**Numerical Analysis on the Thermal Performance of Microchannel Heat Sinks with Al2O3 Nanofluid and Various Fins**

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ABSTRACT

The hydraulic and thermal performance of microchannel heat sink configurations for high performance electronic cooling applications is investigated by numerical modelling. Conjugate heat transfer simulations are obtained through the silicon walls and the fluid domain of a square base prism heat sink traversed by 50 parallel rectangular cooling ducts, under a 150 W/cm2 constant heat flux input through the base. Al2O3 nanofluid coolant with a nanoparticle volume fraction ranging from 0 to 3% is supplied at 298 K, over the Reynolds number range 100 to 350, modelled as a single-phase homogeneous medium. Rectangular, twisted, and zig-zag fins are inserted into the plain rectangular duct to enhance the heat transfer rate. The zig-zag fin and 3% Al2O3 nanofluid provide the best thermal performance, with a 6.44 K lower average heated wall contact temperature, 60% higher Nusselt number, and 15% higher second law efficiency than without fins and plain water cooling. Twist in the microchannel fin unexpectedly reduced the microchannel pressure drop by 2% to 15% compared to a straight fin, possibly due to the more evenly distributed axial mass flux across the microchannel.

**Keywords**: microchannel heat sink, nanofluid, fins, pressure drop, conjugate heat transfer model, computational fluid dynamics.

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| **Nomenclature**  *Symbols*   |  |  |  | | --- | --- | --- | |  | Exergy expenditure rate | W | |  | Exergy destruction rate | W | |  | Convection heat transfer coefficient | W/(m2.K) | | |  | Nusselt number |  | | | *Nux* | Local Nusselt number |  | | |  | Heat flux | W/m2 | | |  | Total thermal resistance | W/K | | |  | Pressure | Pa | | |  | Pumping power | W | | |  | Thermal conductivity | W/(m.K) | | |  | Friction factor |  | | |  | Temperature | K | | |  | Reynolds number |  | | |  | Constant pressure specific heat | J/kg | | |  | Hydraulic diameter | m | | |  | Fluid velocity | m/s | | |  | Volume flow rate | m3/s | | |  | Channel cross-sectional area | m2 | | |  | Channel length | m | |   *Greek symbols*   |  |  |  |  | | --- | --- | --- | --- | |  | Pressure drop | Pa | | |  | Second law thermodynamic efficiency |  | |  | Density | kg/m3 | | |  | Viscosity | kg/(m.s) | | |  | Volume fraction of the nanoparticles |  | | |  | Exergy flow | J/kg |   *Subscripts*   |  |  | | --- | --- | |  | solid | |  | fluid | |  | base fluid | |  | nanofluid | |  | nanoparticles | |  | inlet | |  | outlet | |

# Introduction

The continuous growth in power and miniaturization of modern electronics require more effective thermal management systems. Traditional coolants and cooling methods struggle to ensure low temperatures and hence efficiency and reliability of new high-performance electronic components. The heat sink performance of straight rectangular microchannels is well-documented [1], optimized, and has limited residual growth potential. Consequently, it is necessary to investigate combinations of new cooling flow passages and of unconventional working fluids to enhance the rate of heat transfer in microchannels, adapting heat exchange features that are found effective in conventional channels [2].

Fin arrays in microchannels expand the heat transfer surface area [3]. Ribs and cavities enhance flow mixing and thereby the heat transport away from a hot source [4]. They disrupt the growth of wall boundary layers, creating high local heat transfer regions at the rib and cavity leading edges [1]. Channel bifurcations [5] likewise create new boundary layer leading edges with high local heat transfer characteristics. This may explain the better thermal performance in fractal microchannels at increasing channel height to width aspect ratio [6]. Channel curvature can create Dean vortices to enhance heat transport by flow mixing [7].

Sinusoidal waviness in microchannels increases the flow path length, which improves the heat transfer performance by a factor of 1.7 to 2.9 at a pressure loss penalty of 1.3 to 2.0 due to the lengthened flow run [8]. Transverse waviness causes less pressure loss penalty than longitudinal waviness, while a zig-zag path provides comparatively more heat transport than a sinusoidal path but at a higher pressure loss [9]. A convergent-divergent, or varicose, channel can increase flow mixing and thereby the heat transport [10].

A vertical stacking of microchannels creates a double-layered microchannel heat sink, with either parallel flow or counter flow coolants [11]. The counter flow operation, like in classical heat exchangers, provides greater thermal performance at the expense of greater complexity in manufacturing the channels and in assembling their end connections.

All these approaches can improve the cooling performance, albeit with a pressure-drop penalty, but they need to be combined with other technologies to generate the performance growth capacity required to meet the higher heat flux dissipation demands of advanced electronic components.

Among other technologies, novel coolants with improved thermo-physical properties, such as nanofluids, have gained particular attention. Nanofluids are known to exhibit high stability, flexible properties, minimal particle agglomeration, in addition to an enlarged effective particle surface for optimum interphase heat exchange compared to micrometre-sized particle suspensions [12]. Limited additional pressure drop and mechanical deterioration make nanofluids particularly useful in compact heat exchanger applications [13]. Several hypotheses have been formulated to explain the thermal properties of nanofluids based on micro-scale mechanisms. These mechanisms include clustering, particle rotation, fluid layers particle encapsulation, particle migration, Brownian motion connected with micro convection and corresponding non-uniform property profiles, and the perturbation of the boundary layers [14].

Various studies have investigated the use of nanofluids in microchannels. Pak, et al. [15] investigated the convective heat transfer behaviours of TiO2 and Al2O3 nanofluids using turbulent flow conditions. Seyf, et al. [16] studied the thermal and hydraulic performance of Al2O­3 nanofluid in a counterflow microchannel heat exchanger. Yan, et al. [17] numerically showed an increase in the heat transfer performance of a micro-heat sink resulting from the increase in nanoparticle volume concentration and Reynolds number. Al2O3 nanofluid was also investigated by Irandoost, et al. [18], who predicted heat transfer coefficient improvements based on nanoparticle diameter and volume fraction of Al2O­3 under laminar flow. Shi, et al. [19] performed further research of heat transfer rates in microchannels with different volume fractions of nanoparticles, showing a non-linear relationship between the increase in thermal performance and the volume fraction. Rahimi-Gorji, et al. [20] assessed the heat transfer properties of a microchannel heat sink using several nanoparticles in water and ethylene glycol including Cu, Al2O3, Ag and TiO2 nanoparticles, showing that the temperature difference between the working fluid and microchannel heat sink walls decreased at higher concentrations due to the increased Brownian motion of the nanoparticles.

