Analysis of performances of a manifold microchannel heat sink with nanofluids

Yun Yue, Shahabeddin K. Mohammadian, Yuwen Zhang*

Department of Mechanical and Aerospace Engineering, University of Missouri, Columbia, MO 65211, USA

A R T I C L E   I N F O

Article history:
Received 29 March 2014
Received in revised form
10 November 2014
Accepted 14 November 2014
Available online

Keywords:
Manifold microchannel heat sink
Nanofluids
Entropy generation

A B S T R A C T

Hydraulic and thermal performances of a manifold microchannel heat sink (MMHS) with and without nanofluids as working fluids have been investigated by a finite volume method. Effects of volume fraction, particle diameter of nanoparticles, and Reynolds number on the Nusselt number, pumping power, performance index, and entropy generation in a 3D unit cell were evaluated. The results showed that with increasing volume fraction of nanoparticles, Nusselt number and pumping power increase, but total entropy generation decreases. Increasing particle diameter leads to decreasing Nusselt number, pumping power, and performance index, but increasing the total entropy generation. Finally increasing the Reynolds number leads to increasing the Nusselt number and pumping power, but decreasing performance index and total entropy generation.

© 2014 Elsevier Masson SAS. All rights reserved.

1. Introduction

With increasing number of parts packed on a smaller and smaller chip area, the heat accumulated inside causes the temperature to rise dramatically. The integration of the transistors blocks the heat from dissipation by conventional air-cooling solutions [1]. As a direct result, the computer chips cannot function very well due to high temperatures, even cause failure. Tuckerman and Pease [2] first proposed and investigated a heat sink with micro-channels, along with liquid-cooling approach by using water as the coolant. By consuming 790 W/cm², the maximum temperature increased up to 71°C over the inlet water temperature and a thermal resistance of 0.1 cm²·°C/W was measured with a pressure drop of 2 bar. Harpole and Eninger [3] introduced the concept of manifold and showed the manifold microchannel heat sink could achieve 100 W/cm² with a pressure drop of 2 bar. Copland et al. [4] numerically analyzed the manifold microchannel heat sinks and found that the regions of high heat transfer were near the inlet. They also showed that the secondary maximum heat transfer occurred at the base of the microchannel below the inlet and at the top of the microchannel near the exit.

Brunschwiler et al. [5] studied the improvement in heat transfer efficiency by introducing the manifold system and single-phase liquid jet impingement at the top of the chip. They demonstrated a thermal resistance of 0.17 cm²·°C/W for a flow rate of 2.5 l/min, at the pressure drop of 0.25 bar. Escher et al. [6] experimentally investigated the hydrodynamic and thermal performances of an ultra-thin manifold microchannel heat sink and reported cooling power densities of over 700 W/cm² for a maximum temperature difference of 65 K between the chip and the inlet flow. Sharma et al. [7] studied a novel framework experimentally and computationally to determine the optimal operation conditions of water cooled micro-processor chips: they showed that the optimal operating conditions for chip cooling can be achieved via single- or multi-objective approaches.

Another way to improve the performance of microchannel heat sinks is to use nanofluids as coolants. Many studies indicate that adding low volume fraction of nanoparticles to the base fluid leads to significant increase on thermal conductivity of the coolant [8–11]. Li and Peterson [12] conducted an experimental investigation to examine the effects of temperature and volume fraction on the steady-state effective thermal conductivity of two types of nanofluids. It was reported that 6% CuO–water suspension has higher thermal conductivity up to 1.52 times than the pure distilled water, while 10% Al₂O₃–water suspension has higher thermal conductivity by a factor of 1.3. Some researchers investigated the convection of nanofluids and found that the use of nanofluids increases the heat transfer coefficient and Nusselt number [13–18]. Heris et al. [19] experimentally, investigated the thermal performance of laminar flow by using CuO–water and Al₂O₃–water...
nanofluids as working fluids. The Nusselt number and Peclet number of both nanofluids increased with increasing nanoparticle concentrations. Mohammadian et al. [20] investigated the laminar forced convection and entropy generation in a counter flow microchannel heat exchanger with and without nanofluids in hot and cold channels and reported that the capability of heat transfer of Al₂O₃—water nanofluids is higher when nanoparticles are used in cold channels.

