C_{p,p} \cdot C_{p,f}
\]

(1)

For \( T_1 < T < T_2 \) [15]:

\[
C_{p,b} = (1 - c_m) \cdot C_{p,f} + \frac{c_m \cdot L_h}{T_2 - T_1}
\]

(2)

Where,

\( T_1 = \) Temperature at which phase change starts

\( T_2 = \) Temperature at which phase change ends

GEOMETRY AND MESH

Microchannels containing staggered square and circular pins were selected for computational simulation. The height of the microchannel was taken as 500 µm comprising of pins of 300 µm in height with a base height of 200 µm (same for both square and circular pins). The width and length of the microchannel were 1 cm and 2 cm, respectively. It should be noted that the width of the microchannel does not have any significance on the results since symmetry was used to model only a section of the width, as shown in Figure 2. The assumption of symmetry is valid since there are about 50 pins along the width. The wall boundary condition at the bottom wall is either constant heat flux or constant wall temperature. The fluid and solid interfaces on both sides are specified as symmetric, having no heat flux, or flow perpendicular to the symmetry boundary condition as shown in Figure 1.

Figures 1-3 show the square pins geometry while Figures 4 shows circular pins geometry. Figure 5 shows the microchannel geometry without pins.
Figure 5. Diagram showing the geometry cross section modeled

<table>
<thead>
<tr>
<th>Table 1 Dimensions of Geometry</th>
<th>Description</th>
<th>Dimension (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H₁</td>
<td>Height of entire domain</td>
<td>500</td>
</tr>
<tr>
<td>H₂</td>
<td>Height of solid domain</td>
<td>200</td>
</tr>
<tr>
<td>S₁</td>
<td>Streamwise Pitch</td>
<td>100</td>
</tr>
<tr>
<td>S₇</td>
<td>Transverse Pitch</td>
<td>400</td>
</tr>
<tr>
<td>D</td>
<td>Diameter of circular pins</td>
<td>100</td>
</tr>
<tr>
<td>L</td>
<td>Side of square pins</td>
<td>100</td>
</tr>
</tbody>
</table>

GAMBIT (2.3) was used to create the microchannel geometries and to mesh the simulation domain. The geometries were meshed with hexahedral elements. Three different grid resolutions, 38x13x1400, 43x13x1400, 42x16x1634 were used to obtain a grid-independent solution. The results for these grids are presented in the result section.

**PARTICLE AND FLUID PROPERTIES**

Water was taken as the carrier fluid with PCM volume concentration of 15%. N-Eicosane was selected for the PCM particle with the properties given in Table 2. Water properties were assumed to be constant with temperature. The latent heat of PCM particles was assumed to be 230 kJ/kg, and the volumetric concentration in the bulk fluid was 15%. The exact melting point of N-Eicosane is 310 K. For the current work, the melting range of PCM (N-Eicosane) was assumed to be 309-310K for stability and convergence of the numerical solution.

<table>
<thead>
<tr>
<th>Table 2. Carrier fluid and PCM physical properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
</tr>
<tr>
<td>Water</td>
</tr>
<tr>
<td>PCM</td>
</tr>
</tbody>
</table>

Viscosity [20]:

\[
\frac{\mu_b}{\mu_f} = (1 - c - 1.16c^2)^{-2.5}
\]  

Density [20]:

\[
\rho_b = c \cdot \rho_p + (1 - c) \cdot \rho_f
\]  

Thermal Conductivity [20]:

\[
k_b = k_f \cdot \left(\frac{2^{2+\frac{\rho_p}{\rho_f}+2c}\left(\frac{\rho_p}{\rho_f}-1\right)}{2^{2+\frac{\rho_p}{\rho_f}-c}\left(\frac{\rho_p}{\rho_f}-1\right)}\right)
\]

**MODELING PROCEDURE**

Simulations were performed at three Reynolds numbers of 50, 66.7 and 83.3. First, a simple case with no heat transfer was run, and the velocity profile was extracted at the outlet. The extracted flow velocity profile then was applied as inlet boundary condition. Inlet temperature of the fluid was assumed to be constant (300 K). CHF (constant heat flux) and CWT (constant wall temperature) boundary conditions were specified at the bottom wall. For the case of CHF, a specified amount of heat flux (2 MW/m² for circular pins) is applied at the bottom wall of the solid domain. For the case of CWT, a specified temperature (350 K) is applied at the bottom wall of the solid domain.