Ambreen, et al. [21] studied the combined effects of nanofluids and fins with square, circular and hexagonal outline profiles on a micropin-fin heat sink. Akbari, et al. [22] modelled the heat transfer of Cu nanofluid under turbulent flow in a rectangular microchannel with semi-attached fins, which produced stronger vortices and a better mixture. Wang, et al. [23] investigated the impact of geometric designs on the thermal and hydraulic characteristics of microchannel heat sinks, showing that the thermal resistance increased by increasing the number of channels, but at the penalty of a considerable additional pressure drop. Sheikhalipour, et al. [24] examined nanofluid heat transfer within trapezoidal microchannels and determined that adding nanoparticles has a marginal influence on the heat transfer if the Reynolds number is not sufficiently high. Toghraie, et al. [25] explored the thermal and hydraulic characteristics in smooth, sinusoidal, and zigzag shaped microchannels using nanofluid. Alipour, et al. [26] examined the effect of a semi-attached T-fin on the thermal properties of a silver-water nanofluid at various volume fractions under turbulent flow in a trapezoidal microchannel, confirming that the convection heat transfer coefficient in the boundary layer increases at increasing Reynolds numbers and volume concentrations of the nanoparticles. Sakanova, et al. [27] examined the combined effect of nanofluids and wavy channel structures, showing that the introduction of nanoparticles is dominant compared to the wavy wall effect.

These studies show that certain flow mixing devices, like fins, can enhance the heat removal rate of a microchannel heat sink at the penalty of additional pressure drop. It is therefore essential to quantify the trade-off between any thermal gain and any pressure drop penalty from these flow trips. To do so, Bahiraei, et al. [28] used a Figure of Merit, described as the ratio of the heat removal rate gain fraction to the pumping power increment fraction of a microchannel with fins with respect to a baseline channel without fins. A Figure of Merit higher than one indicates improvement with respect to the baseline. Alternatives to the Figure of Merit are reported in the literature, based on comparisons of entropy generation [29], exergy expenditure and exergy destruction [30], and of the second law efficiency. Improved designs were sought by minimising the entropy generation [31], which is made up by a dominant heat transfer contribution and a secondary contribution due to friction [32]. Increasing the Reynolds number reduces the former, which depends on the temperature gradient in the flow direction [33], but increases the latter [32]. Entropy generation is high at the microchannel entrance, due to locally high thermal and velocity gradients [34], making this region a prime target for microchannel re-designs. Past entropy minimization driven microchannel re-designs used fan-shaped re-entrant cavities and internal ribs [35] and changes to the channel aspect ratio [36] to alter the hydraulic and thermal boundary layers. Increasing the wall thermal conductivity [37] or the solid wall to coolant thermal conductivity ratio [36] lowers the entropy generation in the microchannel. This reinforces the importance of good material selection in microchannel heat sink design.

The trade-off analyses based on the Figure of Merit and its alternatives have led to the identification of heat sink configurations in which inserting fins can enhance the thermal performance at the expense of only a modest pressure drop. These studies have also shown that nanofluid-type coolants can improve the heat exchange process in forced-convection microchannel heat sinks.

To date, no study has been undertaken on combining the three fin designs shown in Figure 1 with nanofluids in a rectangular microchannel heat sink. In an attempt to address this gap in the current knowledge, a numerical study is presented herein that considered the thermal and hydraulic performance of a forced convection microchannel heat sink that has no fins (MC-N), or has a rectangular (MC-R), twisted (MC-T), or zigzag (MC-Z) fin, cooled by an Al2O3 nanofluid with a volume fraction range 0 to 3%. Past numerical investigations of microchannel flows over the Reynolds number ranges 100 to 500 [38] and 99 to 232 [27] prompted modelling the flow over the Reynolds number range 100 to 350, to keep the range consistent with past work. The intent is to combine the increased surface area and the vertical motion due to the fins with the increased thermal conductivity of the working fluid, leading to decreased thermal resistance of the microchannel and decreased contact temperature of the heat sink base.

# Methodology

A numerical method for estimating the steady-state thermo-hydraulic properties of the heat sink shown in Figure 1(a) is defined. The heat sink is essentially a prismatic block of silicon with flow-through perforations. A conjugate steady heat transfer problem is defined by solving the Laplace equation for temperature in the solid portion of the block and the steady, viscous, incompressible Navier-Stokes equations within the perforations.

**2.1 Governing equations and boundary conditions**

This research uses a water-based nanofluid as a working fluid, modelled as a single-phase homogeneous Newtonian fluid. The conservation of mass, momentum, and energy are solved numerically according to the assumptions below:

1. The fluid region is single-phase, laminar, incompressible, and three-dimensional in nature.
2. The working fluid thermal properties are temperature-dependent and steady.
3. Gravitational effects and radiative heat transfer are neglected.

The fluid flow governing equations are reduced to the incompressible scalar transport equations in orthogonal cartesian coordinates, [39]:

|  |  |  |
| --- | --- | --- |
|  |  | (1) |

Here , , and are the transported flow properties, diffusion term, and source term vectors, respectively. , is the molecular viscosity, is the Prandtl number, and is the velocity vector, is the absolute pressure, is the absolute temperature of the fluid, and is the vector transpose.

The Laplace equation

|  |  |  |
| --- | --- | --- |
|  |  | (2) |

governs the temperature in the microchannel walls [39].

Boundary conditions with the external environment are imposed for both solid and fluid regions as follows.

At the channel inlet, the inflow is specified by K; for the solid surface co-planar with the inlet, .

At the microchannel outlet, the outflow is specified by: ; for the solid surface co-planar with the outlet, .

At the bottom wall of the microchannel, , whereas at the top wall,, and at the left and right walls, (symmetry).

At the interfaces between fluid and solid, i.e., the inner walls of microchannel and fins, the boundary conditions are: , where is normal to the wall, is the thermal conductivity of the fluid and is the thermal conductivity of the solid.

**2.2 Nanofluid model**

The coolant is a suspension of Al2O3 nanoparticles in water. The volume fraction () of the nanoparticles is 1, 2 and 3%. Table 1 reports the thermophysical properties of nanoparticles and pure water.

