A comprehensive review on heat transfer enhancement and pressure drop characteristics of nanofluid flow through micro-channels

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ABSTRACT. *The purpose of this study is to comprehensively review the experimental and numerical studies of heat transfer and pressure drop with different types of ribs and different types of working fluid through micro-channels. In micro-channels, by adding different metallic oxide nanoparticles in the base fluid considerably improve the heat transfer rates as compared to a base fluid, however, the increase in the friction factor is insignificant. The ribs, cavities, porous surfaces, dimple surfaces, and groove structures create the obstacles or disruptions in the flow field. It is analyzed that these obstacles inside the microchannel are helpful to augment the heat transfer rate due to better mixing with a small increase in pressure drop. The purpose of this review paper is to encourage the researchers to pay more attention in the field of heat transfer augmentation and lessening the pressure drop to improve the performance of the thermal system. Lastly, some ideas for future work are also explored.*

RÉSUMÉ. *Le but de cette étude est de passer en revue les études expérimentales et numériques sur le transfert thermique et la chute de pression avec différents types de nervures et différents types de fluide de travail par l'intermédiaire de microcanaux. Dans les microcanaux, l'ajout de différentes nanoparticules d'oxydes métalliques dans le fluide de base améliore considérablement les vitesses de transfert thermique par rapport à un fluide de base. Toutefois, l'augmentation du facteur de friction est insignifiante. Les nervures, les cavités, les surfaces poreuses, les surfaces de fossettes et les structures de rainures créent les obstacles ou les perturbations dans le champ de flux. Il est analysé que ces obstacles à l'intérieur du microcanal sont utiles pour augmenter le taux de transfert thermique grâce à un meilleur mélange avec une légère augmentation de la chute de charge. Le présent article de synthèse a pour but d'encourager les chercheurs à accorder plus d'attention à l'augmentation du transfert thermique et à la réduction de la chute de pression afin d'améliorer les performances du système thermique. Enfin, quelques pistes de travail futur sont également explorées.*

KEYWORDS: *nanofluid, micro-channel, heat transfer enhancement, pressure drop.*

MOTS-CLÉS: *nanofluide, microcanal, amélioration du transfert thermique, chute de pression.*

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1. Introduction

1.1. Importance of microchannel

The ability of Micro-Electro-Mechanical Systems (MEMS) to perform their functions with good safety and reliability is mainly dependent upon their effectiveness to quickly transport the heat away from the localized heat source. The electric current flowing through an electronic component is always accompanied by heat generation attributed to the Joule heating effects. When thousands or even millions of electronic components are packed in a small volume, the heat generated increases to such high levels that its removal becomes a terrible task and the main concern for the safety and reliability of the electronic devices. As an example, to accommodate a heat flux of 100 W/cm^2 at a temperature difference of 50 K in an IC chip requires an effective heat transfer coefficient of 20,000 $W/(m^2{\cdot}K)$. In 1993, a high level of heat dissipation i.e. 200 W/cm^2 seemed to be an impossible task now seems to be a feasible target. The new challenge for the next decade is on the order of 600–1000 W/cm^2. These high levels of heat dissipation want a dramatic reduction in the channel dimensions, coordinated with appropriate coolant loop systems to ease the fluid movement away from the heat source (Kandlikar *et al.*, 2005). A microchannel heat sink (MCHS) is a passive heat exchanger that transfers the heat generated by an electronic or a mechanical device to a fluid medium, often air or a liquid coolant, where it is dissipated away from the device, thereby allowing regulation of the device's temperature at optimal levels. In computers, heat sinks are used to cool central processing units or graphics processors.

The MCHS cooling concept was first introduced by Tuckerman and Pease (1981). They made-up rectangular MCHS in a 1×1 cm^2 silicon wafer. The channels had a width of 50 μm and a depth of 302 μm and were separated by 50 μm thick walls. Using water as cooling fluid, the micro-channel heat sink was capable of dissipating 790 W/cm^2 with a maximum substrate temperature raise of 71℃ above the water inlet temperature. Due to their integral advantages, MCHS has received considerable attention since Tuckerman and Pease's innovative study. They suggested that as the hydraulic diameter of flow reduces, the coefficient of heat transfer increases. Thus, the higher coefficient of heat transfer can be achieved at the microscale level.

Several studies have been done which can be grouped as analytical (Keyes, 1984; Samalam, 1989; Knight *et al.*, 1991; 1992; Bejan and Morega, 1993; Lee and Vafai, 1999), numerical (Weisberg *et al.*, 1992; Fedorov and Viskanta, 2000), and experimental (Kishimito and Ohsaki, 1986; Sasaki and Kishimito, 1986; Nayak *et al.*, 1987; Harms *et al.*, 1999; Rahman, 2000). Among others, Knight *et al.* (1991, 1992) presented an optimization method that integrated both laminar and turbulent flow. Their results indicated that when the pressure drop (Δp) is small, laminar flow prevails, yielding low thermal resistance. Conversely, when the pressure drop is large, the optimal thermal resistance is found in the turbulent region. Weisberg *et al.* (1992) performed a two-dimensional numerical analysis for both hydraulically and thermally fully developed flow in the micro-channels.

