Numerical Study of Single Jet Impingement Cooling Using SiO_2 –Water Nanofluid

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Abstract
In this paper, numerical investigation on heat transfer enhancement on confined jet impingement using nanofluids as coolants were described. The governing equations with k- ε standard turbulent model solved by implementing ANSYS Fluent R14.5 code. The same geometrical details employed and the flowing jet of nanofluid impinges on a surface under a constant heat flux. The performance of jet impingement cooling with SiO_2-water nanofluid with different volume fractions of 1% - 5%, is analyzed and discussed. The nanoparticles diameter, another parameter taken into consideration, varied from 20 to 100 nm. The results indicate that the average Nu number and the required pumping power increase with increasing the Re number, volume fraction of nanoparticles and with decreasing in the diameter of nanoparticles. Moreover, the required pumping power increases as Re number and volume fraction increase, which is almost 4.2 times higher than the values calculated of base fluid. Furthermore, it is demonstrated that the application of the SiO_2-H_2O are recommended to achieve overall heat transfer enhancement and thermal performance when compared with the base fluid.

Keywords: Heat transfer enhancement, Jet impingement cooling, Nanofluids, Turbulent flow.

1. Introduction
Jet impingement cooling is a highly efficient technique in heat treatment, thermal management, and cooling of hot surfaces with relatively simple equipment. In this method solid surfaces are cooled by means of fluid jets. Thus, at a very high rate heat transfer can be acquired with a minimum flow rate. The jet impinging is used to obtain a high local heat transfer coefficient between the fluid and the surface, resulting in considerable increases in heat transfer as well as energy savings. This technology is widely used in many industrial applications such as cooling of electronic chips and microelectronic circuits, and nuclear power plants, [1]. Liquid jets mainly can be classified as; confined, semi confined and unconfined or free surface jet. In the confined jet impingement, the fluid can get recirculated and be entrained back into the impinging jet. In the unconfined jet the heated fluid is not return back into the jet which interacts with the ambient air. The semi-confined jets have characteristics of both confined and unconfined jets. The selection of liquid jet impingement type is depending on the industrial application. In addition, the flow of the jet may be laminar jet at (Re_{jet}<300), fully laminar jet (300<Re_{jet}<1000), transition jet (1000<Re_{jet}<3000) and fully turbulent (Re_{jet}>3000) based on nozzle diameter and velocity [2]. In order to obtain a heat transfer enhancement, different techniques have been employed, such as inserts and extended surfaces, but they required modification of the cooling system. Hence the use of nanofluids in a coolant is simpler in achieving heat transfer enhancement. The working fluid in liquid jet impingement system is usually water or traditional fluids, these fluids cannot meet the requirements of high heat flux removal

because of its low thermal conductivity. To meet the needs of heat transfer enhancement, nanofluids has been proposed.

A nanofluid is defined as a suspension of solid particles which have 1-100 nm size in a base fluid. Nanoparticles demonstrate remarkable features in the cooling system; among these are: high mobility, enhanced thermal conductivity, great stability, reduced erosion and pumping cost and high surface area, [3]. Numerous materials are utilized to make nanoparticles and disperse them in base liquids such as nitride ceramics (SiN), semi-conductors (SiO_2, TiO_2), carbide ceramics (SiC, TiC), oxide metals (CuO, Al_2O_3), metals (Al, Ag, Fe, Cu, Au), single double or multi wall carbon nanotubes (MWCNT, SWCNT, DWCNT) and composites. These nanoparticles have been dispersed and developed as additives to the conventional coolants such as water, poly-α-olefin oil and ethylene glycol, [4]. The Combining the liquid jet impingement and the nanofluid technologies is thought to capture the advantages of both and consequently enhances the heat transfer significantly.

There are many parameters that affects the heat transfer in the jet impingement cooling including the inlet pressure, flow velocity, nozzle diameter and working fluid and the heat flux on the impingement surface. A comprehensive review of the single-phase jet impingement cooling technique and its heat transfer methods has recently been conducted by Ekkad and Singh [5]. They reviewed a variety of modifications and applications of the jet impingement cooling technique focusing on impacting novel design, implementation, and improved manufacturing techniques for heat transfer enhancement. Using nanofluids as coolant was investigated experimentally and numerically, and it was found that the amount of heat transfer increase with the use of nanofluids compared with base fluids.