Table 1 The thermo-physical properties of water and nanoparticles

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  |  |  |  |  |
| Water | 997 | 4179 | 0.613 | 0.000855 |
| Al2O3 [39] | 3880 | 733 | 36 | - |

The properties of the equivalent homogeneous fluid are estimated as follows [39]:

The density of the nanofluid is:

|  |  |  |
| --- | --- | --- |
|  |  | (3) |

The specific heat capacity of the nanofluid is defined as:

|  |  |  |
| --- | --- | --- |
|  |  | (4) |

The thermal conductivity of the nanofluid is:

|  |  |  |
| --- | --- | --- |
|  |  | (5) |

The effective dynamic viscosity of the nanofluid is:

|  |  |  |
| --- | --- | --- |
|  |  | (6) |

**2.3 Numerical solver**

The three-dimensional conjugate heat transfer problem was solved numerically using FLUENT 19.5. The SIMPLE solution method was used to evaluate the pressure-velocity coupling [39]. Second-order upwind discretization was used for the momentum and energy equations. The fluid region boundary conditions of section 2.1 were applied by using the velocity-inlet, pressure-outlet, and wall boundary types in FLUENT 19.5. At the velocity-inlet, was uniform and normal to the boundary. At the pressure-outlet, a uniform gauge pressure of zero was used. The Thermal Conditions of the velocity-inlet were set to a uniform total temperature = 298 K. The interior walls bounding the fluid region were modelled as no-slip wall type boundaries with Coupled type Thermal Conditions. The front, back, and top exterior walls of the microchannel used a zero heat flux Thermal Condition, to impose the zero wall-normal temperature gradients specified in section 2.1. A constant heat flux of 150 W/cm2 was prescribed at the bottom wall. To reduce the computational effort, the computational domain was constrained to one out of the 50 microchannels that make up the heat sink shown in Figure 1(a). The FLUENT 19.5 symmetric boundary type has used either side of this spanwise-repeating unit that is shown in Figure 1(d). The solution was deemed to have converged when the residuals for continuity, momentum and energy equation were 1×10-6, 1×10-6, and 1×10-7, respectively [39]. The computational cost of each simulation was 40 core hours on a 3.2GHz shared memory high-performance computer cluster.

**2.4 Geometry**

Figure 1(a) illustrates the schematic of a rectangular microchannel heat sink, uniformly heated from below and cooled by nanofluid through flow. 50 parallel rectangular ducts run through a square base silicon block, along . Six out of the 50 ducts are shown in Figure 1(a), for clarity. Due to the relatively large number of parallel ducts, only one duct is modelled and the temperature field is assumed symmetric between adjacent ducts, about the sectional planes shown by the dashed lines in Figure 1(a). Within the microchannel, different fin arrangements are considered, as shown in Figure 1(d-e). These are microchannels (MC) with either no fins (MC-N), or a rectangular (MC-R), or a twisted (MC-T), or a zigzag (MC-Z) fin. The twist and zigzag patterns, shown in Figure 1(b), are spatially periodic and repeat along channel length. The pattern wavelength in is and the pattern amplitudes in and are 0 and , respectively. All microchannels use a top and bottom wall uniform thickness and all side walls have constant thickness 2. Table 2 lists the relevant dimensions of the microchannel and of the fins. In order to decrease the thermal stress between the heat sink and chip, all solid parts of the microchannel are made from the same material used in most modern integrated circuit chips, i.e., silicon. The properties of silicon used in the conjugate heat transfer simulation are shown in Table 3.

Table 2 Geometric dimensions of the square microchannel heat sink, in mm.

|  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| *Lx1* | *Lx2* | *Lx3* | *H* | *h* | *W* | *w* | *T* |  | Fin thickness |
| 10 | 0.35 | 10 | 0.25 | 0.05 | 0.12 | 0.04 | 0.175 | 0.035 | 0.01 |

Table 3 The thermo-physical properties of silicon [40]

|  |  |  |  |
| --- | --- | --- | --- |
|  |  |  |  |
| Silicon | 2330 | 710 | 148 |

Diagram, engineering drawing

Description automatically generated

Figure 1 (a) Isometric view of the rectangular microchannel heat sink. (b) See-through top view of the twisted and of the zigzag fins. (c) Front view of the microchannel heat sink. (d) Microchannel heat sink without fin (MC-N). (e) Microchannel heat sink with a rectangular fin (MC-R). (f) Microchannel heat sink with a twisted fin (MC-T). (g) Microchannel heat sink with a zigzag fin (MC-Z).

* 1. **Data reduction**

The nanofluid Reynolds number is defined as:

|  |  |  |
| --- | --- | --- |
|  |  | (7) |

and ranges between 100 and 350. is the average fluid velocity at the microchannel heat sink inlet and and are defined in equations (3) and (6), respectively. is the hydraulic diameter, which is defined by:

|  |  |  |
| --- | --- | --- |
|  |  | (8) |

is the average convection heat transfer coefficient given by [39]:

|  |  |  |
| --- | --- | --- |
|  |  | (9) |

where *Tw*is the average bottom wall temperature of the microchannel heat sink and is the heat flux from the electronic component.

|  |  |  |
| --- | --- | --- |
|  |  | (10) |

where and are the inlet and the outlet fluid temperatures of the microchannel heat sink, respectively.

The average Nusselt number is then given by [39]:

|  |  |  |
| --- | --- | --- |
|  |  | (11) |

where is the thermal conductivity of the working fluid defined in equation (5).

The thermal resistance is [39]:

|  |  |  |
| --- | --- | --- |
|  |  | (12) |

The friction factor is [39]:

|  |  |  |
| --- | --- | --- |
|  |  | (13) |

where is the pressure drop of the working fluid between the inlet and the outlet:

|  |  |  |
| --- | --- | --- |
|  |  | (14) |

The pumping power of the microchannel heat sink is [39]:

|  |  |  |  |
| --- | --- | --- | --- |
| |  |  |  | | --- | --- | --- | |  |  | (15) | |

where is the volumetric flow rate.

