Three dimensional numerical study of heat-transfer enhancement by nano-encapsulated phase change material slurry in microtube heat sinks with tangential impingement

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ABSTRACT

This paper presents a three-dimensional model describing thermal and hydrodynamic characteristics of a Microtube heat sink with tangential impingement with nanoencapsulated phase change materials (NEPCM) slurry as coolant. In this study, octadecane for NEPCM and polyalphaolefin (PAO) is used as a base fluid. The continuity, momentum, and energy equations are solved using a finite volume method. The model is validated by comparing results with available data in the literature. The effects of dominant parameters including mass concentration and melting range of NEPCM as well as Reynolds number on temperature uniformity, thermal resistance, Nusselt number, pressure drop and generated entropies in the system are investigated. Results indicated that adding NEPCM to base fluid leads to considerable heat transfer enhancement. However, using NEPCM slurry as coolant has also induced drastic effects on the pressure drop that increases with mass concentration and Reynolds number. It was found that that an increase in nanoparticles mass concentration, inlet Reynolds number and melting range of NEPCM, results in a higher Nusselt number, better temperature uniformity and lower thermal resistance. Furthermore, the effects of different parameters on slurry entropy production are demonstrated. It is found that generated total entropy decreases with increasing mass concentration and Reynolds number.

1. Introduction

Due to the miniaturization of electronic components or systems, problems associated with overheating of electronic components have increased drastically. Most electronic components or systems, from microprocessors to high power converters generate heat which is necessary to reject it for optimum operation so thermal management of electronic devices becomes more and more important. Traditional microchannel heat sinks using conventional coolants for computer chip cooling are reaching their maximum potential. The limit of the heat transport capability of a microchannel heat sink with conventional fluids such as water or ethylene glycol coolants will be exceeded, leading to the use of new types of heat sinks with a new class of coolants such as a mixture of nano-encapsulated phase change material to dissipate the heat from the heat source. The microtube heat sink with tangential impingement represents a new class of heat sinks which results in distributing thermal and hydrodynamics boundary layers in microchannels. Many studies have been performed in recent years, which show how tangential impingement injection of fluid inside the channels enhances heat transfer significantly [1–4]. Lelea [5] optimized geometry of tangential microchannel heat sink with straight circular microchannels. The cross-section of the injection channel was in the form of rectangular shape and positioned tangentially to the tube axis. In Lelea’s second paper [6], he studied the thermal and hydrodynamic of a tangential characteristic of heat sink with multiple inlet jets. The results showed that at a fixed pumping power, the minimum temperatures are lower for a single inlet compared with multiple inlet jets. He also showed where a microchannel heat sink with multiple inlet jets has a lower temperature difference on the microtube’s bottom surface between the inlet and outlet. Recently, Shalchi and Seyf [7] investigated the effect of nanofluid on thermal performance of a tangential microchannel heat sink. Results showed that with decreasing nanoparticles’ diameter and increasing volume fraction, the system’s thermal performance increases significantly.

With the high heat power generated in modern electronic components or systems, more advanced coolants need to be used. Recently, a new coolant that utilizes nanoencapsulated phase change material (NEPCM) in a heat transfer fluid such as water...
has been proposed. In general, NEPCM particles are composed of a core of paraffin wax phase change material (PCM) with a wall around it which is made of cross-linked polymer. The wall material is usually 14–20% of the total capsule mass and is sufficiently flexible to accommodate volume changes that accompany solid/liquid phase change [8]. The latent heat of the NEPCM particles provides an approach to store energy and enhance heat transfer when the particles experience phase change. In this approach to thermal management, high effective thermal conductivity of mixture due to micro-convection induced by nanoparticles is coupled with temperature and melting fraction distributions. To the best of the authors’ knowledge, there is no experimental, analytical and numerical work to study thermal performance of microtube heat sink with tangential impingement using NEPCM slurry as coolant, and the present study is the first attempt that surveys these types of coolants in microtube heat sink with tangential impingement. Therefore, this study offers a new direction for ultra-high cooling performance of microtube heat sinks.

