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Utilization of Nanofluids in Minichannel for Heat Transfer and Fluid Flow Augmentation: A Concise Research Design

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ABSTRACT

The current trend of size reduction of electronic devices and heat exchangers to enhance their performance and energy conservation is pushing the limit of their heat transfer enhancement capabilities. Conventional fluids failed to provide required heat removal from high heat flux generating electronic devices and heat exchangers, due to their inherent low thermal conductivity. Nanofluid is an advance innovative thermal engineering fluid capable of providing outstanding heat transfer improvement than the conventional fluids, thus, increasing thermal system productivity and ensure energy sustainability. In this paper, the design of a numerical study of the cooling application of high heat flux dissipating devices is proposed using hybrid passive techniques of using nanofluids and the corrugated (Diverging-converging) minichannel heat sink to determine the performance of heat transfer and flow behavior. The result expected to be achieved at the end of successful conduct of this research include: declaration of superiority of nanofluid over base fluid on improvement of heat transfer rate, and consequently enhanced convective heat transfer coefficient (HTC), minimal pressure drops which may not necessarily demand more pumping power of working fluid and reasonable level of thermal resistance.

1. Introduction

Rapid technological advances in the areas of the electronic device provide the system with high performance and energy conservation, but with highly concentrated heat flux which affects the efficiency and reduces the Mean time before failure (MTBF). The conventional cooling system of air and liquids failed to offer enough cooling effect. Nanofluid technology is regarded as one of the key emerging technologies that are presently attracting great research efforts in thermal engineering with the aim to provide improved working fluid for efficient thermal dissipation from high heat flux generating devices.

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the use of micro (MC) and minichannels (MiC) in heat sinks was pioneered by Tuckerman and Pearse [1]. It's distinguished from the conventional channels in terms of channel hydraulic diameters. They postulated that reduction in channel hydraulic diameter can increase the heat transfer coefficient. The minichannel is usually within 200µm to 3mm hydraulic diameter based on Kandlikar and Grande classification scheme that distinguished the channels based on manufacturing restrictions and the Knudsen number. [2]. Though microchannel offers higher heat transfer enhancement than minichannel, its smaller hydraulic diameter leads to increase pumping power and pressure drop, as well as high cost and more sophisticated manufacturing techniques [3], thus, minichannel still receive interest for utilization in heat exchangers and heat sinks.

Many researchers have shown remarkable achievement of nanofluids in their works, such as in heat exchangers [4-6], electronic cooling [7], solar energy harvest [8], refrigeration and energy recovery [9, 10] and other applications. Some researchers compiled an extensive review of literature in relation to utilization of nanofluid in minichannel as a passive means of heat transfer enhancement [11-14]. It is the view of the authors of this work that, the research in this field is contemporary and dynamic, hence more insight into the state-of-the-art techniques and methods for further research on heat dissipation in electronic micro-devices and heat exchangers in industries is necessary. The objective of this paper is to propose a systematic approach in conducting numerical research on hydrothermal performance analysis using nanofluid and minichannel as a hybrid technique for heat transfer enhancement.

2. Methodology

A comprehensive review of related works and expression of researches conducted in this area was conducted to have a better understanding of the concepts involved in the study of the hydrothermal analysis of nanofluids in minichannel thermal devices with emphasis on heat transfer enhancement mechanisms and techniques employed, and achievement of thermal improvement. First, a classification of nanofluid was discussed, then the techniques used in heat transfer enhancement, as well as the numerical method employed in the study, were overviewed.

2.1 Classification of nanofluids.

Nanofluids are normally produced by dispersing powdered nanoparticles (NP) into the base fluid in two distinct methods, these include one step and two methods. Various Nanoparticle materials used in nanofluids production include metals (Cu, Ag, Au), oxide ceramics (Al_2O_3, CuO), nitride ceramics (AlN, SiN), carbide ceramics (SiC, TiC), semiconductors (TiO_2 and SiO_2) and carbon-based (Carbon nanotubes and Graphene). In addition, a combination of two or more nanofluids provides a hybrid nanofluid, which shows a better enhancement than the individual nanofluids that formed it, though with increased viscosity which sometimes reduces the level of enhancement.