The overall performance of each of the proposed designs is evaluated by a Figure of Merit () in order to determine the trade-off between rising pumping power requirements and the rate of heat transfer gains. The Figure of Merit, or Performance Evaluation Criterion (PEC) [10], compares the fractional gain in Nusselt number to the fractional gain in the friction factor [28]:

|  |  |  |
| --- | --- | --- |
|  |  | (16) |

**2.6 Exergetic performance assessment**

The removal of heat from the hot base of the heat sink by a colder coolant flow is referred to as a natural process in thermodynamics as it occurs unaided by work input expenditure, unlike in a heat pump. As for all real natural thermodynamic processes, higher grade energy is converted to lower grade heat, i.e., that has a lower potential for conversion into useful work by interaction with the surrounding environment, which is referred to as the dead state. This change in work potential can be evaluated by determining the exergy balance of the microchannel across the computational domain control volume. Referring to the schematic in Figure 2, the exergy balance across the dashed line computational domain control volume is:

|  |  |
| --- | --- |
|  | (17) |

Figure 2: Diagram of exergy control volume analysis of a microchannel heat sink. The bottom wall is heated, the top wall is thermally insulated.

where is the dead state temperature that is taken to be equal to the coolant inflow temperature K, is the control volume surface temperature, is the heat flux vector (which is defined as being positive away, or outwards, from the control volume), is the control volume boundary normal unit vector (again, positive when outwards), is the coolant density, is the coolant velocity, and is the rate of exergy destruction. The flow exergy can be defined as , where is the specific stagnation enthalpy, kJ kg-1 is the specific stagnation enthalpy at , is the specific entropy, J kg-1 K-1 is the specific entropy at , and the contribution to the flow exergy due to gravity is neglected as the inlet and the outlet of the microchannel are horizontal (i.e., have no vertical component) [39]. The closed-surface integration in is acted anticlockwise through the six surfaces of the microchannel in Figure 1(a). Rearranging equation (17) gives the exergy destruction rate as:

|  |  |
| --- | --- |
|  | (18) |

where, for simplicity, the first term on the right-hand side is evaluated only over the heated bottom surface of the microchannel, as on all other solid faces that are adiabatic, as stated in section 2.1.

The rate of exergy expended, , in cooling the microchannel has its main contribution from the rate of heat transfer through the heated base, which was also used by Liu, et al. [30], plus a further contribution from the coolant flow supply, the work rate potential of which is considered fully expended at the coolant outflow. This gives:

|  |  |
| --- | --- |
|  | (19) |

Equations (18) and (19) differ by the extent over which the surface integration is evaluated. The estimated rates of exergy expenditure and destruction are used to determine the second law efficiency of the microchannel, [39].

Whereas exergy and the second law efficiency are usually associated with the quality of energy utilization in heat engines, their contextualization as quality parameters for heat sinks requires that they are interpreted from the alternative perspective of indicators of the least effort heat removal rate from the heat sink. Given the reference dead state of the coolant temperature, it is desirable to transfer heat at a temperature that is as close to the coolant temperature as possible, since having the source of heat at a higher temperature is not only undesirable with regard to the longevity of electronic components but also presumes the upfront use of higher-grade energy to generate such higher temperature heat, which is wasteful. It is also desirable to use low grade work rate for circulating the coolant. Both these aspects are combined in the rate of exergy expenditure, , that thus becomes a useful quality indicator for comparing the performances of different microchannel heat sinks.

**2.7 Numerical mesh**

The numerical mesh is fully hexahedral in the solid and the fluid regions of the computational domain of Figure 1(a). Figure 3 shows the mesh structure of the four configurations, MC-N, MC-R, MC-T, and MC-Z. The mesh spacing is uniform along the channel length, in , and height, in . The fin is uniformly discretised across its thickness, in , as are the channel side walls. The mesh spacing in the narrowest gap between the fin and the side walls is uniform, as shown to the right of the fin in Figure 3(c) and in Figure 3(d). The fluid domain first interior cell dimension is upper bound limited to the mesh spacing across the fin thickness. Geometric mesh stretching in from the solid boundaries allows the discretization to accommodate the fin thickness, twist, and zigzag. The top and bottom channel walls use the same discretization of the fluid domain between them. A grid-independence study was conducted for cases MC-N, MC-R, MC-T, and MC-Z using water at a Reynolds number of 250. Figure 4 shows the microchannel average bottom wall temperature and the loss in static pressure , between the microchannel inlet and outlet, as a function of the number of unit cells in the numerical mesh. The and in Figure 4 show no significant difference between 0.8M and 5.2M cells. Therefore, the mesh with 2.3M cells was selected for all cases.

|  |  |  |  |
| --- | --- | --- | --- |
|  |  |  |  |
| (a) | (b) | (c) | (d) |

Figure 3 Cross-section mesh structure of (a) MC-N, (b) MC-R, (c) MC-T, and (d) MC-Z.



(a)



(b)

Figure 4 Numerical mesh sensitivity of (a) the predicted bottom microchannel wall temperature and (b) the predicted pressure drop for MC-N, MC-R, MC-T, and MC-Z with water at Re = 250.

**2.7 Validation**

To assess the accuracy of the numerical predictions, the local Nusselt number of MC-N is compared in Figure 5 with that obtained from an empirical correlation in Phillips [41]. Both methods predict the same exponential reduction in Nusselt number with increasing distance from the channel inlet . The numerical prediction of Nu is below the empirical correlation estimate along the full channel, by a constant . As the local Nusselt number ranges between 5 and 9, this difference amounts to a maximum error of about 5%. The thermal behaviour of a microchannel is expected to change by the presence of a fin. Xie, et al. [42] reported the thermal resistance of a rectangular microchannel with a rectangular fin running from about half-length to the channel exit. The geometry can be considered as a compound of the MC-N and the MC-R joined head to tail and enables testing the consistency of the current numerical method for microchannels with a fin against published work. The thermal resistance is closely related to the microchannel bottom wall to coolant temperature difference by equation (12). By matching the specific geometry in Xie, et al. [42], Figure 6 shows that the temperature differences predicted by the current numerical model and the ones estimated from Xie, et al. [42] are in close agreement over the Reynolds number range 250 to 350. This builds confidence in the current method of estimating the thermal parameters of microchannels with and without a fin.

Figure 7 compares the numerical predictions of the pressure drop across MC-N against that from an empirical correlation by Steinke, et al. [43], for pure water, and against numerical simulations at = 0 and 5% nanoparticle volume fraction in Sakanova, et al. [27]. The empirical correlation by Steinke, et al. [43] accounts for the channel aspect ratio () both in its friction loss term and in its channel entry major loss term, where the Hagenbach factor [43] is used. Over the Reynolds numbers range 100 to 350 the pressure drop shows good agreement among the numerical predictions and the empirical correlation, for = 0. At = 5%, both numerical predictions report a greater pressure loss with the nanofluid running through the rectangular microchannel than with pure water and both simulations at = 5% predict the same rising trend with increasing Reynolds number. Using the specific geometry of the partially finned microchannel by Xie, et al. [42], essentially the same pressure drop is predicted as that reported as in [44]. This indicates that the proposed model can correctly predict the thermal and hydraulic performance for a reliable comparative evaluation of the various designs considered.