2. Mathematical modeling and governing equations

The schematic diagram of the heat sink is shown in Fig. 1. Tangential microchannel heat sinks (TMHS), coolant or NEPCM slurry flows in tangentially through the gaps on the top surface of the heat sink at a specified temperature and velocity, and its temperature increases as it moves through the channels and reaches the melting temperature of phase change material (PCM). There is no mass transfer between the carrier fluid and capsules because when PCM melts inside the capsules; the melted PCM will not mix with the carrier fluid because it remains contained in the capsules. The carrier fluid exhibits lower temperature change when the PCM melts. Different geometries for inlet cross-section of system can be used, but in the present study, the rectangular inlet cross-section is adopted based on previous studies in literature [5,6]. The NEPCM slurry consists of octadecane as nanoparticle and polyalphaolefin (PAO) as base fluid [26]. The total height, length, width and number of heat sink channels are 1.5 mm, 1 cm, 1 cm and 9, respectively. The area of inlet cross-section and tube diameter are 0.45 mm² and 900 μm, respectively. The center of tube is 900 μm away from the bottom surface of the heat sink. Because of the heat transfer model will be used to determine the effect of NEPCM mass concentration and melting range as well as inlet Reynolds number on the temperature uniformity, thermal resistance, pressure drop, and Nusselt number as well as generated entropies in the system. This paper also summarizes the effects of certain parameters on the temperature and melting fraction distributions. To the best of the authors’ knowledge, there is no experimental, analytical and numerical work to study thermal performance of microtube heat sink with tangential impingement using NEPCM slurry as coolant, and the present study is the first attempt that surveys these types of coolants in microtube heat sink with tangential impingement. Therefore, this study offers a new direction for ultra-high cooling performance of microtube heat sinks.
sink’s geometry, only one symmetrical microtube region is selected within the heat sink for computational domain.

The present model is based on the following assumptions:

1. The coolant flow is incompressible, laminar.
2. Radiation and compressibility effects are very small in the current investigation so are not considered in this study.
3. The particle concentration is less than 0.3; therefore, the fluid is Newtonian [18].
4. The melting range of NEPCM particles is between $T_1$ and $T_2$.
5. The micro-convection caused by the particle–wall interactions, particle–fluid and particle–particle is lumped together, and its effect is accounted by an effective thermal conductivity.
6. The difference in the particle and fluid velocities is negligible, i.e., the particles follow the fluid without any lag [26].
7. The particle distribution is homogeneous so that the bulk properties are assumed constant except for thermal conductivity and heat capacity, which are functions of space and temperature, respectively [26].
8. The particle melts instantaneously; there is no temperature gradient inside the particle [26].

9. NEPCM–PAO slurry properties are functions of NEPCM concentration and slurry temperature.
10. The effect of the particle depletion layer is negligible. The particle depletion layer is of the order of the particle radius if the channel size to particle size is large [28,29].
11. The shape of the encapsulated particles is spherical.
12. The effect of the shell material is neglected.
13. The NEPCM–PAO slurry enthalpy consists of two parts: sensible heat and PCM latent heat.

The governing equations of flow and heat transfer inside a Microtube heat sink with tangential impingement and temperature dependent material properties can be expressed as follows:

**Continuity equation:**

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0$$

**X-Momentum equation:**

$$\rho_{\text{eff}} \left( u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = - \frac{\partial p}{\partial x} + \frac{\partial}{\partial y} \left( \mu_{\text{eff}} \frac{\partial u}{\partial y} \right) + \frac{\partial}{\partial z} \left( \mu_{\text{eff}} \frac{\partial u}{\partial z} \right) + \Phi$$

**Y-Momentum equation:**

$$\rho_{\text{eff}} \left( u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = - \frac{\partial p}{\partial y} + \frac{\partial}{\partial x} \left( \mu_{\text{eff}} \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial z} \left( \mu_{\text{eff}} \frac{\partial v}{\partial z} \right)$$

**Z-Momentum equation:**

$$\rho_{\text{eff}} \left( u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = - \frac{\partial p}{\partial z} + \frac{\partial}{\partial x} \left( \mu_{\text{eff}} \frac{\partial w}{\partial x} \right) + \frac{\partial}{\partial y} \left( \mu_{\text{eff}} \frac{\partial w}{\partial y} \right)$$