Most researchers observed that, nanofluid has higher surface to volume ratio than the base fluid, hence, adding nanoparticles (NPs) usually in size of 1 – 100 nm in a base fluid can considerably improve heat transfer rate and consequently enhanced convective heat transfer coefficient (HTC), however, with a drawback on pressure drop, which subsequently demands more pumping power of working fluid. Other important factors of consideration include long-term stability and agglomeration of nanoscale to macroscale particles, which may block and erode the minichannel surface.

Few researchers used metallic NPs such as Bahiraei and Heshmatian [15] dispersed spherical silver NP of 40 – 50 nm in water (Ag-H_2O) to evaluate hydrothermal characteristics and entropy generation of a biological nanofluid in a liquid block heat sink for cooling of an electronic processor.

The result at a concentration of 1% and Reynolds number of 500, indicates a temperature reduction of 2.21°C for the NP against water. Investigation of corrugation effect on the flow and thermal characteristics of Au-H_2O nanofluid in the wavy channel was conducted by [16] using concentrations of 0% - 5% and Re 250-1500. They highlighted that the use of a wavy channel with a 90° phase shift is not desirable to dissipate heat from the devices. Triangular channel gave better enhancement, then by sinusoidal at 45°, 90°, and 135° phase shift. Azwadi and Adamu [17] investigated the effect of Silver-Graphene (Ag/HEG) and Copper-oxide Graphene (CuO/HEG) nanofluids in a circular channel under constant heat flux within turbulence regime using concentration of 0.4 – 1 vol%. and Re 10000 - 120000. At 1vol.%, enhancement of 34.34% and 38.72% were obtained for Ag/HEG at Reynold numbers of 60000 and 40000, respectively. similarly, 35.95% and 43.96% were obtained for CuO/HEG at the same Reynolds number and concentration respectively.

The commonest used *non-metallic* nanoparticles are Alumina (Al_2O_3), Titania (TiO_2) and Silica (SiO_2), with Alumina as the most widely preferred by most of the researchers due to its lower density and viscosity as well as increased reactivity when compared with other conventional nano-sized particles. Dominic et al. [18] used Alumina with 40 nm particle size, volume fraction of 0.5% and 0.8% and at Re of 700 – 3300 to investigate heat transfer and pressure drop between wavy divergent and wavy cross-sections and reported that, in the laminar regime, the heat transfer performance of divergent wavy minichannels was 9% higher and the pressure drop was 30–38 % lesser than that of the wavy minichannels having constant cross-section. The performance factor of divergent wavy minichannels was 110–113 % for nanofluids compared to 115–126 % for water. Zhou et al. [19] confirmed Alumina enhanced the heat transfer performance and the average saturated flow boiling HTC of specified concentrations of nanofluid respectively increased by 11.2%, 15.4% and 18.7% in comparison with deionized water.

Hybrid nanofluid and Carbon based nanorods and flakes like Carbon Nanotube (Single [20] and multi-walled CNT [21]) and Graphene are receiving interest from researchers. Shahsavar et al. [22] studied thermal and hydraulic characteristics of a non-Newtonian hybrid nanofluid Fe_3O_4 with different concentrations of 0.5–0.9% and 0.1 –1.1%, respectively and found that increasing Fe_3O_4 and CNT concentrations enhance the convective HTC of inner and outer walls and total entropy generation. Diao et al. [23] show that heat transfer improved with an increase in concentration at 0.01% or above but degenerated when concentration falls below 0.01% when they studied the thermo-hydraulic performance of Multi-walled CNT (MWCNT) passing through multi-port minichannel (MPMiC). It can be construed from the works discussed that, among the NPs, Cu has better enhancements, followed by Al_2O_3 and TiO_2 in terms of heat dissipation capability.

2.2 Heat transfer enhancement techniques

The mechanism used to enhance heat transfer without upsetting the overall thermo-hydraulic performance of the thermal system are simply categorized into active and passive methods. The later involves modifying properties and structure of the heating surface by increasing the effective surface area and residence time of the thermal fluid and has exhibited advance energy efficiency and material saving, while the former, demands some external power input for the heat transfer enhancement, it is rarely employed due to energy conservation nowadays. Few available types of research that employed active method include; Ozbey et al. [24], Mohammadpourfard [25] Naphon and Klangchart [26].