Many researchers have reported numerical and experimental studies on jet cooling heat transfer have been carried out using water or water based nanofluids. Umar et al. [6], Tie et al., [7], Kilic et. al. [8] and Darwish et. al. [9], carried out experimental investigations to study the heat transfer performance of nanofluids for confined and unconfined impinging jets. The experimental results obtained for both laminar and turbulent flow regimes, showed that the inclusion of nanoparticles into water has produced a considerable enhancement of the convective heat transfer coefficient. Darwish et. al. [9], Mohaghegh, et. al. [10], Etminan and Harun [11], Teamah et al. [12], and Lam et al . [13], numerically investigated the coffined nanofluid jet heat transfer. Pure water is used as base fluid with Al_2O_3 and CuO nanoparticles. The numerical results indicate that as the Reynolds number and the concentration of the particles in the fluid increases, the local heat transfer coefficient and the Nusselt number increase.

The main objective of the present research is to investigate the combined effect of heat transfer enhancement by jet impingement cooling with nanofluids. SiO_2 – water based was considered with various volume fractions (1% - 5%) and different size of spherical nanoparticles (dp = 20 nm, 50 nm, 80 nm and 100 nm) in a constant heat fluxed surface by means of ANSYSFLUENT software V14.5.

2. Physical and Mathematical Modelling

The physical domain considered in this study is shown in Figure (1), the domain regards the impinging jet on a heated wall with nanofluids to evaluate the thermal and fluid dynamic performances. The two-dimensional model has a length L of 480 mm while the height H is 36 mm and the jet orifice width W is 6.0 mm. A single-phase model was adopted because nanofluids can be considered as Newtonian fluids for low volume concentration fractions,[14]. The flow under consideration is governed by the incompressible, steady, two dimensional form of the continuity, the time-averaged incompressible Navier–Stokes equations and energy equation.

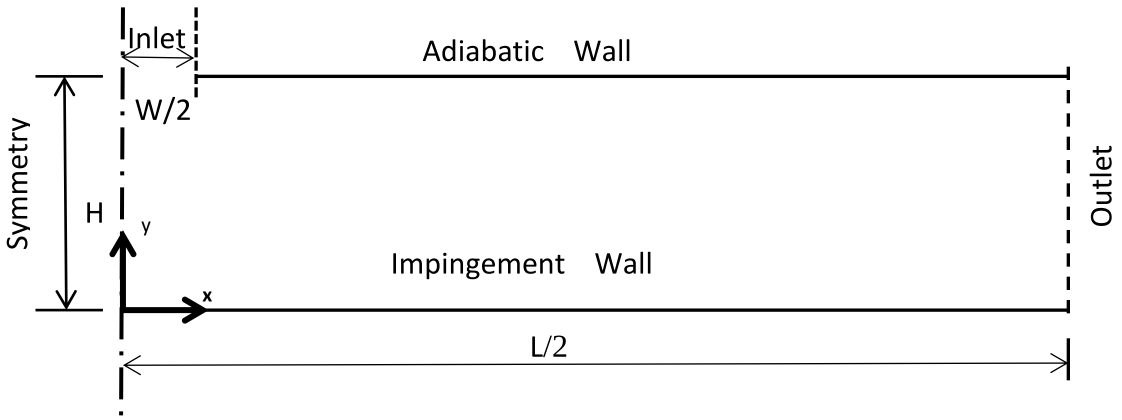

Fig. (1) The computational domain

In the Cartesian system these equations can be written as [15],

$$\frac{\partial}{\partial x_i}\ (\rho u_i) = 0 \tag{1}$$

$$\frac{\partial}{\partial x_j}\ (\rho u_j u_i) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right) \tag{2}$$

$$\frac{\partial}{\partial x_j}\ [u_i\ (\rho T)\ ] = \frac{\partial}{\partial x_j}\left[\ \left(\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x_i}\right] \tag{3}$$

The normal Reynolds stress which is combined by Boussinesq relationship and the eddy viscosity is given by

$$-\rho\overline{u_i' u_j'}\ = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \tag{4}$$