**Energy:**

$$\rho_{\text{eff}} C_{p_{\text{eff}}} \left( u \frac{\partial T_{s}}{\partial x} + v \frac{\partial T_{s}}{\partial y} + w \frac{\partial T_{s}}{\partial z} \right) = \frac{\partial}{\partial x} \left( k_{s} \frac{\partial T_{s}}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_{s} \frac{\partial T_{s}}{\partial y} \right) + \frac{\partial}{\partial z} \left( k_{s} \frac{\partial T_{s}}{\partial z} \right) + \Phi_{T}$$

where $u$, $v$, and $w$ are velocity components in $x$, $y$, $z$ directions, respectively. $p$ and $T$ are pressure and temperature, respectively. $\rho_{\text{eff}}$ and $C_{p_{\text{eff}}}$ are the density and heat capacitance of the NEPCM slurry, respectively. Heat transfer in TMHS is a conjugate problem which combines heat conduction in the solid and convective heat transfer to the flowing fluid. The energy equation in solid region is expressed as:

$$\frac{\partial}{\partial x} \left( k_{s} \frac{\partial T_{s}}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_{s} \frac{\partial T_{s}}{\partial y} \right) + \frac{\partial}{\partial z} \left( k_{s} \frac{\partial T_{s}}{\partial z} \right) = 0$$

Index $s$ and $f$ shown in Eqs. (5) and (6) respectively refer to solid material of heat sink and fluid region of computational domain. $k_{s}$ is thermal conductivity of solid material. $\Phi_{T}$ shown in Eq. (5) is viscous dissipation term and it represents the time rate at which energy is being dissipated per unit volume through the action of viscosity. For an incompressible flow, it is written as follows:

$$\Phi = \sqrt{2 \left[ \left( \mu_{\text{eff}} \frac{\partial u}{\partial x} \right)^{2} + \left( \mu_{\text{eff}} \frac{\partial v}{\partial y} \right)^{2} + \left( \mu_{\text{eff}} \frac{\partial w}{\partial z} \right)^{2} + \left( \mu_{\text{eff}} \frac{\partial u}{\partial y} + \mu_{\text{eff}} \frac{\partial v}{\partial x} \right)^{2} + \left( \mu_{\text{eff}} \frac{\partial u}{\partial z} + \mu_{\text{eff}} \frac{\partial w}{\partial x} \right)^{2} + \left( \mu_{\text{eff}} \frac{\partial w}{\partial y} + \mu_{\text{eff}} \frac{\partial v}{\partial z} \right)^{2} \right]}$$
The thermal conductivity of silicon can be calculated using the following equation [34,35]

\[ k_{\text{silicon}} = 290 - 0.4T \]  

(8)

It is assumed the heat goes to computational domain only from the bottom wall of the heat sink so a constant heat flux is applied at the heated wall \( (q_w = 10,000 \text{ W/m}^2) \). A constant and uniform velocity and temperature is applied at the inlet of computational domain. The boundary conditions for all surfaces are described as follows:

\[ \text{Inlet condition (} y = H): \]

\[ u = w = 0, \quad v = -v_{in}, \quad T_f = T_{in} \]  

(9)

\[ \text{Outlet condition (} Z = L): \]

\[ \frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = \frac{\partial w}{\partial z} = \frac{\partial T_f}{\partial z} = 0 \]  

(10)

\[ \text{Symmetry condition (} Z = 0): \]

\[ \frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = 0 \quad \text{and} \quad u = v = 0 \]  

(11)

\[ \text{Symmetry condition (} x = \pm w_{in}/2): \]

\[ \frac{\partial T_s}{\partial x} = \frac{\partial T_f}{\partial x} = 0 \]  

(12)

All external walls are insulated except for the bottom wall, one in contact with chip \( (Y = 0) \):

\[ q_w = k_s \frac{\partial T}{\partial y} \]  

(13)

Solid–liquid interfaces:

\[ k_{\text{eff}} \frac{\partial T_f}{\partial n} = k_s \frac{\partial T_s}{\partial n} \]  

(14)

where \( n \) is normal vector drawn outward the boundary.