In this paper, performance of the manifold microchannel heat sink with nanofluids as coolant was compared to the one with water by conducting computational simulation with various inlet velocities, nanoparticle diameters and solution concentrations. The performance index was investigated to evaluate the balance between the hydrodynamic and thermal performance to obtain an overall heat transfer performance. In addition, entropy efficiency of the manifold microchannel heat sink was also investigated. Several objective functions were represented to define optimal operation conditions for nanofluids cooled chip.

2. Physical and mathematical models

The manifold microchannel heat sink in this study is made of silicon. A schematic of the manifold microchannel heat sink is shown in Fig. 1(a). The dash line region illustrates the computational domain of a unit cell of the manifold microchannel heat sink. The fluid impinges into the inlet manifold and divides equally into two streams. Each stream takes away heat from the electronic chip when passing through the microchannels respectively. Fig. 1(b) shows the computational domain of a unit cell of the manifold microchannel heat sink in detail. In order to save computational resources, only half on the inlet to the outlet is modeled. Al₂O₃—water nanofluid is selected to be the working coolant. The properties of nanofluids, such as density, viscosity and thermal conductivity of the coolant, are functions of temperature, volume fraction, and diameter of the nanoparticles.

A constant heat flux of \( Q_w = 55 \text{ W/cm}^2 \) is applied uniformly to the bottom wall. The top front wall and bottom front wall are adiabatic. At the inlet, a uniform velocity is applied to simplify the problem. The inlet temperature is fixed at 300 K during the entire cooling procedure. The gauge pressure and the gradients of velocity and pressure at the outlet are assumed to be zero. All other faces are considered as walls with non-slip boundary conditions. Symmetric boundary condition is applied on the left-side of the inlet, the front part of the channel and the backsides of both of the inlet and outlet [21]. Other geometric dimensions of unit cell of manifold microchannel heat sink are summarized in Table 1.

It was assumed that the Reynolds number of the flow is in the range of 100—400, which means that the flow is laminar during the process of water flowing from the inlet to the outlet [7]. Also, it was assume that the low concentration of nanoparticles in the base fluid makes it behave like a single phase fluid and there is no agglomeration or sedimentation.

The continuity, momentum, and energy equations for a steady state flow can be written as:

![Physical model](image-url)
Continuity:
\[ \nabla \cdot \mathbf{V} = 0 \]  (1)

Momentum equations:
\[ \rho V \cdot \nabla V = -\nabla p + \nabla (\mu \nabla V) \]  (2)

Energy:
\[ (\rho c_p) V \cdot \nabla T = \nabla (k \nabla T) + \varphi \]  (3)

where \( \varphi \) is dissipation function and for an incompressible flow it is written as:
\[ \varphi = \left\{ \frac{2}{(\mu_{\text{ox}})^2} + \left( \frac{\mu_{\text{oy}}}{\mu_{\text{ox}}} \right)^2 + \left( \frac{\mu_{\text{oz}}}{\mu_{\text{ox}}} \right)^2 \right\} \]  (4)

For conjugated heat transfer in MMHS, the above equations should be solved simultaneously with the energy equation for solid material of MMHS which is.
\[ \nabla^2 T_s = 0 \]  (5)

3. Nanofluid properties

The density of nanofluid is determined on a basis of physical principle of mixture rule [22]. The expression is as the following format:
\[ \rho_{\text{eff}} = (1 - \alpha) \rho_f + \alpha \rho_p \]  (6)

where the subscript \( p \) and \( f \) are nanoparticle and the base fluid, which is pure water in this study and \( \alpha = V_p/V_f + V_p \) is the volume fraction of nanoparticles.