The turbulence standard κ–ε model is used as proposed by Bahmani et al, [16]:

$$\frac{\partial}{\partial x_i}[\rho k u_i] = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] - \rho\overline{u_i' u_j'}\frac{\partial u_j}{\partial x_i} - \rho\varepsilon \tag{5}$$

$$\frac{\partial}{\partial x_i}[\rho\varepsilon u_i] = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] - C_{1\varepsilon}(\varepsilon/k)\ \rho\overline{u_i' u_j'}\frac{\partial u_j}{\partial x_i} - C_{2\varepsilon}\ \rho(\varepsilon^2/k) \tag{6}$$

The turbulent viscosity which is expressed as:

$$\mu_t = \rho C_\mu\ k^2/\varepsilon \tag{7}$$

The empirical constants for the turbulent model were achieved by Launder and Spalding [17] : $C_\mu = 0.09$; $\sigma_k=1.0$; $\sigma_\varepsilon=1.3$; $C_{1\varepsilon}=1.47$; $C_{2\varepsilon}=1.92$; and $Pr_t = 0.85$.

In order to present the results of numerical solution, the Reynolds, the average Nusselt numbers, pressure loss and the required pumping power are defined as [18] :

$$Re = \frac{u_j W \rho_{nf}}{\mu_{nf}} \tag{8}$$

$$Nu_{ave} = \frac{q W}{(T_s - T_j)\, k_f} \tag{9}$$

$$\Delta P = f \frac{L}{W} \frac{\rho_{nf}\, u_j^2}{2} \tag{10}$$

$$PP = \dot{V} \Delta P \tag{11}$$

Determining Nanofluids thermos-physical properties play important role in modeling the forced convective with nanofluids and accuracy of the results. For density and specific heat of nanofluids are taken as the average of fluid and solid particle densities based on the volume fraction φ of the particles in the suspension, [19] :

$$\rho_{nf} = (1-\varphi)\rho_f + \varphi \rho_p \tag{12}$$

$$\left(\rho C_p\right)_{nf} = (1-\varphi)\left(\rho C_p\right)_f + \varphi\left(\rho C_p\right)_p \tag{13}$$

The nanofluids effective viscosity and thermal conductivity the correlation for oxide metals nanoparticles suspended in water proposed by Corcione [20], which includes the influences of Brownian motion, viscous sublayer thickness, and temperature :

$$\frac{\mu_{nf}}{\mu_f} = \frac{1}{\left(1 - 34.87 \varphi^{1.03} \left(d_p/d_f\right)^{-0.3}\right)} \tag{14}$$

Where :

d_f is the equivalent diameter of the base fluid molecule calculated by :

$$d_f = 0.1 \left(\frac{6M}{N \pi \rho_{f0}}\right)^{1/3} \tag{15}$$

Where :

M is the molecular weight of base fluid, N is the Avogadro number = 6.022 × 10^{23} mol^{-1}, ρ_{f0} is the mass density of the based fluid calculated at temperature T_0 = 293 K, and water molecule diameter is d_f = 0.2 nm.

The thermal conductivity ratio is proposed by Corcione [20]:

$$\frac{\gamma_{nf}}{\gamma_f} = 1 + 4.4\, \varphi^{0.66} \left(\frac{\gamma_p}{\gamma_f}\right)^{0.03} \left(\frac{T}{T_{fr}}\right)^{10} Pr_f^{0.66}\ Re_B^{0.4} \tag{16}$$

Where the Brownian Reynolds number defined by

$$Re_B = \frac{2 \rho_f B\, T}{\pi\ \mu_f^2\, d_p} \tag{17}$$

where, B is the Boltzmann constant (1.3807 × 10^{-23} J/K), and T_{fr} is the freezing point temperature of the base fluid.