Qu and Mudawar (2002a) numerically studied the three-dimensional heat transfer and fluid flow in a rectangular MCHS. Liu and Garimella (2004) also convinced that the behavior of liquid-flow through a microchannel is quite similar to the conventional channel and their observation showed that conventional correlations offer good predictions for the flow characteristics in micro-channels up to a Reynolds number (*Re*) of approximately 2000, in a hydraulic diameter range of 244-974 μm. Li *et al.* (2004) numerically investigated the effect of the geometric parameters of the MCHS and the thermo-physical fluid properties on the flow and heat transfer.

Some of the investigators have optimized the geometrical parameters of the MCHS. Steinke and Kandlikar (2006) reported the validity of the continuum theory and showed the entrance region effects, entrance and exit losses, and experimental uncertainties as for the main reasons for the discrepancy in the earlier data reported in the literature. They also presented a detailed analysis of the uncertainties associated with fluid flow and heat transfer data available in the literature. Lee and Garimella (2006) studied the effect of the entrance region on heat transfer under circumferentially uniform wall temperature condition and axially uniform wall heat flux thermal boundary conditions. Mishan *et al.* (2007) and Xu and Song (2008) confirmed that the conventional theory is applicable for water flow through microchannels including the entrance effects. They developed a method for the measurement of fluid temperature distribution, which provides the fluid temperature distribution inside the channel. Sabbah *et al.* (2008) observed that the estimation of heat transfer in micro-channels becomes difficult with the increase in complicacy of the geometry of the micro-channels, which requires three-dimensional analysis of heat transfer in both solid and liquid phases. Despite the small width of the channels, the conventional Navier-Stokes (N-S) and energy conservation equations still apply to the MCHS due to the continuum of the working fluid where the channel width is much larger than the mean free path of liquid(water) molecules. Rosa *et al.* (2009) reported that heat transfer in microchannels can be appropriately described by the standard theory and correlations. However, the scaling effects which includes entrance effects, conjugate heat transfer, viscous heating, electric double layer (EDL) effects, temperature dependent properties, surface roughness, rarefaction and compressibility effects (which are often negligible in macro-channels) may have a significant influence and these should be accounted.

In the past decades, commonly air and water were used as the working fluid due to their availability. Air has limited heat transfer capabilities however water needs a higher pumping power as well as causing a potential threat to the device damage at the smallest leakage. In 2012, Ammonia gas was used in MCHS because it significantly reduces the total thermal resistance (up to 34 %) compared to air-cool MCHS under similar operating conditions (Adham *et al.*, 2012). In addition, the pumping power requirement for Ammonia gas is much lower than air or water. With the latest development in the nanotechnology, presently, the researchers are strenuously investigating the utility of the combination of nanofluids and micro-channels which offers high rates of heat transfer from small surface areas. Nanofluids are diluting suspensions of nanoscale metallic particles in liquids and exhibit higher heat transfer performance because suspended nanoparticles have higher thermal conductivity than the base fluid (Ho *et al.*, 2010; Chein and Chuang, 2007; Anoop *et al.*, 2012). The objectives of this review paper are

a. To review the fluid flow (i.e.- pressure drop)

b. To review the heat transfer results of recent studies

c. To spotlight the key parameters affecting the heat transfer and

d. To present the research gaps for future research work

1.2. Literature review

Nanofluids are colloidal suspensions of nanoparticles (length scales 1-100 *nm*) in a base fluid. These particles can be metallic (Cu, Au) or metal oxides (Al_2O_3, TiO_2, ZrO_2), carbon (diamond, nanotubes), glass or another material, with the base fluid being a typical heat-transfer fluid, such as water, light oils, ethylene glycol (radiator fluid) or a refrigerant. Suspending particles in a base liquid to improve the thermal conductivity is not a new idea; earlier the setback for scientists was the particle size. The formation of microparticles was only possible due to manufacturing restrictions and these particles rapidly settled out of the fluid, and deposited in pipes or tanks, clogging flow passages, causing damage and erosion to pumps and valves, and increasing pressure drop. However, with the latest development in the nanotechnology, the nanoparticles are easily available in the present days with reasonable cost. Nanoparticles can be diffused in base fluids and stay on suspended in the fluid to a much greater extent than was earlier achieved with microparticles. This is mainly thought to be due to Brownian motion preventing gravity settling and agglomeration of particles, resulting in a much more stable, suspended fluid. Nanofluids (colloidal suspensions of nanoparticles in base fluid) acquire novel properties including the greater specific surface area, more stable colloidal suspension, lower pumping power for a specific heat transfer rate, reduced clogging compared to regular cooling colloids, and the ability to adjust the thermo-physical properties of suspensions by changing the nanoparticle materials and physical conditions (volume fraction of particles, particles size, and their shape). These novel characteristics make nanofluids suitable for several industrial applications such as pharmaceutical processes (drug delivery), surfactant and coating, cooling in heat exchangers, fuel cells, hybrid-powered engines, solar PV, and microelectromechanical systems (MEMS) (Minea, 2013).