In order to satisfy the energy conservation across solid interfaces, the grids coincide on the fluid–solid boundaries heat sinks and the conservative interface flux condition for heat transfer is adopted at the interface between solid and fluid:

\[
\left( \frac{\partial T}{\partial y} \right)_f = \left( \frac{\partial T}{\partial y} \right)_s \Rightarrow k_s \frac{\partial T_s}{\partial y} - k_f \frac{\partial T_f}{\partial y} = T_{s,f} - T_f = T_{s,s} = \frac{c_k T_s + k_f T_f}{c_k + k_f} 
\]  

(15)

\( T_{s,f} \) is the temperature at the common boundary and is applied for calculation of temperature gradients \( (k \frac{\partial T}{\partial y}) \) within each elementary grid volume \( (s,f) \). \( c \) is the ratio of the normal distance between the first cell node and the solid-fluid boundary of the solid \( (dy_s) \) and fluid \( (dy_f) \) side (Fig. 2):

\[ c = \frac{dy_f}{dy_s} \]  

(16)

3. Bulk properties of slurry

In order to calculate the heat transfer rate of NEPCM slurry, thermal properties of NEPCM slurry such as heat capacity, thermal conductivity and viscosity should be calculated. In this study, it is assumed that the slurry consists of PAO and NEPCM with octadecane as the phase change material with melting point of 301.9 K. The properties used are summarized in Table 1. The dynamic viscosity of slurry calculated using following equation [30,31]:

\[ \frac{\mu_{\text{eff}}}{\mu_{\text{PAO}}} = \left( 1 - \xi - 1.16\xi^2 \right)^{-2.5} \]  

(17)

where \( \xi \) is volume concentration of NEPCM and \( \mu_{\text{PAO}} \) is the PAO viscosity, which is a function of temperature.

The static thermal conductivity of the suspension or thermal conductivity of slurry at rest (no shear rate) can be expressed as [21]:

\[ k_{\text{eff}} = k_{\text{PAO}} \frac{2 + \frac{k_s}{k_{\text{PAO}}} + \frac{2\xi (\frac{k_s}{k_{\text{PAO}}} - 1)}{1 - \xi}}{2 + \frac{k_s}{k_{\text{PAO}}} - \xi (\frac{k_s}{k_{\text{PAO}}} - 1)} \]  

(18)

Due to particle–particle, particle–liquid and particle–wall interaction, the effective thermal conductivity of slurry under motion increases. These interactions have the major role in enhancement of thermal conductivity of mixture. The effective thermal conductivity of slurry flow is specified by the following correlation [26]:

\[ k_{\text{eff}} = f = 1 + B \xi Pe_p^m \]  

(19)

where \( \xi \) is volume concentration of NEPCM and \( k_{\text{PAO}} \) is the PAO viscosity, which is a function of temperature.

The particle Peclet number is defined as

\[ Pe_p = \frac{ed_p^2}{\nu_{\text{PAO}}} \]  

(20)

where \( \nu_{\text{PAO}} \) is thermal diffusivity of PAO. The shear rate is a function of all the spatial coordinates and corresponding velocities. The magnitude of the shear rate can be calculated using the following equation:

\[ e = \frac{1}{2} \sum_i \sum_j \gamma_{ij} \gamma_{ij} \]  

(21)

where \( \gamma \) is the shear rate. The effective thermal conductivity correlation given above shows that the thermal conductivity is strongly dependent on (i) the shear rate, which can be increased by decreasing the conduction dimension or increasing slurry flow rate, and (ii) particle size due to the interactions (drag force, lift force, and virtual mass) between the liquid and the particles. Slurry specific heat capacity and density are given by

\[ C_{p,\text{eff}} = (1 - C_m)C_{p,\text{PAO}} + C_m C_{p,\text{PAO}} \]  

(22a)

\[ \rho_{\text{eff}} = (1 - C_m)\rho_{\text{PAO}} + C_m \rho_p \]  

(22b)

where \( C_{p,\text{PAO}}, \rho_{\text{PAO}}, C_m \) and \( \rho_p \) are the specific heat and density of PAO, mass concentration or loading fraction of slurry and density of NEPCM, respectively. According to Alisetti and Roy [23], the difference between using various profiles for calculating the specific heat of PCM is less than 4%. Therefore, in this study we used the sine profile to represent the NEPCM particle specific heat (Fig. 3) of

\[ C_{p,\text{PAO}} = C_{p,\text{PAO}} + \left\{ \pi \left( \frac{h_{\text{eff}}}{T_{m'}} - C_{p,\text{PAO}} \right) \cdot \sin \pi \left( \frac{T - T_f}{T_{m'}} \right) \right\} \]  

(23)
where $T_{mr}$ is the melting range ($T_{mr} = T_2 - T_1$). For temperatures higher or lower than melting range the value of specific heat of particle is equal to $C_p$, and for temperatures in melting range, the specific heat of particle is calculated using above equation.