According to Figure (1), all geometrical details and boundary conditions are symmetrical, and only half of the computational domain is solved. The boundary conditions imposed on the computational domain can be summarized as follows;

a) At inlet, uniform velocity jet inlet and temperature of 298K, different inlet uniform velocities, corresponding to Reynolds numbers ranging from 5000 to 30000, furthermore, the inlet turbulence intensity value is set to 5%.
b) At outlet, an atmospheric pressure ($P = P_{atm}$) condition applied because the ambient pressure is prevailed there.
c) At the Impingement wall, no slip conditions and uniform heat flux (6000W/m^2).
d) Adiabatic conditions are applied to the confinement walls.
e) Symmetry conditions at the symmetry plane.

3. Solution Methodology and Validation

A commercial CFD solver, ANSYS FLUENT 14.5 was used in modeling the forced convection flow and heat transfer of jet impingement cooling with nanaofluids. The governing equations for mass, momentum, energy, and the standard turbulent $k - \varepsilon$ model are discretized within the computational domain by the finite volume approach. The discretization of the convective and diffusive terms were executed with a second-order upwind scheme, while the velocity and pressure fields were coupled using SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm.
The resulting set of algebraic equations solved by imposing the boundary conditions, and the enhanced wall treatment functions are activated to increase the model accuracy in the near-wall region. The convergence criteria of 10^{-6} for the residuals for all equations. Sample of the grid used in the simulations is illustrated in Figure (2). It consists of a two-dimensional mesh. In order to capture the gradients accurately, grids are finer near the hot surface and jet regions while coarser at the core region and adiabatic walls.

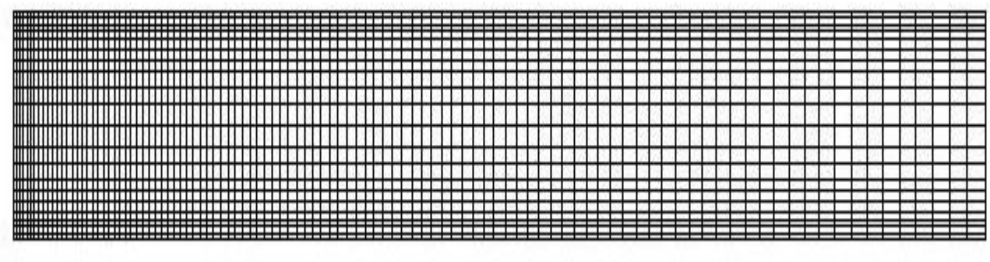

Fig. (2) Computational mesh

Grid independency test is performed and shown in Figure (3). Five different grid distributions were tested on the model with H/W = 6 at Re = 10000, with pure water as working fluid, by monitoring the average Nusselt number which is taken as a criterion for grid independence to ensure that the results are grid independent. Comparing the fourth and fifth-mesh configurations gives less than 1 % variation of average Nusselt number, as a result, the fourth grid case has been adopted because it ensured a good compatibility between the machine computational time and the accuracy requirements.

Figure (4) shows a comparison of the current study code results with the results obtained by Manca et al. [21]. The comparisons in terms of average Nu values for the case *H/W* =

6, Reynolds number range of 5000 – 20000, and constant wall temperature condition at the hot surface. The results of the water-Al_2O_3 as a working nanofluids with volume fraction of 0 % and 4 % were used for comparisons. It is observed that the numerical results, obtained in this work, in good agreement with the numerical ones given by Manca. As a result of this comparison, the deviation between the results is found to be between 9% and 16 %.

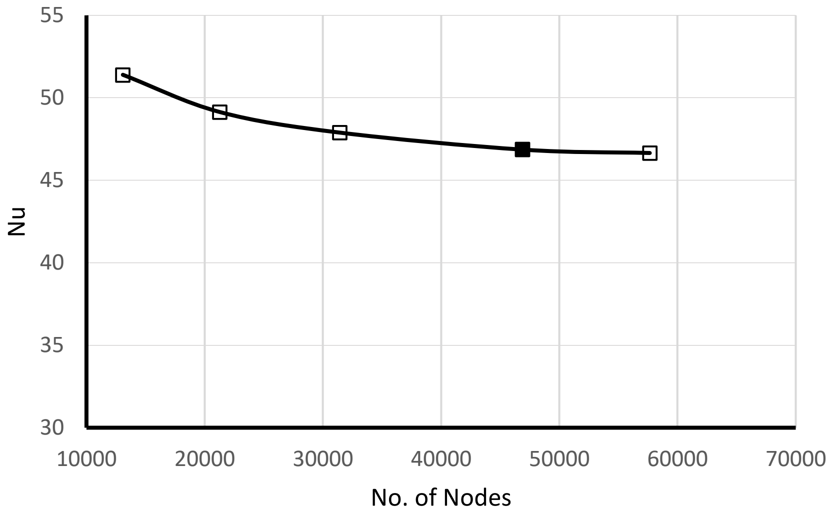

Fig. (3) Grid independence testing, H/W = 6, R = 5000 with base fluid.