The thermal conductivity of the metallic or nonmetallic solids such as Al, Ag, Cu, Si, Al_2O_3, and TiO is typically orders-of-magnitude higher than the base fluids even at low concentrations, result in significant increases in the heat transfer coefficient. The thermal conductivity of Cu is about 700 times higher than that of water and about 3000 times greater than that of engine oil, as shown in Table 1(Eastman *et al.*, 1996). The heat transfer enhancement mainly depends upon factors such as particle volume concentration, particle material, particle size, particle shape, base fluid material temperature, and additives.

Table 1. Thermal conductivities of various solids and liquids (Eastman et al., 1996)

Solids or liquids	Material	Thermal conductivity (*W/m·K*)
Metallic solids	Al	237
	Ag	429
	Cu	401
Non-metallic solids	Si	148
	Al_2O_3	40
Metallic liquids	Sodium @644 K	72.3
Non-Metallic liquids	Water	0.613
	Engine oil	0.145
	Ethylene glycol	0.253

Choi *et al.* (2001) formed carbon nanotube-in-oil suspensions and measured their effective thermal conductivity. The measured thermal conductivity was greater than theoretical calculations and was nonlinear with nanotube loadings. They suggested the physical concepts for comprehension the abnormal thermal behaviour of nanotube suspensions. The nanotubes offer the highest thermal conductivity enhancement as compared to the other nanostructured materials. Buongiorno (2006) proved that the single-phase model, as well as the dispersion models, cannot accurately follow the experimental observations. A two-component (solid and fluid) four-equations (continuity, momentum, energy, and nanoparticle flux) heterogeneous equilibrium model was proposed to illuminate the experimental findings. In the Buongiorno model, nanoparticle fluxes are considered in accordance with the two important slip mechanisms: Brownian diffusion (or Brownian motion) and thermophoresis (or thermophoretic diffusion). Computational investigation on the effect of nanoparticle volume fraction on fluid flow and heat transfer characteristics had been carried out by Soleimani *et al.* (2012). Even at low concentrations 0.02-0.12% of MgO nanoparticles in base fluid (water) for Re ranging from 11,000-49,000, the heat transfer coefficient augmentation from 6 to 12% was observed as compared to base fluid (Motevasel *et al.*, 2017).

Xie *et al.* (2002) observed the consequence of particle shape on the thermal conductivity enhancement in nanofluid and the results were compared with respect to the geometric shape of the particle with the same material and base fluid. The outcomes indicated that elongated particles provide better enhancement of the heat transfer. They used spherical and cylindrical particles of size 26 *nm* and 600 *nm* respectively of SiC in ethylene glycol base fluid and found that at 3% volume concentration, the thermal conductivity ratio of 1.16 and 1.10 was obtained for cylindrical and spherical particles respectively. The thermal conductivity of nanoparticles was dependent on their shape and the cylindrical particles have more thermal conductivity than that of spherical particles. The enhancement in convective heat transfer was higher for Ethylene Glycol/CNT-Ag than that of Ethylene Glycol/CNT (Elahmer *et al.*, 2017).

1.3. Useful definitions

In this review paper, some significantly terms used related to pressure drop and heat transfer enhancement. They are elaborated below-

***Nusselt Number* (*Nu*),** in heat transfer at a boundary (surface) within a fluid, is the ratio of convective to conductive heat transfer across (normal to) the boundary. In this context, convection includes both advection and diffusion. The *Nu* is defined as:

$$Nu = \frac{h \cdot d}{k}$$

Where h is the convective heat transfer coefficient, d is the characteristic dimension, and k is the thermal conductivity of the fluid.

Friction factor (f) is proposed by Fanning and Darcy. The Darcy friction factor is also known as the Darcy Weisbach friction factor or the Moody friction factor. It is a measurement of pumping power.

$$f = \frac{\Delta p}{(\rho u^2/2)(L/d_H)}$$

Where Δp is the pressure drop, ρ is the density of the fluid, d_H is the hydraulic diameter, u is the velocity of fluid and L is the length of the tube.

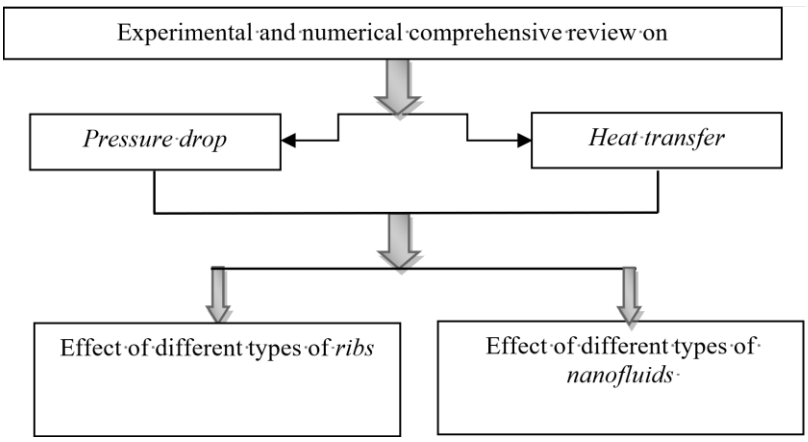

Figure 1. Flow chart of the present study

Thermal Performance factor is a function of heat transfer coefficient and friction factor. It is useful to analyze the performance of microchannel with ribs.

$$\eta = \frac{Nu/Nu_s}{(f/f_s)^{1/3}}$$

Where Nu and Nu_s are Nusselt numbers for a microchannel with and without ribs, respectively, f and $f_s$are friction factors for a microchannel with and without ribs, respectively. Figure 1 shows the flow chart of the present study.