### 4. Numerical solution and validation

In the present study, a validated finite volume code has been employed as the numerical solver [7]. The three-dimensional governing equations are discretized by applying a finite volume method in which conservation laws are applied over finite-sized control volumes around grid points, and the governing equations are then integrated over the volume. The collocated grid arrangement is adopted where all variables are stored at the control volume geometrical center. Quick scheme is used to discretize convection/diffusion terms in momentum and energy equations. The numerical simulation is accomplished by using SIMPLE algorithm. In this technique, using a guessed pressure field, the velocity components in three directions is first calculated from the Navier–Stokes equations, and then to satisfy the continuity equation, the pressure and velocities are corrected. The numerical solution is regarded as convergent at an iteration in which the summation of absolute values of relative errors of temperature, velocity components and pressure reach $10^{-6}$, $10^{-7}$ and $10^{-5}$, respectively.

The computation was carried out using hexahedral volume elements in both solid and fluid regions of computational domain. The relatively fine grids are used in the regions of the boundary layers where the gradients of velocity and temperatures are steeper. Fig. 4 illustrates the detailed generated meshes in each section of computational domain. For grid independency test, three different grid systems are investigated which include about 619,059, 896,548, and 1,104,584 cells, respectively. The predicted temperature uniformity and pressure drop for the three grid systems are shown in Table 2. From the table, it is obvious that the change of the temperature uniformity and pressure drop respectively is less than 0.32% and 0.64% among the third and second grid systems. Therefore, for the present study, the final grid number is selected as about 1,104,584. It is important to note that the grid independency was performed for worst Reynolds scenario, i.e., highest Reynolds number.

To the knowledge of authors, there are no available data in the published literature related to study of Microtube heat sink with tangential impingement in laminar flow regime and with NEPCM slurry as coolant. Furthermore, since the concept of Microtube heat sink with tangential impingement is novel, there is no experimental study in the literature on this type of heat sink. Therefore, in order to validate the reliability of the numerical method being used, the numerical simulation is conducted for a tangential

<table>
<thead>
<tr>
<th></th>
<th>Density (kg/m$^3$)</th>
<th>Viscosity (Pa s)</th>
<th>Specific heat (J/kg K)</th>
<th>Thermal conductivity (W/m K)</th>
<th>Latent heat (J/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PAO</td>
<td>784</td>
<td>$4.45 \times 10^{-3}$</td>
<td>2242</td>
<td>$143 \times 10^{-3}$</td>
<td>–</td>
</tr>
<tr>
<td>Octadecane</td>
<td>774</td>
<td>–</td>
<td>2180</td>
<td>0.15</td>
<td>$244 \times 10^{-3}$</td>
</tr>
</tbody>
</table>

**Fig. 3.** Specific heat of NEPCM as a function of temperature.

**Fig. 4.** Computational domain and detailed meshes in different regions.
microchannel heat sink with the same geometrical configurations as presented in Lelea [32]. Lelea numerically analyzed the performance of microtube heat sink with tangential impingement with water as coolant. The length of microchannel, tube diameter and inlet cross-section are 1 cm, 300 μm, 1 mm, and 50 μm, respectively. In the simulation, a constant heat flux of 100 W/cm² was applied at the bottom of heat sink, while all other walls were insulated. The inlet mass flow rate and temperature were 110 kg/s and 293 K, respectively. Further numerical details can be found in the original paper by Lelea [32]. The predicted results are compared with the numerical results from Lelea. The temperature distribution at bottom wall of heat sink is shown in Fig. 5 which shows the maximum discrepancy between the predicted results with the one for Lelea [32] as 0.5%.