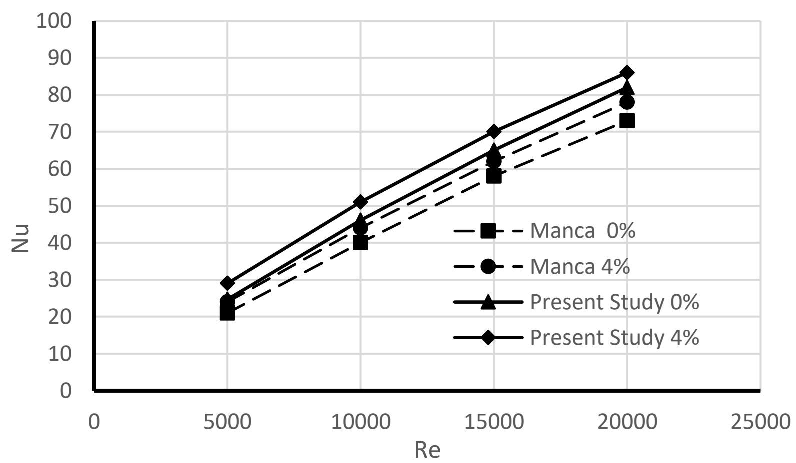

Fig. (4) Validation comparison between this study and Manca et al. [21].

4. Results and Discussion

In this section, numerical results were analyzed and discussed, the effect of SiO_2 water nanofluid with 20 nm sized particles at different volume fractions (φ=1%, 2%, 3%, 4% and

5%) on heat transfer enhancement was investigated also the effect of different nanoparticles diameter (dp = 20nm – 50nm 80 and 100 nm) with constant volume fraction of (φ=3%) studied.

Figure (5) illustrates the effects of nanoparticle volume fraction on Nusselt number for the SiO_2 – water nanofluid. It demonstrates that adding a low volume fraction of nanoparticle (0.01-0.05) with particle diameter of 20 nm to the base fluid leads to significant increase in Nusselt number. Therefore, the use of nanoparticles with a larger volume fraction and a higher Reynolds number can improve the heat transfer characteristics. At all cases, an increasing the Reynolds number, the Nusselt number increases, but the Nusselt number is not very sensitive to the volume fraction of nanoparticles at lower Reynolds number. This can be explained as at higher Reynolds number, as the volume fraction increases, irregular and random movements of the particles increase the energy exchange rates in the fluid enhancing the thermal dispersion of the flow and heat transfer. As indicated in Figure (6), the results show that the pressure loss of the SiO_2– water increases with the increase of Reynolds number and nanoparticle concentration. With an increase in the particle volumetric concentration in the nanofluids, the density and viscosity increase and hence they cause an increased pressure loss.