Dominic et al. [27] observed that passive method for forced convective heat transfer enhancement can be achieved through a decrease in thickness of thermal boundary layer, increase in fluid interruption and an increase in velocity gradient near a heat transfer wall. Furthermore, investigators observed that reduction of hydraulic diameter and higher heat transfer surface area per unit fluid volume of nanoparticles can effectively remove excess heat and improves heat transfer coefficient (HTC), thus, a lot of methods were introduced by changing minichannel geometrical parameters, such as channel number, aspect ratio, cross-sections, and path configurations.

2.3 Method of heat transfer analysis:

Heat transfer analysis like in other science and engineering fields employ experiments, numerical simulations and theoretical methods as tools to support research and development. The experimental method is more reliable, but factors such as speed, cost, repeatability, and safety, coupled with recent technological advancement and wide-spread access to computers make the simulation more preferred than experimental measurements or theoretical analysis. Some of the prospects and challenges of these methods were highlighted by [28, 29].

Solutions of governing equations in numerical methods known as discretization normally involves a Finite Volume Method (FVM), Finite Element Method (FEM) and Finite Difference Method (FDM). The FVM and FEM methods are applicable to both structured and unstructured grids, while FDM uses only structured grids. [30] Mostly researchers in thermo-hydraulic analyses employ Finite Volume Method (FVM). It is based on the integral form of governing equations and solves for the represented variable value for the cell volume and the governing equations used to include that of mass, momentum, and energy for steady-state laminar incompressible fluid usually in non-dimensional form. The governing equations can be expressed either in polar or cartesian coordinates for cylindrical and channel geometries, respectively.

Some researchers considered nanofluid as a homogenous mixture and they used the single-phase model to study heat transfer performances of NP with various thermophysical parameters. Classical theories [31-35] were employed for the study, however, mostly these classical theories failed to adequately solve thermo-hydraulic properties, hence some researchers proposed some correlations [36, 37]. Pati et al. [38] used the single-phase model to investigate the thermohydraulic performance of two different configurations of the sinusoidal wavy-walled channel formed by changing phase shift angles between the bi-opposite heated walls. They observed that heat transfer depends on the geometry of the wall and highly controlled by the wavelength of the wall waviness. Moraveji et al. [39] modelled laminar forced convection on Al_2O_3 nanofluid with size particles of 33 nm and concentrations of 0.5, 1 and 6 wt.% within Re of 130 - 1600 in mini-channel heat sink performed in CFD by four individual approaches (single phase, VOF, mixture, Eulerian) of FVM.

Mostly, works on nanofluid for heat transfer enhancements were numerically solved using two-phase models with available approaches such as Eulerian- Eulerian (compose of Eulerian, mixture and volume of fluid (VOF)) and Lagrangian- Eulerian, with an accurate prediction of the models indicated by researchers [40-42]. Naphon and Nakharintr [43] studied laminar convective heat transfer of single and two-phase models in 3D using TiO-H_2O with PS 21 nm, VF 0.4 vol% and Re 80 – 200 in mini-rectangular fin heat sinks made of copper. The two-phase numerical model gave better enhancement than experimental single-phase results for all the Reynolds numbers. Saeed and Kim [44] studied numerically the thermo-hydraulic performance of Al_2O_3-H_2O in mini-channel heat sinks with four different channel configurations using single-phase and two-phase models and observed that two-phase mixture model predicted results agreed closely with an experimental model while single phase numerical model has underpredicted values of convective HTC.

3. Methodology

3.1 Problem statement and Mathematical modeling

The main scope of the currently proposed simulation is to illustrate the performance of the diverging-converging minichannel in term of heat transfer and flow characteristics by studying the effect of nanofluid types, concentration, and thermophysical properties. To achieve this, proper modeling of the minichannel was conducted. Also, appropriate models were employed to evaluate the thermophysical properties of the nanofluids, since, the classical "effective medium theory" proposed by Einstein [33] failed to accurately predict the behavior of the nanoparticles especially at a concentration above 0.2%.

3.1.1 Model geometry

The physical model considered in the numerical study is depicted in fig. 2 shows the schematic design of the divergent-convergent minichannel heat sink (DCMCHS) with dimensions of $L \times W \times H_b$=30mm×30 mm×2.25 mm having 10 parallel channels each of width and height of 1mm and 1.25mm, respectively. Since the minichannels are made-up to be identical in terms of both heat transfer and hydrodynamics, thus, only one of the channels is used as a computational domain as shown in figure 2(b).