**Governing Equations and solution procedure**

The governing equations for flow and heat transfer were solved using the commercial CFD solver Fluent (6.3.26). Fluent uses finite volume method that discretizes the domain, and solves for the fluid flow. The governing equations, continuity, momentum and energy equations [22] solved by Fluent are shown below respectively.

\[
\nabla \cdot (\bar{v}) = 0
\]  

\[
\nabla \cdot (\rho \bar{v} \bar{v}) = -\nabla p + \nabla \cdot (\bar{T})
\]  

\[
\nabla \cdot (\bar{v}(\rho E + p)) = k \nabla \cdot \nabla T + \nabla \cdot (\bar{T} \cdot \bar{v})
\]  

The default under relaxation factors for Continuity (0.3), Momentum (0.7) and Energy (1.0) equations were used to get the proper numerical solution. The residual limit for
convergence was set as $10^{-6}$ for all the equations. Initially, each solution was obtained with first-order discretization. Then the first order converged-solution was taken as an initial guess to get the converged-solution with second-order accuracy.

The Nusselt number and other parameters were based on the following equations [23].

\[ Nu_p = \frac{hD_{hp}}{k_b} \]  
\[ Nu_i = \frac{hD_{hi}}{k_b} \]

Where,

- $Nu_p$ = Nusselt number based on pin hydraulic diameter
- $Nu_i$ = Nusselt number based on inlet hydraulic diameter
- $D_{hp}$ = Pin hydraulic diameter, based on pin cross section given by,
  \[ D_{hp} = \frac{4A_p}{P_p} \]
- $A_p$ = Cross sectional Area of pin
- $P_p$ = Perimeter of pin cross section
- $D_{hi}$ = Inlet hydraulic diameter, based on inlet cross sectional area given by,
  \[ D_{hi} = \frac{4A_i}{P_i} \]
- $A_i$ = Cross sectional inlet Area
- $P_i$ = Perimeter of inlet cross sectional area

\[ h = \frac{q^*}{(T_w-T_b)} \]

Where,

\[ q^* = \text{Average wall heat flux along the periphery at a particular axial location} \]
\[ T_w = \text{Average wall temperature along the periphery at a particular axial location} \]
\[ T_b = \text{Mass weighted average of fluid temperature on the cross section at a particular axial location} \]

Non-dimensional Distance was calculated using:

\[ z^+ = \frac{z}{rRePr} \]

Where,

\[ r = D_h/2 \]

$D_h = \text{Either } D_{hi} \text{ or } D_{hp} \text{ depending upon the variable plotted against the dimensionless distance (explained below)}$

$Re = \text{Reynolds number for equation } 13 \text{ is either } Re_p \text{ or } Re_i$, depending upon the variable plotted against the dimensionless distance (explained below)

$Pr = \text{Prandtl number}$

$z = \text{Distance along flow direction from inlet}$

\[ Pr = \frac{c_{p,b} \mu_b}{k_b} \]

Reynolds numbers and Fanning friction factor were calculated using:

\[ Re_p = \frac{\rho V D_{hp}}{\mu} \]
\[ Re_i = \frac{\rho V D_{hi}}{\mu} \]

\[ f = \frac{D_{hi} \Delta p}{2L \rho V^2} \]

Where,

- $Re_p = \text{Reynolds number based on the pin hydraulic diameter}$
- $Re_i = \text{Reynolds number based on the inlet hydraulic diameter}$

It can be observed that the pin based hydraulic diameter for circular pins will simply be the pin diameter and for square pins, it will be the side length. Two ways of calculating the hydraulic diameter are considered. The pin hydraulic diameter is based on the pin cross section, whereas the inlet hydraulic diameter is based on the inlet cross section. The pin hydraulic diameter is more appropriate for flow across pins and is widely used in literature. But, for comparison with the no-pins case, it is necessary to use the inlet diameter as the pin hydraulic diameter cannot be calculated for no-pins case. The use of two hydraulic diameters results in different Reynolds numbers and Nusselt numbers for the same inlet velocity (or mass flow rate). The comparison of Reynolds number and the corresponding fully developed Nusselt number, for single phase fluid (SPF, water) between these two methods is presented in Table 3. It can be observed that both the Reynolds number and Nusselt number differ by a factor of 3 for the two methods of calculation. This is because; the inlet hydraulic diameter is 3 times the pin hydraulic diameter for both square and circular pin cases.