Figure 5 Local Nusselt number along the rectangular microchannel heat sink (MC-N). Prediction from the current numerical model (circles) and by the empirical correlation by Phillips [41] (dashed line) reported as equation (19) in Wang, et al. [44].



Figure 6 Difference between the microchannel bottom wall average temperature and the cooling flow inflow temperature as a function of Reynolds number for MC-N, MC-R, MC-T, and MC-Z at = 3% Al2O3 nanoparticles. Current model predictions are compared with Xie, et al. [42] at = 0 for a partially finned microchannel.



Figure 7 Static pressure drop from inlet to outlet of rectangular microchannels with fins (MC-R, MC-T, MC-Z) and without fins (MC-N), at different Reynolds numbers and Al2O3 nanoparticle volume fractions. Current model predictions are compared with the empirical correlation reported as equation (12) in Steinke, et al. [43] from MC-N at , with numerical results from Sakanova, et al. [27] for MC-N at = 5%, and with Xie, et al. [42] for a partially finned microchannel similar to MC-R at .

# Results and Discussion

This section reports the thermal and the hydraulic performance of the microchannel heat sink with different fin designs and nanoparticle concentrations. It concludes with considerations of the second law efficiency.

**3.1 Effects of fin design on the microchannel heat sink temperature**

Figure 8 compares the temperature distribution on cross-sections in the and planes for the four different fin configurations, MC-N, MC-R, MC-T, and MC-Z. The Reynolds number and nanoparticle concentration are fixed at 350 and 3%, respectively, in this comparison, but similar conclusions can be drawn for all other regimes and working fluids investigated. The results show that the temperature for all designs increases monotonically from the inlet to the outlet, and that the highest temperature is predicted at the bottom wall near the outlet. Cases MC-Z, MC-T, and MC-R display lower solid wall temperatures towards the outflow compared to MC-N. This indicates a lower thermal resistance across the working fluid thermal boundary layer, resulting in a cooler bottom wall. The average bottom wall temperature in the MC-Z, MC-T and MC-R configurations is consequently lower than in MC-N, due to the presence of the fins in each case. The MC-Z configuration provides the most desirable solid wall temperature distribution among the four configurations as it keeps the microchannel heat sink walls temperature lower than the other designs.

Background pattern

Description automatically generated

Figure 8 Temperature colour iso-levels for MC-N, MC-R, MC-T, and MC-Z at = 3% Al2O3 nanoparticles at a Reynolds number of 350. and planes.

The analysis of the temperature range in the solid part of the microchannel heat sinks yields similar results. Figure 9 illustrates the temperature distributions on the cross-sectional plane at from the inlet, for the MC-N, MC-R, MC-T, and MC-Z configurations at Re = 350 and = 3% Al2O3 nanoparticles. The predicted temperature at the bottom wall of MC-Z is the lowest of the four configurations, followed by MC-T, MC-R, and MC-N in ascending order. The microchannel heat sink with the zigzag fin provides the lowest resistance to the transfer of heat from the solid to the working fluid among the four microchannel designs at the same heat input rate.

|  |  |  |  |
| --- | --- | --- | --- |
| MC-N | MC-R | MC-T | MC-Z |
|  |  |  |  |
| A screenshot of a computer  Description automatically generated with low confidence | | | |

Figure 9 Colour iso levels of the temperature distributions for MC-N, MC-R, MC-T, and MC-Z at = 3% and Re = 350. plane is 5 mm from the inlet.

**3.2 The effect of fin designs on the fluid flow pattern**

Figure 10 displays selected streamlines through MC-N, MC-R, MC-T, and MC-Z. The colour of the streamlines is based on the velocity magnitude. The working fluid is Al2O3 nanofluid at = 3% and the Reynolds number is 350. The streamlines for the MC-N and MC-R configurations follow a similar path but differ in that the magnitude of the velocity through MC-R is larger above the fin and smaller towards the bottom of the channel. Inserting the rectangular fin increases the wetted wall area, which produces more resistance to flow. This flow resistance is uneven across the planes in that the flow is channelled through narrower passages on both sides of the rectangular fin. As the flow runs through the path of least resistance, it flows preferentially through the top section of the channel, as shown by the higher MC-R velocity magnitude in this region. This is therefore mainly the effect of solid blockage due to the rectangular fin. The streamlines through MC-T and MC-Z differ both in velocity magnitude and direction compared to MC-N and MC-R. The nearly parallel flow in MC-N and MC-R is replaced by a three-dimensional flow, with vortex-like structures forming about the crests of the fins. This indicates that the throughflow is impeded by wake blockage that adds to the solid blockage effect. One vortex-like structure appears to form almost after every reversal of the slope of the fin in the plane. These slope reversals are more frequent in the zigzag fin geometry due to the smaller fin pitch compared to MC-T. It is therefore likely that this generates a greater wake blockage effect in the MC-Z configuration compared to the other configurations. Although this blockage raises the pressure drop in the microchannel and thus the pumping power required, as shown in Figure 12, it also provides secondary motion in the plane that promotes the transport of heat away from the microchannel walls, thus increasing the thermal mixing in the centre of the microchannel. Figure 10 also illustrates the influence of the hydrodynamic entry length in the MC-N and MC-R streamlines, which bend slightly towards the centre of the microchannel at the inlet. The hydrodynamic entry length is the axial distance over which the laminar flow develops a streamwise self-similar velocity profile [45]. In laminar flow, , which corresponds to 0.28 and 0.25 for MC-N and MC-R, respectively.

Diagram

Description automatically generated

Figure 10 Three-dimensional streamlines through MC-N, MC-R, MC-T, and MC-Z at = 3% Al2O3 nanoparticles and Re = 350.