5. Prediction procedure

The definitions of Re number is as follows:

\[ Re = \frac{4}{\pi} \frac{(M/2)}{D \cdot \mu} \]  

(24)

where \( M \) and \( D \) are the inlet mass flow rate of single microtube and tube diameter, respectively. The pressure drop can be calculated as follows:

\[ \Delta P = \int_{A_{in}} P \, dA - \int_{A_{out}} P \, dA \]  

(25)

where \( A_{in} \) and \( A_{out} \) are the areas of outlet and inlet of computational domain. One of the main parameters for evaluating the thermal performance of heat sink is thermal resistance which can be defined as

\[ R_{th} = \frac{T_{w,\text{max}} - T_{w,\text{min}}}{Q} \]  

(26)

where \( T_{w,\text{max}} \) and \( T_{w,\text{min}} \) are the maximum and minimum temperature of heat sink at bottom wall, respectively. \( Q \) is dissipated heat rate and \( T_{in} \) is the coolant inlet temperature. In MCHS, the generated heat by semiconductor chips results in high temperature rises along the microchannels because the heat is carried out by a relatively small amount of coolant. High temperature rise causes the non-uniform temperature distribution on the chip. Non-uniform wall temperature can produce destructive thermal stresses in packages and elements due to the differences in thermal expansion coefficient, which poses reliability concerns to the devices. Furthermore, the spatial temperature gradient may adversely affect the stability and performance of chips. The temperature uniformity is often quantified by

\[ \Delta T_{u} = T_{w,\text{max}} - T_{w,\text{min}} \]  

(27)

The overall Nusselt number can be calculated from the following equation [26]

\[ Nu = \frac{q \cdot D}{(T_{w} - T_{in})k_{f}} \]  

(28)

where \( T_{w} \) and \( T_{in} \) are average temperature of wall and inlet temperatures, respectively.

\[ T_{w} = \frac{1}{A_{w}} \int_{A_{w}} T \, dA_{w} \]  

(29)

In this paper, the entropy generation which characterizes the irreversible behavior of system is used to optimize a device performance by evaluating best operational parameters as well as fluid-properties. In the flow of viscous fluid inside the microtube heat sink with tangential impingements, the entropy generation must consider irreversibilities caused by heat flow and fluid friction. The entropy generation rate per unit volume can be written as [33]

\[ S^{\text{gen}} = S^{\text{gen}}_{\text{irr}} + S^{\text{gen}}_{\text{visc}} \]  

(30)

In the above equation, the first term expresses for irreversibilities due to heat transfer and the second term is related to viscous dissipation in the fluid which can be defined as follows:

\[ S^{\text{gen}}_{\text{irr}} = \frac{k_{f}}{T} \left[ \left( \frac{\partial T}{\partial x} \right)^2 + \left( \frac{\partial T}{\partial y} \right)^2 + \left( \frac{\partial T}{\partial z} \right)^2 \right] \]  

(31)

\[ S^{\text{gen}}_{\text{visc}} = \left\{ 2 \left[ \mu_{\text{eff}} \frac{\partial u}{\partial x} \right]^2 + \left[ \mu_{\text{eff}} \frac{\partial v}{\partial y} \right]^2 + \left[ \mu_{\text{eff}} \frac{\partial w}{\partial z} \right]^2 \right\} + \left( \mu_{\text{eff}} \frac{\partial u}{\partial x} + \mu_{\text{eff}} \frac{\partial v}{\partial y} \right)^2 + \left( \mu_{\text{eff}} \frac{\partial w}{\partial z} \right)^2 \]  

(32)

It is worth mentioning that for all the simulations presented in this paper, the entropy source due to frictional effect is much smaller than the one due to heat transfer, i.e., \( S^{\text{gen}}_{\text{visc}} \ll S^{\text{gen}}_{\text{irr}} \).
6. Results and discussion

In order to study the influence of Reynolds number and mass concentration of NEPCM slurry on the heat transfer characteristics of microtube heat sink with tangential impingement, a comparative investigation for different inlet Reynolds numbers and mass concentrations of NEPCM slurry was performed. Fig. 6 shows the contour of temperature distribution for both pure PAO and NEPCM slurry as coolant in a plane which possesses through center of tube and parallel to Y axis. As expected the temperature is higher at bottom wall where the heat from chip comes in the heat sink. It can be seen that the temperature distributions in solid and liquid phases are non-uniform, and using NEPCM slurry as coolant will cause the bulk temperature of coolant as well as temperature of solid phase to decrease which means more energy transfers through the fluid with little increase in coolant temperature. It is clear that increasing the mass concentration of slurry intensifies the reduction of wall temperature. Such behavior is obviously due to the fact that with increasing mass concentration, the specific heat of coolant increases considerably which results in more effective cooling.