2. Review on experimental and numerical studies of pressure drop

2.1. Effect of ribs inside the microchannel

Cui and Fu (2012) investigated a numerical study on pressure drop in microchannel flow with different bionic micro-grooved surfaces. They reported four types of bionic surfaces placoid-shaped, V-shaped, rib-let-shaped, and ridge-shaped grooved surfaces, are employed as the microchannel surfaces for the purpose of reducing pressure loss. The D_2Q_9Lattice Boltzmann method (LBM) was used to carry out simulations. The results showed that the micro-grooved surfaces possess the drag reduction performance. The existence of the vortices formed within the grooves not only decreases the shear force between fluid and wall but also minimize the contact area between fluid and walls, which can lead to a reduction of pressure loss. The drag reduction coefficient (η) for these four types of microstructures could be generalized as follows: $\eta_{\text{ridge-shaped}} > \eta_{\text{V-shaped}} > \eta_{\text{placoid-shaped}} > \eta_{\text{riblet-shaped}}$. In addition, they optimized the geometry of the ridge-shaped grooves, which acquired the highest drag reduction performance, were performed. The orthogonal simulations were conducted with five width to height ratios (*s/h* = 1, 2, 3, 4 and 5) and seven Reynolds number (*Re* = 5, 10, 20, 50, 100, 150 and 200). It was reported that, for the purpose of drag reduction, the ridge-shaped grooves with smaller width to height ratio are suggested for lower Reynolds number flow, larger width to height ratio is more appropriate for larger Reynolds number flow.

Heris *et al.* (2011) numericallyanalysed the laminar flow-forced convection of Al_2O_3-water nanofluid in a triangular duct. It was observed that the addition of nanoparticles to base fluid, besides the thermal conductivity increment, affects the structure of the flow field and leads to heat transfer enhancement, because of the scattering and haphazard movement of nanoparticles inside the fluid. It is noted that the decrement in the size of nanoparticles results in the increment in *Nu* at a particular concentration and increment in the concentration of the nanoparticles results in the increment in *Nu* at constant particle size. In the preliminary study they indicated that, in the case of triangular cross-sectional ducts in thermal systems, heat exchange rate can be compensated by the use of nanofluids because of the low-pressure drop. Therefore, the flow of the nanofluids throughout the triangular channel has both the benefits of low-pressure drop and high heat transfer rate.

Manay *et al.* (2012) analyzed the thermal performance of nanofluids in MCHS with the square duct. They used water-based nanofluids created with Al_2O_3 and CuO particles in four different volume fractions of 0%, 0.5%, 1%, 1.5% and 2% to analyze their special effects on heat transfer. It was suggested that the existence of the nanofluids gives better heat transfer characteristics than the traditional fluid i.e. pure water. The heat transfer increased with increasing Reynolds number as well as particle volume concentration. The CuO-water nanofluid gives better heat transfer than Al_2O_3-water nanofluid. For CuO-water nanofluid, the best heat transfer improvement was achieved at volume fractions of 2% and *Re*=100. At *Re* = 100 and volume fractions of 2%, the heat transfer improvement was 2.87 and 3.21for Al_2O_3-water and CuO-water nanofluids, respectively. For both nanofluids, the heat transfer enhancement was lowest at volume fractions of 0.5% and *Re*=1.000.

Croce and D'Agaro (2005) studied the effect of surface roughness on pressure drop and heat transfer rate in the plane and circular microchannels. The shapes of surface roughness were taken as squared obstructions and triangular obstructions. The observation gave some useful information on the impact of roughness. For both configurations, an increase in Poiseuille number was found with respect to the smooth wall cases. The prediction of the growth of friction losses was consistent with a large number of experimental data.

Quand Mudawar (2002b) experimentally and numerically investigated the pressure drop and heat transfer characteristics of a single-phase flow through a micro-channel heat sink. The heat sink was made-up of oxygen-free copper and built-in with a polycarbonate plastic cover plate. The heat sink consisted of an array of rectangular micro-channels which is 231 μm height and 713 μm long. The cooling liquid used was deionized water and two heat flux levels, q''_{eff}= 100 W/cm^2 and q''_{eff}= 200 *W/cm*^{2}, were tested. For q''_{eff}= 100 *W/cm*^{2}the value of *Re* ranged from 139 to 1672, and for q''_{eff}= 200 *W/cm*^{2}the value of *Re* ranged from 385 to 1289. They resulted that the good agreement between both the pressure drop and heat sink temperature information and consequently arithmetical predictions proves the conventional N-S and energy equations can precisely calculate the heat transfer characteristics of MCHS.