| Table 3 Reynolds number and Nusselt number based on inlet and pin hydraulic diameter |
|----------------------------------------|-----------------|-----------------|-----------------|-----------------|
| Reynolds Number | Circular Pins | Square Pins | No Pins |
| $Re_i$ | $Re_p$ | $Nu_i$ | $Nu_p$ | $Nu_i$ | $Nu_p$ | $Nu_i$ | $Nu_p$ |
| 150 | 50 | 22 | 7.5 | 12 | 4 | 3.6 | N/A |
| 200 | 66.7 | 30 | 10 | 12 | 4 | 3.6 | N/A |
| 250 | 83.3 | 40 | 13 | 12 | 4 | 3.6 | N/A |
The Nusselt number was calculated only at the cross sections at the midpoints of pins using equations (9), (10) and (13). The first five axial locations where the Nusselt number was calculated are shown in Figure 2 (vertical dash lines). At each location (cross section), a line is drawn that represents the fluid-solid interface at that location. The average heat flux transferred from the walls to the fluid can be calculated by taking the heat flux average along this line (\(q^\prime\)). Similarly, average wall temperature can be calculated along this line (\(T_w\)). Once the heat flux and average wall temperature are known, heat transfer coefficient can be calculated using the equation (13). Also, the use of \(Nu\) and \(Nu_p\) matters only for circular and staggered pin microchannels.

The dimensionless distance \((Z+)\) is based on the hydraulic radius. The choice of this hydraulic radius (among the two mentioned above) is based on the variable plotted against the dimensionless distance. For example, when plotting the variation of \(Nu\) with dimensionless distance, \(Dh\) is used to calculate the dimensionless distance. When plotting \(Nu_p\), \(Dh_p\) is used to calculate the dimensionless distance.

**RESULTS**

First, a single phase fluid (water) was considered in simple and straight microchannel geometries (circular and rectangular cross sections with tree different aspect ratios) without pins. The fully developed Nusselt number and pressure drop were compared with those available in the literature for validation of results for different boundary conditions. Table 4 and Table 5 below summarize the results for different aspect ratios. H2 boundary condition stands for constant axial and peripheral heat flux (peripheral temperature varies).

<table>
<thead>
<tr>
<th>Table 4. Friction factors for various geometries</th>
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</thead>
<tbody>
<tr>
<td>Geometry</td>
</tr>
<tr>
<td>Circular</td>
</tr>
<tr>
<td>1:2</td>
</tr>
<tr>
<td>1:4</td>
</tr>
<tr>
<td>1:8</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 5. Nusselt number with H2 boundary condition for single phase fluid (water)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Geometry</td>
</tr>
<tr>
<td>1:2</td>
</tr>
<tr>
<td>1:4</td>
</tr>
<tr>
<td>1:8</td>
</tr>
</tbody>
</table>

It can be clearly seen that the results match very well with the expected literature values \([24-25]\). Also, it can be observed that the Nusselt number for H2 boundary condition remains constant for various aspect ratios. This is because the 1:2 geometry is the limiting case for the H2 boundary condition \([25]\) as which can be found using equation (21) given below. From the equation, it can be found that the Nusselt number varies significantly for aspect ratios 0.5-1, but for aspect ratios lower than 0.5, it remains between 2.8 and 3.4.

Nusselt number for H2 boundary condition for such an aspect ratio is as follows \([25]\):

\[
Nu_{H2} = 8.235 \cdot (1 - 10.6044\alpha + 61.1755\alpha^2 - 155.1803\alpha^3 \\
+ 176.9203\alpha^4 - 72.9236\alpha^5)
\]

(21)

Where, \(\alpha\) = Ratio of the cross section dimensions (height/width)

**Grid Independence**

Three different grids of varying resolution were considered including 38x13x1400, 43x13x1400, and 42x16x1634. The overall solution varied within 4% for these three grids. Figure 6 shows that a 38x13x1400 grid was sufficient to provide grid-independent results. The results are for Reynolds number \((Re_p)\) of 50.