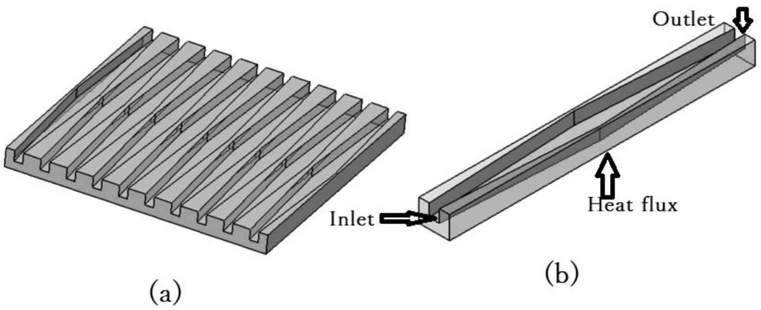


Fig. 1 (a) 3D Schematic design of DCMCHS and (b) Computational domain

The hydraulic diameter is determined from the trapezoidal section at the midplane of either the divergent or convergent section [45]. Thus: the hydraulic diameter can be obtained as

$$D_h = \frac{4\mathrm{A}}{\mathrm{P}} = \frac{2(\acute{\mathrm{W}}\cdot\dot{H})}{\acute{\mathrm{W}}\cdot\dot{H}} \tag{1}$$

where: $\dot{H}$, W_t and W_b represent the slant height, widths at the top and bottom sections of the fluid domain. Table 1 illustrated the geometrical values used in the study.

3.1.2 Governing equations

The steady-state conservation equations for mass, momentum, and energy in the fluid are respectively presented in a non-dimensional form as follows:
The continuity equation is given by

$$\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y} + \frac{\partial u}{\partial z} = 0 \tag{2}$$

whereas momentum equations in x, y and z components, respectively are given as:

$$\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \tag{3a}$$
$$\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \tag{3b}$$
$$\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{1}{\rho}\frac{\partial p}{\partial z}+\frac{\mu}{\rho}\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{3c}$$

where u, v and w are the velocity components in x, y and z directions, respectively. In addition, the pressure drop, weight density and dynamic viscosity of the fluid are represented with p, ρ, and μ, respectively.

Energy equation for the fluid

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\frac{k_f}{\rho Cp}+\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \tag{4}$$

Energy equation for the solid:

$$k_s\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right)=0 \tag{5}$$

where T, k_f, and k_s are temperature, and thermal conductivities of fluid and solid materials, respectively.

3.1.3 Boundary conditions

Boundary conditions were applied to close the above mathematical equations. Inlet Temperature and pressure of the nanofluid are specified as 30°C (303K) and 1 bar, respectively, while “Pressure outlet” is imposed at the outlet with 0 Pa (gauge pressure). A constant heat flux of 45 kW/m^2 is applied on the bottom wall while, for all other walls, no-slip condition (viscous flow) is experienced, thus velocity gradient forms in fluid and exerts flow resistance as pressure drop. All variables are initiated from the inlet boundary condition.

3.1.4 Thermophysical properties of the fluid

The working fluids used in this simulation are Alumina (Al_2O_3), Silica (SiO_2) and Copper (Cu) nanofluids dispersed in deionized water with volume fractions of 0.001, 0.005 and 0.008. The properties of nanofluid and the base fluid (water) considered in the present study are obtained from [46] as presented in Table 1.

Table 1
Thermophysical properties of water and nanoparticles at Temperature of 27°C [46]

Materials	Density (kg/m^3)	Specific heat (J/kgK)	Thermal conductivity (W/mK)	Viscosity (kg/ms)	Particle size (nm)
Water (H_2O)	995.8	4178.4	0.615	8.03E-04	-
Alumina (Al_2O_3)	3970	765	36	-	<50
Copper (Cu)	8933	385	401	-	<50
SiO_2	2220	745	1.38	-	<50

The thermo-physical properties of nanofluid are calculated using the relations below