Figure 11 illustrates the velocity distribution on the cross-sectional plane at mm from the inlet, for MC-N, MC-R, MC-T, and MC-Z at = 3% and at a Reynolds numbers of 350. In this composite figure, the coloured contours denote the velocity magnitude whereas the vectors denote the in-plane velocity, i.e., the two-dimensional velocity as projected onto the plane. The microchannel without fins, MC-N, develops a classical Hagen-Poiseuille type flow, featuring a maximum axial velocity at its geometric centre and a monotonic velocity decay towards its boundaries. Inserting a rectangular fin causes the flow to run preferentially through the top part of the channel, where the flow passage is wider and the flow resistance is lower, as discussed in the context of Figure 10. Consequently, MC-R reaches a higher flow speed above the fin than MC-N. Both MC-N and MC-R have very small in-plane velocity components, which are negligible compared to the in-plane motion in MC-T and MC-Z and effectively disappear from the figure when plotted at the same vector scaling length as MC-T and MC-Z. The in-plane velocity vectors for MC-T and MC-Z show a clockwise circulation centred to the right of the fin tip. This circulation is probably caused by the narrowing of the gap between the fin and the channel left wall and the concurrent widening of the gap between the fin and the channel right wall. The working fluid moves from the narrowing channel section towards the widening channel section, setting up a motion that resembles a wing tip vortex. It is important to note that the centre of the recirculation is offset with respect to the location of maximum velocity magnitude. This is likely to be a useful feature with regard to the cooling performance of the channel. It enables the transport of heated fluid away from the walls into a channel region in which a higher velocity can transport this heat away, towards the channel outlet.

|  |  |  |  |
| --- | --- | --- | --- |
| MC-N | MC-R | MC-T | MC-Z |
|  |  |  |  |
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Figure 11 Colour iso-levels of velocity magnitude and in-plane velocity vectors for MC-N, MC-R, MC-T, and MC-Z at = 3% Al2O3 nanoparticles and Re = 350. Cross-sections at a constant , 5 mm from the inlet.

The vortical flow structure produced by the twisted and zigzag fins are depicted in Figure 12 by iso-surfaces of the Q-criterion, which defines a vortex as a region where , i.e., where the norm of the vorticity tensor, , dominates that of the strain rate tensor, . The iso-surfaces are coloured by the velocity component , revealing the alternating direction of rotation of the vortices. These vortical structures appear to originate from the cusps of the twisted and zigzag fins, generating a rotational motion along an axis which gradually moves towards the widening portion of the channel. As the fin inverts its direction in , the vortical structure is accelerated towards the other side of the fin together with the bulk flow and becomes more elongated. The dissipation due to viscous effects and the generation of the counter-rotating vortex downstream weaken the previous structure until it fully decays. This pattern repeats consistently with every turn of the fin, which indicates that the effects of one vortex are not felt far downstream, but rather fade within the first fin inversion.

Diagram

Description automatically generated

Figure 12 The vortical flow structure for MC-N, MC-R, MC-T, and MC-Z at = 3% Al2O3 nanoparticles and Re = 350.

**3.3 The Effect of fin design on thermal characteristics**

Figure 13 shows the predicted change in Nusselt number with increasing Reynolds number from the MC-N, MC-R, MC-T, and MC-Z simulations at = 3% Al2O3 nanoparticles. For all four microchannel configurations, the average Nusselt number increases monotonically with Reynolds number. All microchannel configurations with a fin (MC-R, MC-T, and MC-Z) are predicted to give Nusselt number distributions above that of the microchannel with no fin (MC-N) at all Reynolds numbers. Inserting the fin delivers the largest gain in Nusselt number, with further smaller incremental gains in Nusselt number achieved by twisting the fin and then by using a zigzag fin. The main gain associated with the insertion of the fin is likely to be due to the increase in the microchannel internal surface area. Redistributing the heat transfer rate from the microchannel bottom over a larger surface reduces the heat flux through the fluid-solid interface, leading to a lower temperature difference between the flow and the walls. The further gain in the value of the Nusselt number associated with the twisted and zigzag fins is probably due to the secondary flow, as discussed in the context of Figure 10. This secondary flow motion generates heat convection streams not only in the channel bulk flow direction, , but also in both the vertical and transverse directions. This motion gives further homogeneous heating of the nanofluid through the microchannel cross-section, as shown by the more homogeneous temperature distributions in Figure 9, so that the thermal cooling capacity of the liquid coolant is better used. Below a Reynolds number of 250, the () narrow gap between the channel internal side wall and the twisted fin restricts the flow through it and hence constrains the convection of heat away from these surfaces. This is evidenced by the elevated coolant temperature through this gap, shown by MC-T in Figure 9, compared that through the () gap. This reduced convection offsets the benefit of the increased internal surface area of MC-T compared to MC-R. At Reynolds numbers above 250, the larger streamwise pressure gradient moves more coolant through the () narrow gap and confers a thermal performance advantage to MC-T compared to MC-R. With a volume fraction of 3% at a Reynolds number of 350, the Nusselt numbers of MC-Z, MC-T, and MC-R are increased by 60.01%, 46.26%, and 39.69%, respectively, compared to the MC-N configuration.



Figure 13 Change of the average Nusselt number with increasing Reynolds number for MC-N, MC-R, MC-T, and MC-Z with a volume fraction = 3% of Al2O3 nanoparticles.

Figure 6 illustrates how the difference between the microchannel bottom wall average temperature and the inlet temperature of the fluid varies with the Reynolds number. Predictions for the MC-N, MC-R, MC-T, and MC-Z configurations are reported, with = 3%. The microchannel bottom wall average temperature reduces with increasing Reynolds number, with and without fins. The presence of a fin produces a lower microchannel bottom wall temperature at any given Reynolds number over the range . At a Reynolds number of 350, the bottom wall average temperature for MC-Z, MC-T, and MC-R is predicted to be 9.15°C, 7.92°C and 6.44°C lower than that of MC-N, respectively. Such heat sink contact temperature reductions are of practical interest, since they can benefit the lifespan and the performance of electronic chips. For instance, Sahini, et al. [46] found that increasing the coolant inlet temperature from 25°C to 50°C of an Enterprise-class server increased the processor electric power consumption by 4%. According to equation (12), the total thermal resistance of the microchannel is equal to the temperature difference shown in Figure 6, divided by the bottom wall heat flux . Since the value of was held constant throughout these simulations, the total thermal resistance is simply proportional to and, therefore, it follows the same trend. At a Reynolds number 350 and = 3%, the average thermal resistances of MC-Z, MC-T, and MC-R are 40.03%, 34.64%, and 28.15% lower than that of MC-N, respectively.