The heat capacity of nanofluid is obtained by assuming thermal equilibrium between the nanoparticles and the basic fluid [23].
\[ (\rho c_p)_{\text{eff}} = (1 - \alpha) (\rho c_p)_f + \alpha (\rho c_p)_p \]  (7)

The thermal conductivity for nanofluid, obtained based on the Brownian-motion effect has been found to agree with the experimental data sets very well [24,25]. This model takes particle size, volume fraction, types of particles and base fluid into consideration; it is valid for a temperature range from 300 K to 325 K and volume fraction up to 4%. The effective thermal conductivity of nanofluid can be expressed as follows:
\[ k_{\text{eff}} = k_{\text{static}} + k_{\text{Brownian}} \]  (8)

where the static part is Hamilton–Cresser model [26] and the dynamic part was based on kinetic theory.
\[ k_{\text{static}} = k_f \left[ \frac{(k_p + 2k_f) - 2\alpha(k_f - k_p)}{(k_p + 2k_f) + \alpha(k_f - k_p)} \right] \]  (9)

where \( k_l \) and \( k_p \) are the thermal conductivity of the base fluid and nanoparticles, respectively.
\[ k_{\text{Brownian}} = 5 \times 10^4 \alpha \rho_f c_p T \sqrt{\frac{M_b}{\rho_p}} g(T, \alpha) \]  (10)

where \( \rho_f \) and \( c_{p,f} \) are density and specific heat capacity of the fluid. \( \alpha \rho_p \) and \( \alpha \) are the particle density and volume fraction of nanofluids. \( \kappa_B \) is the Boltzmann constant, which has a value of 1.380710^{-23} \text{ J/K}.

The function \( g \) was determined semi-empirically and for the \( \text{Al}_2\text{O}_3 \)--water nanofluids, it has the following form:
\[ g = (a + b \ln(d_p) + c \ln(\alpha) + d \ln(\alpha) \ln(d_p) + e \ln(d_p)^2)\ln(T) \]
\[ + (m + h \ln(d_p) + i \ln(\alpha) + j \ln(\alpha) \ln(d_p) + k \ln(d_p)^2) \]  (11)

With the values of coefficients showed in Table 2, the \( \text{Al}_2\text{O}_3 \)--water nanofluid has a \( R^2 \) (regression) of 96% [27].

For several years, increasing concerns are paid on the importance of the interfacial thermal resistance (Kapitza resistance) [28,29] between the adjacent layers of the two different materials; it had an obvious effect on weakening the effective thermal conductivity of the nanofluids. In the static part, the relation between the effective nanoparticle thermal conductivity and the nanoparticle thermal conductivity has the following format:
\[ R_0 + \frac{d_p}{k_p} = \frac{d_p}{k_{p,\text{eff}}} \]  (12)

where \( d_p \) is the nanoparticle diameter and \( R_0 = 4 \times 10^{-8} \text{ km}^2/\text{W} \) [13].

The dynamic viscosity of nanofluids has a familiar format with the effective thermal conductivity [24]:
\[ \mu_{\text{eff}} = \mu_{\text{static}} + \mu_{\text{Brownian}} \]  (13)

where the static part [30] and dynamic part can be calculated respectively as follows:
\[ \mu_{\text{static}} = \frac{\mu_f}{(1 - \alpha)^2} \]  (14)
\[ \mu_{\text{Brownian}} = 5 \times 10^4 \alpha \rho_f c_p T \sqrt{\frac{M_b}{\rho_p}} g(T, \alpha) \]  (15)

The thermo-physical properties of pure water and silicon are as follow [31,32]:
\[ k_f = 0.6 \left(1 + 4.167 \times 10^{-5} T \right) \]  (16)
\[ \mu_f = 2.761 \times 10^{-6} \exp \left(\frac{1713}{T} \right) \]  (17)
\[ k_{\text{silicon}} = 290 - 0.4T \]  (18)