The model developed by Pak et al. [47] was used to evaluate the density of nanofluids, whereas the specific heat was calculated using Xuan and Roetzel [48]

$$\rho = (1-\phi)\rho_{bf} + \phi\rho_p \tag{6}$$

$$Cp_{nf} = \frac{\phi(\rho Cp)_p + (1-\phi)(\rho Cp)_{bf}}{\rho_{nf}} \tag{7}$$

The thermal conductivity of the nanofluid (k_{nf}) was evaluated using the Maxwell model [49] for nanofluids with volume fraction less than unity and considered Brownian motion. It is given by equation (8) as follows

$$k_{nf} = \frac{k_p + (n-1)k_{bf} - \phi(n-1)(k_{bf}-k_p)}{k_p + (n-1)k_{bf} + \phi(k_{bf}-k_p)} k_{bf} \tag{8}$$

The viscosity of the nanofluids was calculated using the viscosity correlation proposed by Maiga *et al.*, [50] as follows

$$\mu_{nf} = \mu_w(1 + 7.3\phi + 123\phi^2) \tag{9}$$

where, φ, Cp, k and ρ are the concentration of the nanoparticle, heat capacity, thermal conductivity, viscosity, and density respectively. While subscripts bf, p, and nf denote the base fluid, the nanoparticle and nanofluid respectively.

3.2 Numerical approach

The three-dimensional forced convection flow and heat transfer were modeled using commercial CFD solver, ANSYS FLUENT 17 with the assumptions that: the working fluid is considered three-dimensional, incompressible, and Newtonian. It is flowing in steady state laminar condition. The thermophysical properties of heat sink and fluid are constant, while the effect of radiation heat transfer for fluid flow and the influence of gravity and other body forces are neglected. The finite volume method approach was employed in the simulation, where the velocity and pressure fields were coupled using the SIMPLE algorithm. A second-order upwind interpolation scheme is used for discretization of the convective and diffusive terms. The convergence criteria were set when the normalized residual values are below 10^{-6} for all the variables.

3.2.1 Grid independence test

A hexahedral mapped mesh was used for all the simulations. Various grids of sizes from 600,000 to 1.228 million elements were used in checking the mesh independence of the solution to ensure that, the results obtained do not rely on the size and the number of generated cells. The variation in the Nusselt numbers between the solutions on the grids with 0.90 million and 1.2 million elements is found to be below 0.5%, hence, to save computing time and memory, grid sizes of around 0.90 million elements were used for all the simulations in this study. The mesh of the fluid domain is illustrated in fig. 3.

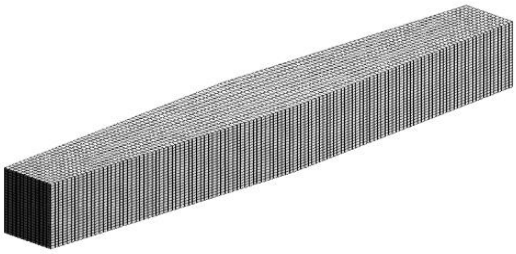
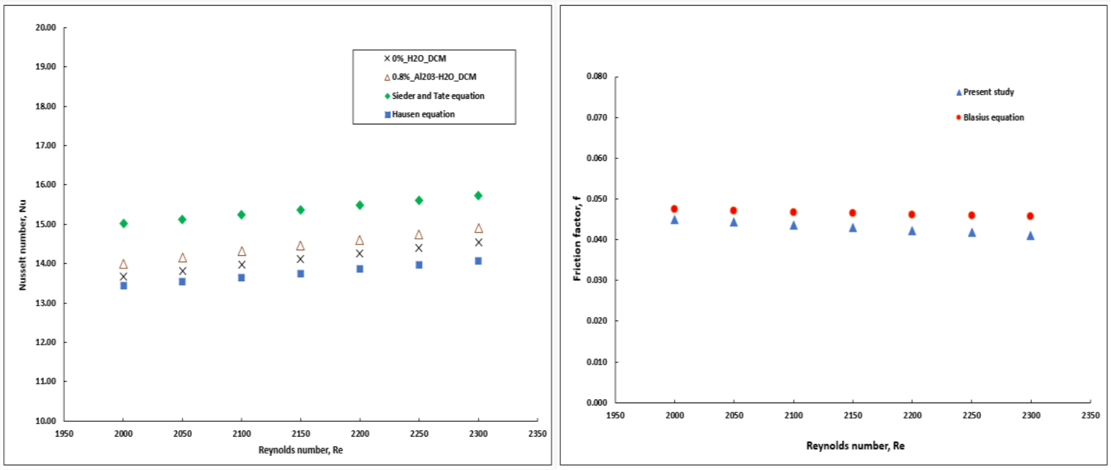


Fig. 2. 3D Mesh of the fluid domain

3.2.2 Validation

The precision and validity of the numerical results were validated through comparison with existing correlations in the literature due to non-availability of experimental results in diverging-converging minichannels. Sieder and Tate, and Hausen correlations [51] for the fully developed laminar region were employed for Nusselt number enhancement, while Blasius relation [52] was used for frictional resistance; to substantiate the ability of the solver to accurately and reliably predict the results.