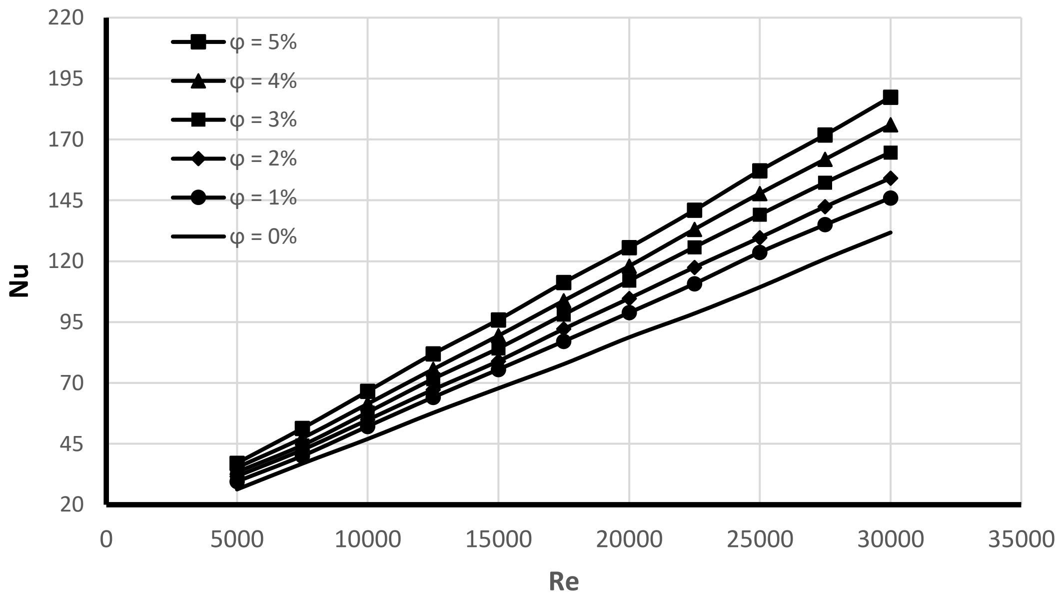

Fig. (5) Heat transfer of SiO2/water with different volume fractions.

The required power has a square dependence on *Re*, but the pumping power ratio, referred to the base fluid values PPnf/PPbf is independent on Re. Figure (7), showed that the ratio increases as concentration increases for SiO_2 water nanofluid . At the same Re = 20000, the required pumping power is 1.38, 1.85, 2.35, 3.05 and 4.18 times greater than the values calculated in case of pure water.

To study the effect of nanoparticles diameter in heat transfer enhancement of jet cooling with nanofluids, SiO_2 - water nanofluid taken as a working fluid, with volume fraction of

3%. The Reynolds number was in the range of 5000–30,000. The range of nanoparticle diameter is 20, 50, 80 and 100 nm.

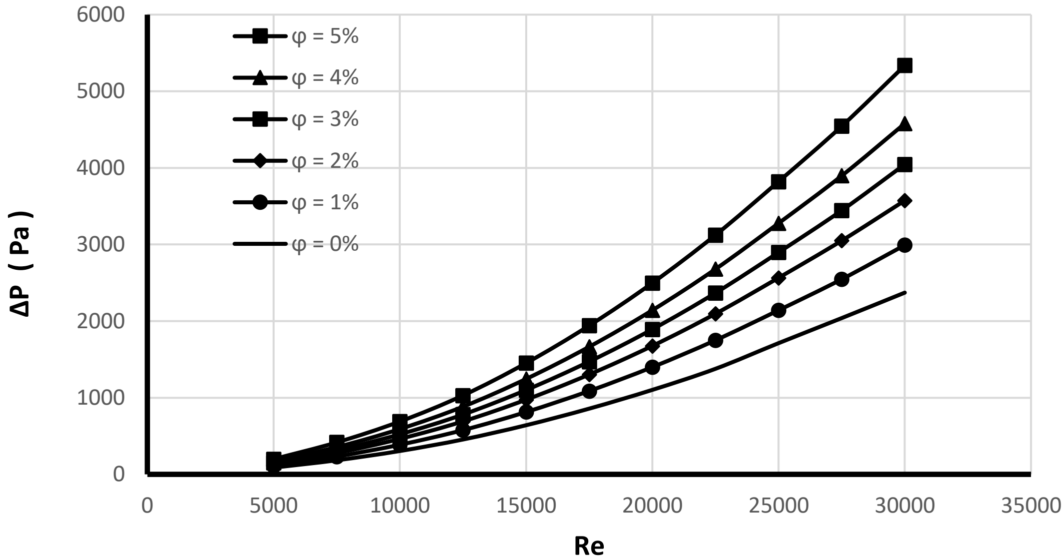

Fig. (6) Pressure loss of SiO_2/water with different volume fractions.