**Pressure drop with staggered pins**

Table 6 shows pressure drop values for different Reynolds numbers for the three geometries considered. It can be seen that pressure drop between inlet and outlet is highest for the square pins configuration.

<table>
<thead>
<tr>
<th>Table 6. Pressure drop between inlet and outlet</th>
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<tbody>
<tr>
<td>Geom.</td>
</tr>
<tr>
<td>1:2</td>
</tr>
<tr>
<td>1:4</td>
</tr>
<tr>
<td>1:8</td>
</tr>
</tbody>
</table>

**Validation of Results with PCM fluid flow**

PCM fluid flow in circular microchannel was simulated under H2 boundary condition and the results were compared with the experimental findings by Chen \([16]\) as seen in Figure 7.
Chen [16] calculated the bulk temperature of the PCM fluid across a cross section using simple energy balance, assuming that all the PCM melting process starts at a particular axial distance (independent of radial distance), and ends at a particular axial location.

Effect of PCM fluid on Nusselt number for different geometries

Figures 8 and 9 show the Nusselt number for square pins, circular pins, and no pins configurations using PCM fluid flow under CHF boundary condition. The dashed lines show the Nusselt number for SPF fluid (without phase change, constant specific heat) and the solid lines show the Nusselt number for PCM fluid (taking into account the phase change process). The results are for Reynolds number of 66.7. Figure 10 shows the fluid temperature variation for the CHF boundary condition for the same Reynolds number.

Figures 11 and 12 show the Nusselt number for three different geometries using PCM fluid flow under CWT boundary condition. The results are for Reynolds number of 66.7.
Figure 11. Nusselt number variation for square pins geometry under CWT boundary condition using PCM fluid

Figure 12. Nusselt number variation for circular pins geometry under CWT boundary condition using PCM fluid

Effect of Geometry on Nusselt number

Figures 13 and 14 show the Nusselt number for three different geometries at Reynolds number of 66.7.

Effect of Reynolds number on Nusselt Number

Figures 15 to 18 show the Nusselt number variation with Reynolds number for square and circular pins geometries.

Figure 13. Nusselt number variation for three geometry under CHF boundary condition using SPF fluid

Figure 14. Nusselt number variation for three geometry under CHF boundary condition using PCM fluid

Figure 15. Nusselt number variation for circular pins geometry under CHF boundary condition using SPF fluid

Figure 16. Nusselt number variation for square pins geometry under CHF boundary condition using SPF fluid
DISCUSSION

Figures 8 and 9 show how Nusselt number varies along the axial direction under CHF boundary condition for microchannels of two geometric configurations. The dashed line shows the value for bulk flow (PCM + water) without any phase change process taking place. This represents a single phase fluid (SPF) case which yields the same Nusselt number as water under fully developed flow conditions. The solid lines show the Nusselt number for PCM fluid undergoing phase change which is taken into account by the specific heat model. It can be seen from the plots that the phase change process significantly enhances the Nusselt number by holding back fluid temperature as depicted in Figure 10. Furthermore, circular pins result in higher Nusselt numbers than in square pins. Heat transfer enhancement is more significant in the circular pin case than in the square pin case because the wake region (where fluid is stagnant) behind each pin in smaller in circular pins as seen in Figures 19 and 20.

Figure 11 and 12 show the same variation for CWT boundary condition. Comparing Figures 8 and 9 with Figures 11 and 12, it can be seen that the phase change process results in higher Nusselt number in the case of CHF boundary condition compared to CWT boundary condition. When using the CHF boundary condition, the wall temperature rise continuously while in the CWT case, the surface temperature remains constant. Therefore, the driving potential for phase change becomes stronger with distance in the CHF case than in the CWT case. This is consistent with results obtained for straight (no pin) microchannels as discussed by the authors in previous publication [28].