2.2. Effect of various types of Nanofluid flow

Sahin *et al.* (2015) experimental studied the turbulent flow convective heat transfer and the pressure drop characteristics and observed the heat transfer enhancement capability of the CuO-water nanofluid inside a circular tube. The CuO-water nanofluids with the loadings of 0.5%, 1%, 2%, and 4% were used. The nanoscaled particles in the base fluid increased heat transfer much more than pure water up to a limited value of particle volume fraction (0.5%). At *Re*> 10000, the Nusselt number got closer to that of the pure water by the increase of the particle concentration, and the CuO-water nanofluid exhibited better heat transfer performance, however, no heat transfer augmentation was observed at *Re*= 4000. The highest heat transfer augmentation was obtained in the case of *Re*=16000 and φ=0.005; the lowest one was achieved at *Re*= 20000 and φ=0.02.

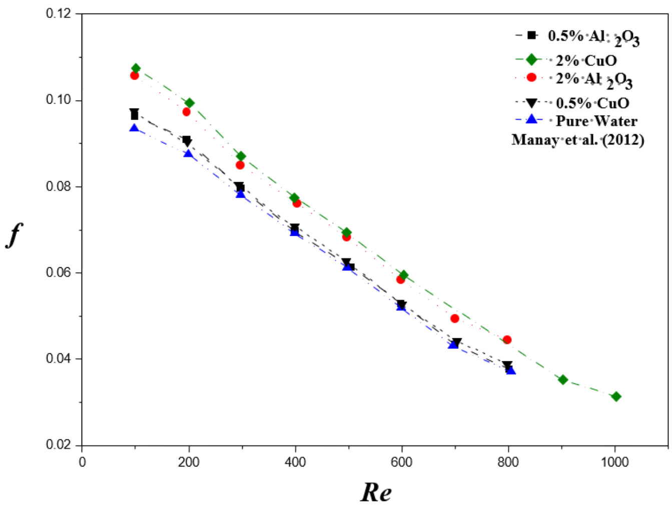

Figure 2. Effect of nanoparticle concentration on friction factor with different nanofluids

Figure 2 shows the variation in friction factor (i.e.-pressure drop) with nanoparticle concentration for different types of nanofluids flow inside the microchannel. The friction factor decreases due to increment in *Re* leads to a decrease in the pressure drop in MCHS. For the nanofluid (like- Al_2O_3 and CuO), as the particle concentration decreases the pressure drop also decreases. At the same concentration (0.5% or 2%), the value of friction factor is higher for CuO as compared to Al_2O_3 and the pressure drop is lowest in case of pure water.

3. Experimental and numerical study of heat transfer

3.1. Effect of ribs inside the microchannel

Chai *et al.* (2016) numerically studied the characteristics of laminar flow and heat transfer in MCHS with offset ribs on sidewalls. They considered the entrance effect, viscous heating, conjugate heat transfer and temperature-dependent properties in 3-D approach. Different types of offset ribs (semicircular, isosceles triangular, backward triangular, forward triangular and including rectangular) were used in the study. Depending on the different offset ribs and Reynolds number ($190 \leq Re \leq 838$), the newly proposed MCHS yield a Nusselt number of 1.42–1.95 and apparent friction factor of 1.93–4.57 times higher than the smooth one. Results explained that as Re < 350, the MCHS with forward triangular offset ribs yields the highest performance evaluation criteria while the one with rectangular offset ribs shows the lowest. As Re >

400, the one with semicircular offset ribs brings about the best performance evaluation criteria while the one with backward triangular offset ribs gives the worst.

Akbari *et al.* (2016) numerically investigated the effect of utilization of semi-attached rib on heat transfer and liquid turbulent flow of nanofluid water–CuO in the three-dimensional rectangular microchannel. The *Re* was ranging in between 10,000 and 60,000 and the volume fraction of CuO nanoparticle in 0%, 2%, and 4% was used. In this numerical simulation, the comprehensive effects of the changes in parameters such as volume fraction of the nanoparticle, dimensions of semi-attached rib, and Reynolds number were analyzed. It is observed that the utilizing semi-attached ribs in a microchannel with a ratio of $0 < \frac{R}{W} \leq 0.325$ in producing stronger vortices, that causes better mixing in fluid layers, was weaker than that with tooth shape of $R/W=0$. However, the major advantage of using a tooth with a ratio of $0 < \frac{R}{W} \leq 0.325$ in comparison to ordinary tooth was the increase in heat transfer and reduction in coefficient friction and pumping power.