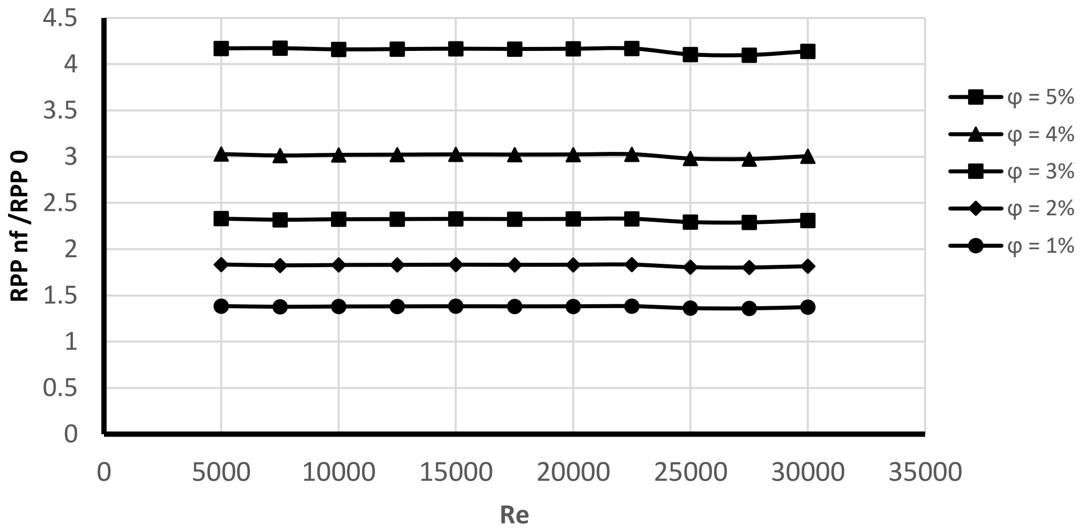

Fig. (7) Required pumping power ratio of SiO_2/water with different volume fractions.

As illustrated in Figure (8), the results revealed that the nanofluid with smaller particle diameter has the higher Nusselt number. As presented in the figure, the nanofluid with 20 nm nanoparticle diameter has the highest Nu, whereas, the nanoparticle with a diameter of 100 nm has the lowest one. In the case of Re = 25000, the decrease of the nanoparticle diameter from 100 to 20 nm leads to an increase of 24.6% of Nusselt number. This can be explained as the use of nanofluids with a smaller nanoparticle diameter increases the

Brownian motion velocity of the nanoparticles and consequently enhances the dispersion thermal conductivity, which results in better heat transfer. The other reason can be attributed to the increase in the surface area per unit volume as the particle size reduced, and thus the effectiveness of nanoparticles in transferring heat increases. Also, it can be observed that the effect of nanoparticles diameter on Nu is more significant for higher Reynolds numbers. Hence, in higher Reynolds numbers, reducing nanoparticle diameter has a higher impact on heat transfer coefficient improvement of nanofluid.

Figure (9) displayed the impact of particle diameter for nanofluid with silicon dioxide nanoparticles on the required pumping power at ϕ= 3% and different Reynolds number. As presented in the figure as the diameter of nanoparticle decreases, the required pumping power increase, due to the increase in pressure loss. This is due to the increase in viscosity, as the particle diameter decreases for the same volume fraction.

5. Conclusions

A numerical analysis of a turbulent, two-dimensional model on a confined impinging jet with nanofluids has been studied by implementing the ANSYS Fluent code. The bottom impinged wall is heated with uniform heat flux, and different jet velocities were considered, over Re in the range of 5,000 - 30,000. Regarding the nanofluid, SiO_2-water tested with different volume fractions in range of 0-5%. Also, the effect of nanoparticle diameter with range of 20 - 100 nm was investigated.

The major conclusion of the study is that the utilization of nanofluid in jet impingement cooling can be offered as an appropriate enhancement technique to get best thermal performance. Other conclusions that can be drawn from the study are:

i. The average Nusselt number, pressure loss, and the required pumping power increase with increasing in the Reynolds number.
ii. The average Nusselt number and the required pumping power increase with increasing in the volume fraction.
iii. The average Nusselt number and required pumping power increase with decreasing in the diameter of nanoparticles.