Table 2

<table>
<thead>
<tr>
<th>Coefficient</th>
<th>Values</th>
<th>Coefficient</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>52.813488759</td>
<td>m</td>
<td>-298.1919084</td>
</tr>
<tr>
<td>b</td>
<td>6.115037295</td>
<td>h</td>
<td>-34.53271906</td>
</tr>
<tr>
<td>c</td>
<td>0.695574584</td>
<td>i</td>
<td>-3.922529083</td>
</tr>
<tr>
<td>d</td>
<td>4.1745552786</td>
<td>j</td>
<td>-0.2354329626</td>
</tr>
<tr>
<td>e</td>
<td>0.176919300241</td>
<td>k</td>
<td>-0.999063481</td>
</tr>
</tbody>
</table>
4. Numerical solution

In this study, a commercial package Fluent is used to solve the conjugate heat transfer problem; and the UDF code has been written to evaluate the nanofluid properties. The equation convergence is checked by monitoring the scaled residuals. The normalized residual target is settled as 10⁻⁶ to get proper results. Compared with conventional-sized channel, the transition from laminar to turbulent flow is small in microchannels, which is 1000 [33]. In this paper, the range of Reynolds number is considered to be 100–400 to ensure that all the simulation are in the laminar regime. The Reynolds number is:

\[ Re = \frac{\rho_{\text{eff}} u_{\text{in}} D_h}{\mu_{\text{eff}}} \]  

(19)

where \( \rho_{\text{eff}} \) and \( \mu_{\text{eff}} \) are density and dynamic viscosity of nanofluids, respectively. \( D_h \) is the hydraulic diameter that is defined as:

\[ D_h = \frac{2HW}{H + W} \]  

(20)

Convection heat transfer coefficient is defined as below:

\[ h = \frac{q}{T_w - T_f} \]  

(21)

where \( q \) and \( T_w \) are the total heat flux and average temperature of all the fluid contact walls, respectively. \( T_f \) is volume-average temperature of the fluid zone. Heat transfer rate \( q \) is [18]:

\[ q = \frac{(\rho_c \rho)_\text{eff} u_{\text{in}} A_{\text{in}} (T_{\text{out}} - T_{\text{in}})}{\Lambda_{\text{h}}} \]  

(22)

Finally, the Nusselt number is defined as:

\[ Nu = \frac{hD_h}{\kappa_t} \]  

(23)

To guarantee high cooling efficiency, minimizing the combination of small channel width and high flow rate is required. However, these two parameters are all contribute to high pumping power, which is proportional to pressure drop. For comparing the cooling efficiency, the energy cost (pressure drop) should be taken into consideration. So the pumping power is defined as:

\[ P = V (p_{\text{in}} - p_{\text{out}}) \]  

(24)

where \( p_{\text{in}} \) and \( p_{\text{out}} \) are the pressure at the inlet and outlet of the nozzle respectively; and \( V \) is the volumetric flow rate (m³/s):

\[ V = u_{\text{in}} A_{\text{in}} \]  

(25)

where \( A_{\text{in}} \) and \( u_{\text{in}} \) are the area of the inlet channel cross section and the inlet velocity, respectively.

The performance index is employed to evaluate the balance between the hydrodynamic and thermal performance to obtain an overall performance:

\[ \eta = \frac{Nu}{P} \]  

(26)

where \( P \) is the pumping power. The total entropy generation is also investigated to analyze the effect of the second law of thermodynamics of the cooling system. The rate of volumetric entropy generation (\( \dot{S}_{\text{gen}} \)) in three-dimensional Cartesian coordinates is [27,34]:

\[
\dot{S}_{\text{gen}} = \frac{k}{T^2} \left[ \left( \frac{\partial T}{\partial x} \right)^2 + \left( \frac{\partial T}{\partial y} \right)^2 + \left( \frac{\partial T}{\partial z} \right)^2 \right] + \nu \left[ \left( \frac{\partial u}{\partial x} \right)^2 + \left( \frac{\partial u}{\partial y} \right)^2 + \left( \frac{\partial u}{\partial z} \right)^2 \right] + \left( \frac{\partial v}{\partial x} + \frac{\partial w}{\partial y} \right)^2 \right] \}
\]

(27)

where \( u, v, \) and \( w \) are velocity components in the \( x-, y-, \) and \( z\)-directions, respectively.