(a) (b)

Fig. 3 Validation of results of (a) Nusselt number and (b) friction factor as functions of Reynolds number

It can be observed from fig. 4 (a) that, Sieder and Tate correlation over predict the average Nusselt number at Re 2300 by about 7.4% for the base fluid. However, Hausen correlation underpredicts the average Nusselt number at the same Reynolds number by 3.5% for the base fluid. Since the deviations of the average Nusselt numbers from the two correlations used is within ±10%, it can be considered that the numerical results were appreciably well predicted by the method employed. For the friction factor, fig. 4 (b) indicates that, the numerical friction factor of the DCMCHS in the fluid shows appreciable agreement with the correlated results of Blasius, however, the deviation of the numerical friction factor from the theoretical values of nanofluid is around 5% and 10% lower at 2000 and 2300 Reynolds numbers, respectively. Perhaps, this could be due to the increase of pressure drops as the flow velocity increases at the entrance of the channel, and to the assumptions made in the mathematical formulation of the simulation.

3.3 Data processing

The following equations (10-15) would be used to estimate the different important thermal and flow parameters of the minichannel heat sink.

Table 2
Parameters used in the evaluations

Parameter	Expression	Equation no
Heat transfer coefficient (h)	$h = \frac{q}{(T_w - T_b)}$	(10)
Nusselt number (Nu)	$Nu = \frac{hD_h}{k}$	(11)
Reynolds number	$Re = \frac{\rho u D_h}{\mu}$	(12)
nanofluid velocity	$u_{nf} = \frac{\rho_{bf}}{\rho_{nf}} \frac{\mu_{nf}}{\mu_{bf}} u_{bf}$	(13)
pressure drop	$\Delta P = \frac{f\rho u^2}{2}\left(\frac{L}{D_h}\right)$	(14)

The performance of flow in the minichannel is evaluated using pumping power which can be determined as a function of the differential pressure drop, frictional resistance, and velocity of the flow. It's given as

$$PP = \Delta P \cdot f \cdot u \tag{15a}$$

However, some other expression of pumping power existed in the literature, such as:

$$PP = \Delta P \cdot \dot{V} \tag{15b}$$

$$PP = \Delta P \cdot \left(\frac{\dot{m}}{\rho}\right) \tag{15c}$$

where q, D_h, L, u, k, T_w, and T_b are the heat flux, hydraulic diameter, channel length, fluid velocity, the thermal conductivity of the fluid, average temperatures on the wall and bulk fluid respectively. Darcy friction factor (f) can be obtained from equation (14).

4. Conclusions

Numerical analysis of heat transfer and fluid flow characteristics of the divergent-convergent minichannel heat sink (DCMCHS) has been proposed using water-based nanofluids, and the following conclusions were made based on the expected results:

- There would be a significant influence of Reynolds number on heat transfer enhancement on both the base fluid and the nanofluid. Deceleration and acceleration of the flow at the middle of the channel cause recirculation and vortices creation which enhances flow mixing.

- There would be a slight increase in friction factor which may be linked to non-uniformity of channel passage due to divergence and convergence nature, and perhaps, decreases with increase in Reynolds number. The trend would be similar across all the volume fractions.
- Heat transfer coefficient would be enhanced by certain magnitude, say 15% due to the influence of nanofluid in DCMCHS and increase in thermal conductivity of the nanofluid over water.
- The Performance factor which is a factor to indicate the augmentation in heat transfer of nanofluid over base fluid would be expected to be above unity.
- Better enhancement in heat transfer and hydrodynamic influence could be achieved and well predicted if the numerical analysis could be extended to the turbulent regime and moderately higher volume concentration, say up to 4 %.

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