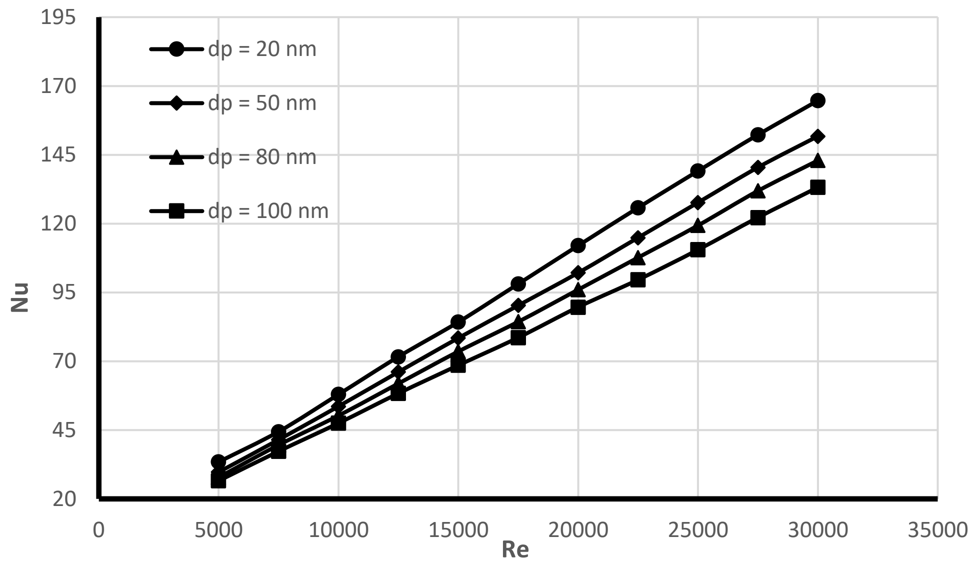

Fig. (8) Heat transfer of SiO_2/water with different nanoparticles diameters.

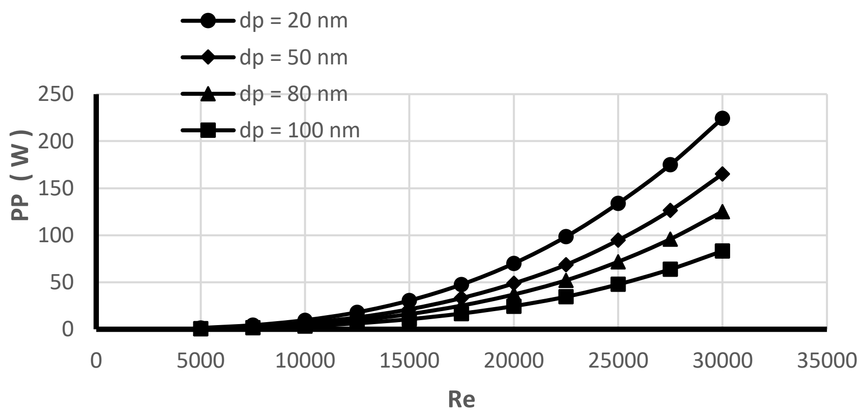

Fig. (9) The required pumping power of SiO2-water with nanoparticles diameters.

Nomenclature

A	Area, m^2.	$\dot{V}$	Volume flow rate, m^3/s.
B	Boltzmann constant, J/K.	W	Jet width, m.
Cp	Specific heat, J/kg K.	x	Spatial coordinate.
d	Diameter, m.	***Greek symbols***	
f	Friction factor.	φ	Nanoparticle concentration, %.
h	Heat transfer coefficient, W/m^2 K.	ε	Dissipated turbulent kinetic energy, m^2/s^3
H	Test section height, m.	γ	Thermal conductivity, W/m. K
k	Turbulent kinetic energy, m/s^2.	μ	Viscosity, kg/ms
L	Channel length, m,	ρ	Density, kg/m^3

Nu	Nusslet number,	ΔP	pressure loss, Pa.
p	Pressure, N/m^2	***Subscripts***	
PP	Pumping power, W.	ave	Average
Pr	Prandtl number	f	Fluid
Re	Reynolds number	in	Inlet
q	Heat flux, W/m^2.	j	Jet
T	Temperature, K	p	Solid particle

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