Figures 13 and 14 show the variation in Nusselt number for the three geometries considered in the present study. The Nusselt number shown in the plots was based on inlet hydraulic diameter. The circular pins case exhibits the highest Nusselt number with and without phase change. A similar observation was made for single phase fluids by Soodphakdee et al. [30]. The enhancement in heat transfer is due to the difference in flow structure especially in the wake region behind the pins.
The wake region behind the circular pins (where fluid is stagnant) is smaller than in the square pin case for the same Reynolds number as can be observed by comparing Figure 19 and Figure 20.

Figures 15-18 show the variation in Nusselt number for different Reynolds numbers for circular and square pins geometries, respectively. From the figures, it can be observed that the Nusselt number converges to the same value independent of Reynolds number in the case of square pins (Figures 16 and 18); but, in the case of circular pins (Figures 15 and 17), the Nusselt number converges to a different value for each Reynolds number value. In fact, according to Figures 15 and 17, it is observed that when the Reynolds number increases, the Nusselt number increases as well. The effect of laminar flow Reynolds number on heat transfer in the presence of circular pins needs to be further investigated. For the Reynolds numbers considered, it has been suggested by some authors in the literature that periodic vortex shedding starts at a Reynolds number range between 55 and 80 depending on the spacing between the pins [31-32]. Hence, an unsteady periodic flow analysis might explain the reasons for such difference in Nusselt number with Reynolds number. The authors are currently considering unsteady periodic flow simulations.

From Table 6, it can be observed that the pressure drop increases with an increase in Reynolds number for any specific geometry. Also, for a given Reynolds number, pressure drop is highest for the square pin cases. This can be explained by observing Figure 19 and Figure 20, showing the flow around the square and circular pins for a pin Reynolds number of 50. It can be observed that the wake region behind the square pin is larger than the wake region behind the circular pins. Analysis has shown that larger wake result in higher pressure drop [27], and hence using the square pins should result in higher pressure drop.

CONCLUSIONS

The following conclusions can be drawn from the results of this study.

- Higher pressure drop is observed in the case of square pins compared to circular pins.
- The specific heat model predicts the heat transfer behavior accurately when compared with experimental results.
- Nusselt number is higher for circular pins compared to square pins in the Reynolds number range considered.
- The effect of the phase change process on heat transfer is found to be more significant in the case of circular pins compared to square pins. This can be established by observing the increase in the Nusselt number with PCM fluid compared to SPF fluid for the same geometry.
- The effect of phase change process on Nusselt number is found to be higher under CHF boundary condition compared to CWT boundary condition for all the geometries considered.
- The axial distance required to obtain a steady Nusselt number is almost same for square and circular pins geometries.
- A maximum increase in Nusselt number of about 14% is found when using PCM fluid compared to SPF, in the case of circular pins at a Reynolds number of 66.7

Future work will consider the effect of vortex shedding on Nusselt number and pressure drop at higher laminar flow Reynolds number within the periodic flow regime.

ACKNOWLEDGEMENTS

The authors would like to thank the National Science Foundation (NSF) STTR/SBIR program and Dr. Thies of Thies Technology, Inc. for their support.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>c</td>
<td>volumetric concentration of PCM</td>
</tr>
<tr>
<td>c_m</td>
<td>mass concentration of PCM</td>
</tr>
<tr>
<td>C_p</td>
<td>Specific heat of bulk fluid (slurry)</td>
</tr>
<tr>
<td>f</td>
<td>Friction factor</td>
</tr>
<tr>
<td>h</td>
<td>Convective heat transfer coefficient</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity</td>
</tr>
<tr>
<td>L</td>
<td>Distance along the axial direction</td>
</tr>
<tr>
<td>L_a</td>
<td>Latent heat of PCM particles</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>p</td>
<td>Pressure</td>
</tr>
<tr>
<td>PCM</td>
<td>Phase Change Material (fluid)</td>
</tr>
<tr>
<td>SPF</td>
<td>Single Phase Fluid</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
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<td>Density</td>
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<tr>
<td>( )_f</td>
<td>Property of the carrier fluid</td>
</tr>
<tr>
<td>( )_p</td>
<td>Property of the PCM particle (N-Eicosane)</td>
</tr>
<tr>
<td>( )_b</td>
<td>Property of the bulk fluid (PCM + water)</td>
</tr>
</tbody>
</table>

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