Another important result shown in Fig. 6 is that the thermal boundary layer growth for NEPCM slurry coolant is slower than that of pure PAO; thus, the entry length is longer for slurry flow.

This is mainly due to the NEPCM particles latent heat, which acts as a heat sink and slows the growth of thermal boundary layer along the heat sink.

In microtube heat sink with tangential impingement, the flowing fluid heated gradually along the length of microtube. Therefore, when it arrives at the end of heat sink, it has already been heated to a relatively high temperature. It is interesting to note that the fluid region of computational domain can be divided into three distinct regions according to temperature of coolant. In the first region which is near the inlet of device, the PCM is at solid state and has a constant and approximately low NEPCM solid phase heat capacity; hence, the main mechanism for heat transfer in this region is the high temperature difference between walls of microchannel and coolant—not the heat capacity of coolant. In the second region, which is somewhere in the middle of heat sink, the PCM melts inside the NEPCM particles and coolant has a heat capacity equal to the summation of the latent and sensible heat of the NEPCM particles so the main mechanism of heat transfer enhancement in this region is a high effective heat capacity of coolant. In the third region, which is near the end of device, the temperature of coolant is very high, and the PCM is completely melted; in addition, the specific heat has a constant value equal to the liquid specific heat of the NEPCM slurry so the minimum
heat transfer enhancement is occurring in this region. Therefore, the cooling capability of coolant in the computational domain first increases and then decreases causing a non-uniform temperature distribution at the bottom wall of heat sink. It is worth mentioning that the formation of hotspots at the base plate of device is mainly due to various cooling capability of coolant along the length of heat sink. Therefore, in order to evaluate the performance of a heat sink, both thermal resistance and temperature uniformity should be evaluated. The effect of Reynolds number and mass concentration on thermal resistance and temperature uniformity of heat sink are presented in Fig. 7. It is clear that for the pure PAO case, thermal resistance and temperature uniformity is high while for the cases using NEPCM slurry as coolant, the cooling performance increases with increasing mass concentration. The improved performance mainly occurs due to the enhanced thermal properties of slurry coolant especially specific heat. Furthermore, it is obvious that with increasing Reynolds number, the thermal resistance and temperature uniformity decreases because increasing the Reynolds number leads to decreasing thermal boundary layers’ thickness in microtube and consequently yields a higher heat transfer coefficient. It is interesting to note that higher Reynolds numbers increases convection heat transfer not only because of higher

Fig. 7. Effect of mass concentration on thermal resistance and temperature uniformity.

Fig. 8. Effect of mass concentration on Nusselt number and pressure drop.
inertia but also due to the NEPCM slurry absorbing the same amount of heat with less temperature rise.

The pressure drop between inlet and outlet of the heat sink and Nusselt number is depicted in Fig. 8. As seen, both pressure drop and Nusselt number increase dramatically with the increment of the Reynolds number and mass concentration. The enhancement of pressure drop with increasing mass concentration of slurry is considerably high at higher Reynolds numbers. Therefore, the high pressure drop for the NEPCM slurry coolant at higher Reynolds numbers should be given careful consideration in designing heat sinks. It is worth mentioning that the sensitivity of the pressure drop to mass concentration of NEPCM is related to the increased viscosity at higher mass concentrations because high values of \( C_m \) leads the fluid to become more viscous which causes an increased pressure drop. The reason for higher values of Nusselt number at higher mass concentration is due to the fact that with increasing mass concentration of NEPCM, the thermal conductivity and capability of heat absorption by the coolant increase which causes yields a better heat transfer coefficient.

The influence of NEPCM mass concentration on entropy generation rate in the TMHS is shown by Figs. 9 and 10. It is clear that compared to pure PAO, NEPCM slurry coolant has a smaller total entropy generation rate, and with increasing mass concentration, the total heat transfer entropy generation rate decreases. Therefore, using NEPCM slurry may lead to improved heat transfer performance for greater system efficiency when compared to pure

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**Fig. 9.** Thermal and frictional entropy generation rate as a function of Reynolds number for different mass concentration.