5. Grid independency and model validation

To ensure the accuracy of the simulation, four grids consisting of about 0.5 million (coarse), 0.8 million (intermediate), 1.3 million (fine), and 1.6 million (very fine) cells are used to study the grid sensitivity. The grid independency was conducted under the highest Reynolds number (\( Re = 400 \)) with a volume fraction of \( \alpha = 0.04 \), and the nanoparticles diameter of \( d_p = 47 \) nm. The results are evaluated by \( T_{\text{out}} \), and are summarized in Table 3. It is obvious that the maximum difference between the fine and the very fine grids results is less than 0.01%. Thus, the third grid (Fig. 2) with a number of about 1.3 million cells is the most suitable one for conducting simulation.

Experimental data with water conducted by Sharma et al. [7] are used for validating the simulations. The experimental data were obtained under a condition that volume flow rate is between 0.3 to 0.9 l/min at interval of 0.2 l/min and at 2 different water inlet temperatures of 30 °C and 40 °C. A constant heat dissipation of 100 W is applied from the chip. Fig. 3 shows the deviations of outlet temperature (\( T_{\text{out}} \)) of heat sink between experimental and numerical analysis.

6. Results and discussions

Thermal and hydraulic effects of Al2O3–water nanofluids on the performance of manifold microchannel heat sink have been studied in this paper. A 3D unit cell of heat transfer structure is developed to evaluate the heat sink performance and validated against experimental measurements. The spatial temperature distribution of both fluid and solid regions and velocity and pressure distribution of fluid region inside a microchannel of the computational domain are discussed. Then, the dependence of heat transfer performance on different volume fractions, nanoparticles diameters, and inlet velocities are discussed. After that, the sensitivity of the pumping power and performance index to the change in the design variables are studied. Finally, the influence of different parameters on total entropy generation is investigated.

Fig. 4 illustrates the temperature, velocity, and pressure contours at different Reynolds numbers. The heat transfer coefficient increases with increasing flow rate along the axial length of the channel. For fully developed laminar flow, the heat transfer coefficient is independent of the Reynolds number. However for this

<table>
<thead>
<tr>
<th>Mesh number</th>
<th>Number of cells</th>
<th>Temperature of outlet (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mesh 1</td>
<td>528,000</td>
<td>303.2</td>
</tr>
<tr>
<td>Mesh 2</td>
<td>825,000</td>
<td>303.1</td>
</tr>
<tr>
<td>Mesh 3</td>
<td>1,320,000</td>
<td>303.0</td>
</tr>
<tr>
<td>Mesh 4</td>
<td>1,606,400</td>
<td>303.0</td>
</tr>
</tbody>
</table>
case, for flow inside the microchannel of MMHS, the flow is developing over most of the axial length of the channel, because the flow impinges directly to the bottom surface of the microchannel when it enters, and blocks the development of boundary layer. After impinging on the microchannel bottom surface, the flow made a turn of 90° and traveled along the microchannels. So increasing Reynolds number leads to increase in the heat transfer coefficient that cause to temperature decline. The most effective cooling occurs in the inlet region just past the stagnant zone. For every Reynolds number, local Nusselt number first increases from the symmetry plane in inlet slot nozzle till the inlet nozzle region ends; this is due to the thinning of the thermal boundary layer caused by flow impingement. After the inlet nozzle region, the thermal boundary layer grows as the fluid is heated up along the channel, and causes to reduce in heat transfer coefficient, and consequently local Nusselt number [7]. Fig. 4(b) depicted the velocity distribution inside a microchannel. It is clear that with increasing Reynolds number, the vortices in both side of inlet and outlet regions become lower that cause to increase the heat transfer coefficient from the heat sink surfaces. As illustrated in Fig. 4(c), with increasing the Reynolds number, the pressure drop increases.