**3.4 Effects of nanoparticle concentration on thermal characteristics**

The effect of changing the volume fraction of the nanoparticles was investigated for the microchannel configuration predicted to deliver the lowest bottom wall average temperature. For this configuration, the nanoparticle volume fractions = 0, 1, 2, and 3% were modelled. Figure 14 shows the variation of the Nusselt number with Reynolds number for the MC-Z configuration at nanoparticle volume fractions between 0 and 3%. Over this nanoparticle volume fraction range, the Nusselt number increases monotonically with the Reynolds number. Adding Al2O3 nanoparticles generates a modest increment in Nusselt number over the full Reynolds number range, an increment that is higher at higher nanoparticle concentrations. Specifically, the Nusselt number for MC-Z with = 1, 2 and 3% at a Reynolds number of 350 increases by 0.19%, 1.31%, and 0.67%, respectively. The Nusselt number expresses the ratio between the heat convection rate through the working fluid and the heat conduction rate. The small change in Nusselt number obtained by adding nanoparticles indicates that the flow remains substantially unchanged, that is, that the thermal transport mechanism established in pure water does not change substantially when the nanoparticles are introduced. This knowledge may have some practical use, for instance, it enables the characterization of the flow of heat and mass by applying a pure water numerical model. Once a satisfactory flow is found, then wall temperatures can be adjusted by adding small concentrations of nanoparticles without reiterating the flow pattern analysis. A comparison between Figure 13 and Figure 14 shows that the increase in Nusselt number obtained by changing the geometry of the channel is significantly higher than that obtained by introducing nanoparticles with a concentration up to 3%.



Figure 14 Change of Nusselt number with increasing Reynolds number for MC-Z at different volume concentrations of Al2O3 nanoparticles.

The effect of adding nanoparticles to the coolant is more substantial on the microchannel bottom wall temperature than on the Nusselt number. Figure 15 shows the difference between the fluid inlet and the bottom wall average temperatures for the MC-Z configuration at the four nanoparticle volume fraction concentrations from 0% to 3%. The bottom wall average temperature approaches the inlet temperature with increasing Reynolds number. The lowest temperature difference is achieved at the highest nanoparticle concentration of = 3%. This improvement is because of the Al2O3 nanoparticles having a higher thermal conductivity and a lower specific heat capacity than the base fluid, so that they can absorb more quickly heat per unit volume and transport it out of the microchannel heat sink. Comparing Figure 14 and Figure 15 shows that, whereas only a modest change in Nusselt number is obtained by the addition of nanoparticles within the concentration range 0% to 3%, a useful decrease in microchannel bottom wall average temperature can be achieved. The definition of the average Nusselt number from equation (11) seems to suggest that the thermal conductivity of the modelled nanofluid is of comparatively greater importance compared to its specific heat capacity effect. This drives a change in the average convective heat transfer coefficient, , that in turn, by Newton’s law of convection, affects the microchannel thermal resistance and hence determines a useful change in the bottom wall temperature. The range of such change, about 6°C in Figure 15, is lower but comparable to the change obtained by introducing a fin, about 10°C, as shown in Figure 6.



Figure 15 Difference between the microchannel bottom wall average temperature and the working fluid inflow temperature as a function of Reynolds number for MC-Z at different volume concentrations of the Al2O3 nanoparticles.

**3.5 Effects of fin design on hydraulic characteristics**

Figure 7 shows the predicted pressure drop over the Reynolds number range for MC-N, MC-R, MC-T, and MC-Z at = 3% volume fraction of Al2O3 nanoparticles. MC-Z is predicted to have the largest pressure drop among the four microchannel configurations. This larger pressure drop is a result of increased flow resistance. As discussed in the context of Figure 10, both solid blockage and wake blockage effects combine in the zigzag fin case MC-Z to produce a larger hydraulic head loss. The bulk motion of the flow in , which is driven by the drop in pressure, , is transformed into secondary flow motion by deflection by the fin, as shown in Figure 10. Some of this motion is then dissipated by viscosity, both along the wall and within the vortical structures. MC-Z, MC-T, and MC-R all show a higher resistance to flow, that is, a hydraulic performance penalty compared to the microchannel without fins, MC-N. At the Reynolds number of 350 and = 3%, the predicted pressure drop for MC-Z, MC-T, and MC-R increases by 147.1%, 114.87%, and 118.51%, respectively, as compared to MC-N.

**3.6 Effects of nanoparticle concentration on hydraulic characteristics**

Figure 16 shows the variation in the friction factor, , with Reynolds numbers for MC-N, MC-R, MC-T, and MC-Z with = 3% of Al2O3 nanoparticles. Consistent with Figure 7, MC-Z is predicted to have the largest friction factor among all microchannel configurations due to it having the largest pressure drop at the same dynamic pressure and channel length . MC-N has the lowest friction factor among all the microchannel configurations over the entire Reynolds number range investigated, which reflects its lowest wetted microchannel wall surface area and mainly axial () flow. The friction factor decreases with increasing Reynolds number for all microchannel configurations as a negative power of the Reynolds number. For reference, Figure 16 shows the Darcy-Weisbach friction factor for fully developed laminar flows in a rectangular duct [45]. This equation is only appropriate to the hydrodynamically fully developed region. Hence, the entry length,, is estimated to calculate the hydrodynamic entrance region [45] and the friction factor for MC-N with = 0% of Al2O3 nanoparticles. The simulated friction factor shows the expected scaling with the inverse of the Reynolds number, in agreement with the friction factor from the Hagen-Poiseuille law. The Darcy-Weisbach friction factor curve is below all predicted friction factors for the finned microchannel configurations, but it is also below that for the no fin MC-N, with = 3% Al2O3 nanoparticles. The presence of nanoparticles produces a steeper near-wall velocity gradient; this causes additional friction that is not accounted for by the fully developed laminar flow of the Darcy-Weisbach model. At the Reynolds number of 100 and ϕ = 3%, the friction factors for MC-Z, MC-T, and MC-R are 72.10%, 49.65%, and 52.19% greater than for MC-N, respectively.



Figure 16 Change of friction factor with increasing Reynolds number for MC-N, MC-R, MC-T, and MC-Z at = 3% of Al2O3 nanoparticles.