**Fig. 10.** Total entropy generation rate as a function of Reynolds number for different mass concentration.
PAO. Furthermore, it is obvious that the frictional contribution of entropy generation rate increases with increasing mass concentration, which means that the hydrodynamic efficiency of the system decreases with increasing mass concentration, but the amount of enhancement in entropy is very small, especially at lower Reynolds numbers.
Fig. 11 shows the effect of melting range on temperature contour. It can be seen that the wall temperature is not same when using NEPCM slurry with different melting ranges and higher melting ranges lead to lower temperature in solid region of heat sink and smaller values of temperatures at hot spots. Therefore, as will be shown, an additional benefit of using NEPCM slurry with higher melting range is the decrease of temperature difference between the hotspots and the low temperature areas, thereby improving the temperature uniformity of heat sink. It is important to mention that the melting range depends on the purity of PCM; hence, it can be concluded that the purity of PCM helps increase the heat absorbing capacity of the bulk fluid and also the heat transfer.

Fig. 12 illustrates the effect of melting range on thermal resistance and temperature uniformity of heat sink. Although all the results have shown a similar trend regarding the decreasing thermal resistance and temperature uniformity with respect to the Reynolds number, it is obvious that melting range of PCM has significant effect on thermal resistance while its effect on temperature uniformity is not significant. It is clear that increasing the melting range decreases both thermal resistance and temperature uniformity. Furthermore, according to Fig. 12, the changes in thermal resistance and temperature uniformity are not pronounced for higher Reynolds number, i.e., \( Re = 600 \). This also holds true for other mass concentrations, e.g., for \( C_m = 0.1 \) and 0.2. Fig. 13 presents the relation of the pressure drop and Nusselt number with the Reynolds number. Compared with NEPCM slurry with lower melting range, the Nusselt number enhancement for the slurry with higher melting range is increased by 46% for \( Re = 600 \). It is
interesting to note that the NEPCM with higher melting range can enhance the heat transfer in the system and still maintain the pressure drop as constant.

Figs. 14 and 15 depict the effect of melting range on the entropy generation rates caused by individual and combined sources for NEPCM slurry flow in microtube heat sink with tangential impingement. It is clear that the effect of melting range on heat transfer contribution of entropy generation rate is significant, while its effect on frictional entropy is not significant. An increase in melting range, increases heat transfer and total entropy generation rates consequently yielding a higher heat transfer enhancement. Therefore, a higher melting range yields more favorable results in the system’s cooling efficiency than larger ones.

Reynolds number has significant effect on entropy generation rates. From the above figures, it can be seen that the frictional entropy generation rate is not very sensitive to mass concentration of NEPCM particles at lower values of Reynolds numbers. Moreover, with increasing inlet Reynolds number the frictional irreversibilities of entropy generation rate increases, while the heat transfer contribution and total entropy generation rate decreases. In other words, higher inlet Reynolds number causes higher cooling efficiency for the heat sink.

7. Conclusions

The heat transfer enhancement in the microtube heat sink with tangential impingement by using NEPCM slurry coolant was studied numerically. Three-dimensional governing equations of laminar, incompressible fluid flow have been solved using the finite volume method. The heat conduction in the solid part of heat sink is taken into account. The influence of mass concentration and melting range of NEPCM, as well as Reynolds number on the thermal-hydraulic performance of a microchannel heat sink is examined. The results indicated that combination of NEPCM with PAO base fluid do measurably enhance the thermal performance of the device. Although with increasing mass concentration, the thermal performance increases but the extra pressure drop will somewhat decrease the beneficial effect. Furthermore, it was observed that decreasing temperature uniformity ($\Delta T_u$) is another advantage of using NEPCM slurry as coolant in TMHS. Moreover, the effect of Reynolds number, mass concentration and nanoparticle size on frictional, heat transfer and overall entropy generation found that using NEPCM slurry as coolant can help achieve lower entropy generation due to their high thermal properties. With increasing mass concentration, Reynolds number and melting range, the overall entropy generation rate decreases.

References


Fig. 15. Total entropy generation rate as a function of Reynolds number for different melting range.