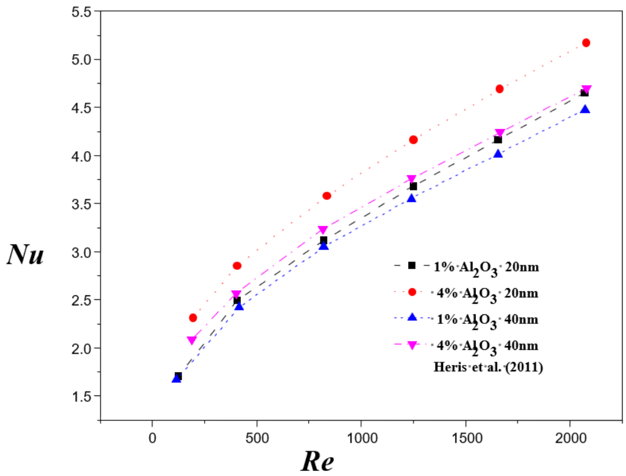

Figure 3. Effect of nanoparticle size and concentration on Nusselt number of Al_2O_3

Figure 3 shows the variation in *Nu* with *Re* at 1% and 4% volume concentration and 20nm and 40nm of nanoparticle size of Al_2O_3. For a particular particle size, *Nu* increases with an increase in the volume concentration (higher concentration of nanoparticle leads to enhance the heat transfer rate). However, for a particular volume concentration, the reduction in the particle size also results in augmentation of *Nu* and heat transfer.

Tabatabaeikia *et al.* (2014) presented a study on heat transfer enhancement by using different types of inserts like helical/twisted tapes, coiled wires, ribs/fins/baffles, and winglets. It was concluded that the Louvered strip insert has better function in backward flow compared to forward one. The increment in inclination angle from 10 to 30 can enhance the heat transfer rate by 5-11%. In the same situation, V-cut twisted tape gives better heat transfer by 10% compared to the plain twisted tube. The *Nu* provided by jagged insert can be up to 40% more than the perforated one. As the twist ratio increased, the heat transfer also increased by up to 12% in the helical screw and twisted tapes. Propeller-type vortex generators enhanced the heat transfer by 18% to 163% more than the plain tube.

3.2. Effect of various types of nanofluid flow

Ebrahimnia *et al.* (2016) numerically investigated the laminar convective heat transfer of TiO_2-water nanofluid flowing through a uniformly heated tube. It was reported that by using TiO_2-water nanofluids, the average heat transfer coefficient can be augmented by 21%.

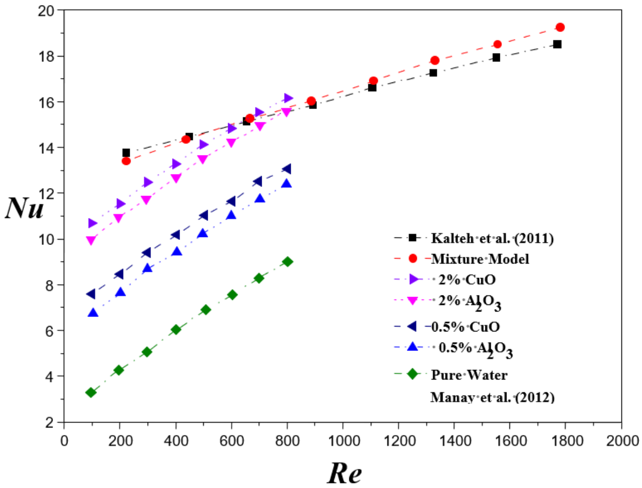

Figure 4. Effect of nanoparticle concentration on Nusselt number with various types of nanofluids

Figure 4 shows the comparison of *Nu* and *Re* with different types of nanofluids and their concentration with water. The increment in *Re* at a particular concentration, the *Nu* is more for CuO than Al_2O_3. Therefore, the heat transfer property is better in case of CuO. The mixture model used the mixture of both the nanofluids for analysis and resulted that it gives better results in heat transfer characteristics then their individual utilization in MCHS. However, the water gives the lowest heat transfer rate among the CuO, Al_2O_3, and mixture. Different researchers had found that the convective heat transfer coefficient is significantly enhanced by using different nanofluids at the different concentration, which aregiven in Table 2.

The comparative study on different nanofluids at various length to diameter ratio of channel is presented in Table 3. Table 3 also shows the different material of the channels used by different researcher. Senthilkumar *et al.* (2012) reported the consequence of inclination angle in Cu made heat pipe performance using copper nanofluid. An experiment was performed to analyze the improvement in the thermal efficiency of the heat pipe. The heat pipe width 600*mm* length and 20*mm* diameter were used. The average size of nanoparticle was 40 *nm* and concentration of Cu nanoparticle in the base fluid was 100 *mg/ltr*. The experiment was conducted for different heat inputs (30*W*, 40*W*, 50*W*, 60*W*and 70*W*) and the different inclination angle of pipe (0, 15°, 30°, 45°, 60°, 75° and 90°). Finally, it was reported that the thermal efficiency of copper nanofluid was higher than the base fluid and thermal resistance was also considerably less than DI water.