Fig. 5 depicts the temperature distributions in MMHS for Al2O3-water nanofluids for different volume fractions, and nanoparticles diameters at Re = 100. It can be seen that with increasing the volume fraction of nanoparticles, the bulk temperatures of the fluid and solid region decrease that cause decreasing the temperature difference between the surface and the fluid; this leads to increase in heat transfer coefficient. Also with increasing the diameter of nanoparticles, the temperatures of both fluid and solid regions increase that cause temperature difference between the surface and the fluid increases which consequently decreases the heat transfer coefficient. Another issue which should be noted is that, in higher volume fractions of nanoparticles or lower particle diameters, the average temperature of the solid phase is lower that increases the lifespan of the microchannels.

This study demonstrates that with increasing the nanoparticles volume fraction, viscosity, density, thermal conductivity, and pressure drop increase but heat capacity declines. The reason of increasing viscosity with increasing volume fraction is due to stronger hydrodynamic interaction between particles at higher volume fractions [18]. As illustrated in Fig. 6(a), due to higher temperature near the surfaces, local molecular viscosity at those areas is lower than that in the central region of the fluid flow. Furthermore, it can be seen that, molecular viscosity decreases through the microchannel from inlet to outlet. It is because, the fluid becomes warmer and its temperature goes up which causes kinetic energy enhancement.

Fig. 6(b) shows that with increasing the volume fraction of nanoparticles, local and average values of thermal conductivity increase. It is clear that local thermal conductivity is higher near the surfaces due to higher temperature at those areas. Also, it can be seen that, thermal conductivity increases through the microchannel from inlet to outlet because thermal conductivity of
Nano-liquid increases with increasing temperature. There are some mechanisms for the enhancement of thermal conductivity of nanofluids including the liquid layering on the surface of nanoparticles, Brownian motion, thermophoresis, osmophoresis, clustering of nanoparticles and ballistic transport of phonon in the nanoparticles [35–36].

The reason of increasing the pressure drop by increasing the nanoparticles concentration is that fluid is more viscous at higher volume fractions. Higher concentration of the nanofluids leads to higher shear stress on walls, so the pressure drop increases [37].

Furthermore, this study shows that with increasing diameter of the nanoparticles, viscosity, thermal conductivity, and pressure drop decrease. As illustrated in Fig. 7(a), with increasing nanoparticles diameter, molecular viscosity decreases that cause decreasing pressure drop. It is due to the fact that, at a constant volume fraction, the number of nanoparticles is higher for smaller nanoparticles that cause higher shear stress on the fluid. Fig. 7(b) shows that with increasing the nanoparticles diameter, thermal conductivity decreases. It is because with increasing the particles diameter, the effect of Brownian motion decreases. Also as the thermal energy transfer depends on the surface area, smaller particles of the same volumetric concentration provide more surface area for transferring the thermal energy [37].

Fig. 8 shows the relations of the heat transfer performance obtained for different volume fractions as well as different nanoparticles diameters, and Reynolds numbers. It is obvious that with increasing the low volume fraction of nanoparticles the Nusselt number increases, especially in the cases that particle diameter is 29 nm. This is due to the fact that, heat transfer coefficient is higher than that of the base fluid (α = 0). Also higher mass concentration due to higher volume fraction, leads to higher momentum, and this higher momentum transfers thermal energy more efficiently [18].

Furthermore, it can also be seen that the Nusselt number is higher for the case of small nanoparticle diameter. It is because small particle diameter makes more particles in a fixed volume fraction, which increases the number of particles and leads to more surface areas for heat transfer. Other reasons of increasing Nusselt number with decreasing the particles diameter are aggregation of nanoparticles, Brownian motion, and formation of liquid layers around nanoparticles that effect on thermal conductivity of nanoparticles.
nanofluids [18]. Additionally, there is also a trend that with increasing Reynolds number, the Nusselt number increases.