**3.7 Overall performance improvement**

Figure 17 compares the Figure of Merit (FoM) for all microchannel configurations (MC-N, MC-R, MC-T, and MC-Z), nanoparticle concentrations, and Reynolds numbers modelled in this work. The intent is to assess the merit of using either a rectangular, a twisted, or a zigzag fin, and/or of adding different volume fractions of Al2O3 nanoparticles. The FoM is defined against the reference case MC-N at = 0%, for all Reynolds numbers. The numerical results indicate that all proposed designs have a FoM > 1.0, indicating that they provide comparatively more heat convection performance than the flow resistance penalty they generate. Configuration MC-R reaches a maximum FoM around Re = 250, whereas MC-T and MC-Z increase monotonically with Reynolds number. The mechanism responsible for the enhancement due to the twisted or zigzag fin, as compared to the rectangular fin design, can be explained by the generation of a secondary flow, as discussed in the context of Figure 12. This shows how the effects of the microchannel design parameters investigated in this study are strongly dependent on one another and suggests that the design optimization process needs to properly consider these coupled effects.

Overall, for applications in which the electrical resistance increases with temperature, such as for copper-based circuits, a reduced electrical energy consumption from running the circuit at a lower temperature may offset the increase in the pumping energy required through the microchannel heat sink. In such cases, the higher FoM may indicate a better overall energy efficiency. For such applications, Figure 17 indicates that the best configuration is that of MC-Z. Moreover, a greater FoM occurs at a Reynolds numbers of 350 for all proposed designs. All values of FoM for MC-R, MC-T, and MC-Z are slightly reduced by increasing the volume fraction concentration of the nanoparticles, which results in greater viscous loss due to friction and in a greater pressure drop, as shown in Figure 16.

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Figure 17 Change of Figure of Merit with increasing Reynolds number for MC-N, MC-R, MC-T, and MC-Z, at = 1, 2 and 3% of Al2O3 nanoparticles, referenced to the performance of MC-N at = 0%.

**3.8 Exergistic characteristics of the microchannel heat sink**

Figure 18 illustrates how the exergy expenditure rate by the microchannel heat sink varies with Reynolds number in MC-N, MC-R, MC-T, and MC-Z at = 0% of Al2O3 nanoparticles. The global trend is not monotone, as the rate of exergy expenditure decreases to a minimum around Re = 150 and then increases for higher Reynolds numbers. As shown in Figure 18, this is due to the opposing effects that the Reynolds number has on the two integrals in equation (18), associated respectively with the reversible work potential of the heat supplied from the hot source and with the reversible work potential of the supplied coolant, respectively. In the first integral, the reduced average bottom wall temperature lowers the exergy rate supplied, , whereas in the second integral the increased kinetic energy of the flow introduces additional exergy, . Overall, the finned configurations (MC-R, MC-T, and MC-Z) require a lower rate of exergy expenditure than the MC-N channel. The MC-Z configuration has the lowest rate of exergy expenditure due to its lowest bottom wall temperature.

  

Figure 18 The global exergy expenditure rate (bottom) for MC-N, MC-R, MC-T, and MC-Z at ϕ = 0%. This is the sum of the exergy expenditure rate from the loss of reversible work potential of the coolant, (middle), and that from the loss of reversible work potential by the heat supplied by the hot source, (top).

Figure 19 (a) shows that the exergy destruction rate decreases monotonically with Reynolds number for all configurations (MC-N, MC-R, MC-T, and MC-Z) for = 0%. This is due to the decrease in the average bottom wall temperature at higher Reynolds numbers that reduces in equation (19). This term appears to have the strongest effect on , with the second term (), associated with the exergy of the coolant only providing a second-order contribution. The MC-Z configuration provides the lowest rate of exergy destruction among the four configurations, due to its lowest bottom wall average temperature. The second law efficiency, , based on the exergy expenditure and destruction rates, is shown in Figure 19 (b). The microchannel with a straight fin provides a second law efficiency increase of 14% compared to MC-N, at the Reynolds number of 350. With a zigzag fin, the corresponding increment is 15% and, with a twisted fin, it is 12%. Similar results are obtained at lower Reynolds numbers. This means that the reduction in exergy destruction rate due to the presence of the fin outweighs the associated reduction in exergy expenditure rate, yielding a better second law efficiency overall.



(a)



(b)

Figure 19 (a) Exergy destruction rate and (b) second law efficiency for MC-N, MC-R, MC-T, and MC-Z at ϕ = 0%.

# Conclusions

This paper numerically investigated three different fin designs (rectangular, twisted, and zigzag) for a microchannel heat sink, in combination with a nanofluid as working fluid. The main conclusions from this study are:

1. The lowest microchannel bottom wall average temperature for a given coolant inflow temperature was predicted with the zigzag fin configuration operating at the highest Reynolds number (350) and with the highest nanoparticle concentration (3%). This reduced base temperature not only is desirable to keep electronic components from overheating, but is also shown to yield an overall increase in second law efficiency.
2. The advantage of the zigzag design, compared to the other fin designs, is suggested to be due to the combination of an increase in the microchannel internal surface area, leading to a lower temperature difference between the flow and the walls, and of an induced secondary flow, which promotes the fluid exchange and therefore the heat transport between the warmer peripheral regions near the walls and the cooler centre of the microchannel. Based on these considerations, a further study on alternative designs could potentially deliver even more effective configurations.
3. The addition of nanoparticles produced a useful decrease in the bottom wall average temperature, thanks to the increased convective heat transfer coefficient of the working fluid. However, the change in Nusselt number was modest, due to the concurrent increase in the fluid thermal conductivity. It is therefore important to identify the purpose of the design optimization (in this case the reduction of the temperature of the electronic component) to properly guide the design process.
4. The rate of heat removal to hydraulic power loss ratio identified by the Figure of Merit is a useful indicator of the overall performance where the solutions adopted to increase the thermal performance also tend to increase pumping power requirements, as it combines the different effects produced by fin configuration, nanoparticle concentration, and Reynolds number.

These observations justify the use of the bespoke model developed in this work that combines different aspects (geometry, nanofluid coolant, and Reynolds number) in one joint optimization process for the thermal and hydrodynamic performance of a microchannel heat sink.

# Acknowledgements

Abdullah Masoud Ali received a PhD scholarship by Libyan Ministry of Higher Education and Scientific Research to perform this research work. Core time on the ALICE High Performance Computing Facility was provided as support in kind from the University of Leicester. The authors are grateful for the guidance received on High Performance Computing from the Research Software Engineering team established under HPC Midlands Plus, funded by EPSRC EP/K000063/1 and EP/P020232/1. The flow visualizations made use of Tecplot licenses purchased under EPSRC GR/N23745/01.

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