Table 2. Comparative study of various nanofluids at the different concentration on heat transfer enhancement

Researchers	Nano-particles	%Concentration	Size of particle *(nm)*	$\frac{L_{pipe}}{D_{pipe}}$	Boundary condition	Maximum % ageenhancement in *h*
Kim *et al.* (2009)	Al_2O_3	3 vol.	20–50	437	q''_w= constant	15 (in laminar Flow)
	Amorphouscarbonic	3.5 vol.	20	437	q''_w= constant	8 (in laminar Flow)
Asirvatham *et al.* (2011)	Ag	0.3-0.9 vol.	<100	683	T_w= constant	69.3 (at 0.9 vol%)
Chandrasekar and Suresh (2011)	Al_2O_3	0.1-0.2 vol.	43	247	q''_w= constant	17-34 (in the entry length under laminar flow)
						33-51 (thermally developed region of turbulent flow)
Kumaresan *et al.* (2013)	CNT	0.15-0.45 vol.	<100	233	-	92-150
Rayatzadeh *et al.* (2013)	TiO_2	≤0.25 vol.	30	652	q''_w= constant	-
Esmaeilzadeh *et al.* (2013)	Al_2O_3	0.5-1 vol.	15	143	q''_w= constant	19
Heyhat *et al.* (2013)	Al_2O_3	0.1-2 vol.	40	400	T_w= constant	32 (with 2 vol.)

Wang *et al.* (2013)	CNT	0.05 vol.	20-30	1000	q''_w= constant	70
		0.24 vol.	20-30	1000	q''_w= constant	190
Li and Xuan (2002)	Cu	0.3-2 vol.	<100	80	q''_w= constant	60
Wen and Ding (2004)	$\gamma-Al_2O_3$	0.6-1.6 vol.	27-56	215	q''_w= constant	30
Ding *et al.* (2006)	CNT	0.1-0.5 wt.	<100	215	q''_w= constant	350(with0.5 wt.% and at *Re*=800)
Heris *et al.* (2006)	Al_2O_3	0.2-3 vol.	20	167	T_w= constant	-
	CuO	0.2-3 vol.	50-60		T_w= constant	-
He *et al.* (2007)	TiO_2	0.2-1.2 vol.	95	483	q''_w= constant	12 (with 1.1 vol. % and at *Re* = 1500) 40 (with 1.1 vol. % and at *Re* = 5900)
Chen *et al.* (2008)	Titanate nanotubes	0.5-2.5 wt.	10	483	q''_w= constant	25 (with 2.5 wt.%)
Hwang *et al.* (2009)	Al_2O_3	0.01-0.3 vol.	30	1380	q''_w= constant	8 (with 0.3 vol.%)
Lai *et al.* (2009)	$\gamma-Al_2O_3$	0.5-1 vol.	20	490	q''_w= constant	55 (with 1 vol.%)
Anoop *et al.* (2009)	Al_2O_3	1-4 wt.	45,150	250	q''_w= constant	25% (for 45 nm particle based nanofluid (4 wt%) with *Re* = 1550 at *x/D* = 147) 11% (for the 150 nm particle based nanofluids (4 wt%) with *Re* = 1550 at *x/D* = 147)
Heris *et al.* (2009)	Cu	0.2-2.5 vol.	25	167	T_w= constant	-
Liao and Liu (2009)	CNT	0.5-2 wt.	10-20	217	q''_w= constant	49–56% (with 1 wt%)

Nandy *et al.* (2012) examined the thermal performance of screen mesh wick heat pipes with nanofluids. The copper made up heat pipe had a length of 200mm and the outer diameter of 8mm. The used nanofluids were such asTiO_2–water, Al_2O_3–ethylene glycol, ZnO–ethylene glycol Al_2O_3–water and TiO_2–ethylene glycol. The nanoparticles concentration was ranging from 1% to 5% volume of the base fluid. It was observed that the Al_2O_3–water nanofluid with 5vol% concentration gives better performance than only water. It was concluded that the thermal performance of heat pipe enhanced with nanofluids is much better than the conventional working fluids.

Table 3. Comparative study on different nanofluids at various length to diameter ratio of channel

Researchers	Nanofluid particles	$\frac{L_{pipe}}{D_{pipe}}$	Channelmaterial
Hung *et al.* (2013)	Al_2O_3	31.5	Cu
Senthilkumar *et al.* (2012)	Cu	30	Cu
Teng *et al.* (2010)	Al_2O_3	60	Cu
Nandy *et al.* (2012)	Al_2O_3, ZnO, TiO_2	25	Cu
Kang *et al.* (2006)	Ag	33.33	Cu
Shafahi *et al.* (2010)	Al_2O_3, CuO, TiO_2	4.45	Cu
Han and Rhi (2011)	Ag, Al_2O_3	50	Stainless steel
Mousa (2011)	Al_2O_3	40	Cu
Kole and Dey (2013)	Cu	30	Cu
Moraveji and Razvarz (2012)	Al_2O_3	23.75	Cu
Saleh *et al.* (2013)	ZnO	25	Cu

4. Overall heat transfer analysis of Al_2O_3 and CuO at various volume concentration of nanofluids

In heat transfer studies, it is important to evaluate the effects of both the heat transfer and the pressure drop. Therefore, the overall enhancement for both Al_2O_3 and CuO nanofluids is presented in the plot of thermal performance factor (η) versus *Re* to determine the overall gain. Figure 5 shows at the same Reynolds number and same volume concentration, the CuO provides better overall enhancement in heat transfer property then Al_2O_3 and the best overall heat transfer enhancement is found at 2% and *Re*= 100 for CuO-water nanofluid. The lowest overall heat transfer enhancement is found at 0.5% and *Re*=1000 for Al_2O_3.