Compared with water, heat transfer performance improves greatly in low Reynolds number regions. In the best condition ($d_p = 29 \text{ nm}$ and $\alpha = 0.04$), the Nusselt number increases by 15.46% at $Re = 400$, by 32.57% at $Re = 200$, and by 38.81% at $Re = 100$. Therefore, to achieve better thermal performance, the use of nanofluids with higher volume fraction and smaller particles diameter at lower Reynolds number should be more favorable for cooling systems.

Fig. 9 illustrates the pumping power variations in different volume fractions of nanoparticles, particles diameters, and Reynolds numbers. It can be seen that with increasing Reynolds number and volume fraction of nanoparticles, the pumping power required to impinge nanofluids into microchannels increases. Compared with water, high volume fraction causes the fluid to become more viscous which results in more pressure drop in the system. It is also apparent from the figure that, pumping power decreases with increasing particle diameter which is because the viscosity decreases. Compared with water, introducing nanofluids as coolant will cost more energy regardless of how big the nanoparticles are, how fast they flow and how much nanoparticles added to the base fluid.

Fig. 10 shows the changes in performance index with volume fraction of nanoparticles, particles diameter, and Reynolds number. In general, addition of low volume fraction of nanoparticles to the base fluids has two opposing effects. The favorable effect is that with increasing the volume fraction of nanoparticles, thermal conductivity increases. But on the other hand, it cause to increase in viscosity and consequently pressure drop. So it can be seen in Fig. 10 that in some cases, with increasing the volume fraction of nanoparticles performance index increases, which shows in these cases the effect of thermal conductivity enhancement prevail over the effect of viscosity increase. Also, this figure reflects the fact that
performance index decreases monotonically with increasing the diameter of nanoparticles and Reynolds number at all values of volume fractions. Compared with water, introducing nanofluids as coolant will obtain higher performance at lower Reynolds numbers, higher volume fractions, and smaller particles diameters.

Fig. 11 reveals the relations of the total entropy generation rate with different volume fraction of nanoparticles, particles diameters, and Reynolds numbers. It can be seen that the total entropy generation of the manifold microchannel heat sink decreases with increasing Reynolds number and volume fraction. Total entropy generation is compounded of thermal and frictional entropy generations. In this study, the order of magnitude of frictional entropy generation is much lower than the thermal entropy generation because of the geometric design; it can be negligible. It is clear that thermal entropy generation is dependent on temperature variations [see Eq. (27)]. As illustrated in Figs. 3 and 4, the bulk temperature of the fluid flow decreases with increasing volume fraction of nanoparticles or Reynolds number; this decreases the temperature gradients inside the flow, and consequently decreases the thermal and total entropy generations. Furthermore, with increasing nanoparticles diameter, total entropy generation increases. According to Fig. 5, with increasing the nanoparticles diameter the bulk temperature of the fluid flow increases that causes to increase in temperature gradients which increases the thermal and total entropy generations. When \(d_p = 29\) and \(\alpha = 0.04\), nanofluids can improve the results by 11.30% at \(Re = 100\), by 9.96% at \(Re = 200\) and by 7.63% at \(Re = 400\). To enable the heat sink function well, cooling procedure should be conducted at smaller particle diameter, lower Reynolds number and higher volume fraction.

7. Conclusions

In this paper, the thermal and hydrodynamic performances of a manifold microchannel heat sink with \(Al_2O_3\)-water nanofluids as working fluid were systematically studied. The following conclusions can be drawn:

1) With the increase of volume fraction, Nusselt number and pumping power increase, total entropy generation decreases, and the performance index may increase or decrease depending on the condition.

2) An increase in particle diameter leads to decrease in Nusselt number, pumping power, and performance index; but total entropy generation increases.

3) An increase in Reynolds number increases Nusselt number and pumping power, but decreases performance index and total entropy generation.
**References**