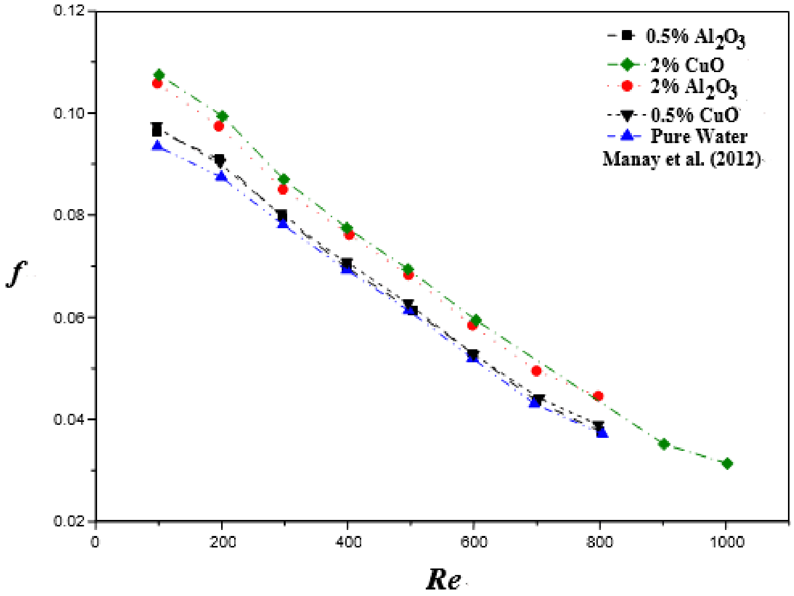

Figure 5. The thermal performance factor versus Reynolds number

5. Conclusions

This article reviews the results of heat transfer and pressure drop characteristic by using nanofluids and different types of ribs inside the microchannel. All the results show that the use of nanofluids increases the heat transfer with slight pressure drop in the microchannel. Some conclusions resulted from this review paper are:

i. The presence of ribs inside the microchannel enhances the heat transfer characteristics because of chaotic motion of the fluid particles in the flow field.

ii. The addition of nanoparticles in base fluid increases the heat transfer capacity and decreases the thermal resistance.

iii. The decrement in the size of nanoparticles enhances heat transfer at a particular concentration and the increment in the concentration of the nanoparticles results in the increment in heat transfer at constant particle size.

iv. The shape of the nanoparticles also plays a vital role to enhance the heat transfer characteristics.

The research gap for future research –

i. The physics of nanofluid flow and heat transfer through microchannels was found to be dissimilar from the conventional channels and its understanding is essential for thermal design, analysis, and safe operation of micro-devices.

However, the characteristics of nanofluid flow and heat transfer are still not fully understood. Therefore, the better understanding of the physics behind nanofluid flow through microchannel needs more systematic numerical studies.

ii. Most of the experimental investigations have been carried out with metal oxides. However, for the case of metallic nanoparticles, the augmentation in the heat transfer is much higher and this augmentation is much higher even at very small volume fraction. Therefore, the comprehensive investigations on nanofluids with metallic nanoparticles are essential to be performed.

iii. It is essential to develop the correlations of Nusselt number and friction factor for different nanofluids flow through different shaped microchannels.

iv. Further work is also needed for the understanding of the characteristics of heat transfer of hybrid nanofluids in the microchannel.

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Nomenclature

T	Temperature (K)
f	Friction factor
Δp	Pressure drop (*bar*)
D	Diameter of nanoparticle (*nm*)
q_w''	Wall heat flux (*W/cm²*)
T_w	Wall temperature (K)
T_{in}	Inlet temperature (K)
q_{eff}''	Effective heat flux (W/cm^2)
u	Velocity of fluid (*m/s*)
h	Convective heat transfer coefficient ($W/m^2 \cdot K$)
D	Characteristics dimension of the channel
L	Length of the tube (*cm*)
d_h	Hydraulic diameter (*mm*)
D_{pipe}	Diameter of pipe (*cm*)
L_{pipe}	Length of pipe (*cm*)

Greek symbols

η	Thermal performance factor
η_d	Drag reduction coefficient
φ	Volume concentration

ρ	Density (gm/cm^3)

Abbreviations

The chemical name of various nanofluid's molecular formulas

Al_2O_3	Aluminium oxide
TiO_2	Titanium dioxide
ZrO_2	Zirconium dioxide
CuO	Cupric oxide
SiO	Silicon monoxide
SiC	Silicon carbide
ZnO	Zinc oxide
MgO	Magnesium oxide
Cu	Copper
Au	Gold
Ag	Silver
C	Carbon

Other abbreviations

MCHS	Micro channel Heat Sink
RNG	Renormalized group
CCZ-HS	Cross cutting zigzag flow channel
CNT	Carbon Nanotube
DI	Deionised
Nf	Nanofluid particle