

Heat Transfer and Pumping Power Using Nanofluid in a Corrugated Tube

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Abstract

Heat transfer enhancement using nanofluid through a plain pipe or corrugated plate has been investigated by many researchers during the last few years. Despite of this improvement, it has not been studied with the justification of using aluminum oxide nanofluid in a corrugated channel in terms of the increased power needed for the flow due to the change in thermophysical properties of nanofluid compared to that of the basefluid. In the present study, the use of nanofluid in a V-shaped corrugated tube for a turbulent flow in order to enhance the heat transfer rate and corresponding pumping power required has been analyzed for Reynolds number- 4000 to 20000. For a constant heat transfer coefficient of 10000 W/m²K, the minimum pumping power is obtained for 3% volume fraction of nanofluid flowing through corrugated pipe which is 40% lower than that of water.

Introduction

Numerous researches have been performed in the last few decades to discover a more suitable heat transfer fluid to design compact heat exchangers with excellent thermal performance for industrial and engineering applications. Nanotechnology is an innovative technique to improve the heat transfer characteristics of working fluid by the uniform and stable suspension of solid nanoparticles in conventional base fluid. By the addition of nanoparticles with preferred volume concentration, the thermo-physical properties of working fluid are substantially improved which potentially lead to enhance the thermal and hydrodynamic behavior of nanofluid [1-3]. The implementation of corrugation in the geometry and suspension of nanoparticles in base fluid are becoming an innovative and inexpensive ways for increasing the heat transfer rate in many aspects of heat transfer applications.

Heidary and Kermani [4] investigated the heat transfer improvement using Cu–water nanofluid through a sinusoidal wavy channel under laminar flow condition. Nusselt number and skin friction coefficient were studied for a range of Reynolds number from 5 to 1500, and volume fraction up to 20%. Ahmed et al. [5] carried out numerical investigation of laminar heat transfer and pressure drop characteristics of Cu–water nanofluid through an isothermally heated triangular shaped corrugated channel and observed the enhancement of average Nusselt number with an increase of Reynolds number and volume fraction. Yousoff et al. [6] performed the effects of volume fraction, wave amplitude and Reynolds number on the improvement of friction coefficient and Nusselt number through a uniformly heated sinusoidal wavy channel using nanofluid. Adnan et al. [7] performed experiment to observe the significant improvement of heat transfer using TiO₂-water nanofluid in a car radiator for Reynolds number 4,000-16,000 and volume fraction 1% to 2.5%. Three different geometries (Circular, flat and elliptical tube) were employed to study the friction factor and heat transfer characteristics and flat tube showed highest convective heat transfer coefficient. Enhancement of Nusselt number with an increase of Reynolds number, particle volume fraction and wave amplitude were also studied numerically

through a trapezoidal-corrugated channel for laminar forced convection [8]. Ranjbar et al. [9] investigated the turbulent convective heat transfer augmentation in a wavy channel under constant wall heat flux using Al₂O₃-water nanofluid and two phase model was considered. Ashkan and Muhammad [10] studied the thermal and hydraulic behavior of flow in a horizontal periodic rib-grooved channel like triangular, square and arc shaped for turbulent forced convection utilizing Al₂O₃, SiO₂, CuO, ZnO nanoparticles dispersed in water and reported a considerable amount of heat transfer enhancement with negligible increase in friction factor.

From the literature study it may be mentioned that most of the works focused on the heat transfer enhancement utilizing nanofluid in a corrugated channel. To accomplish heat transfer enhancement, nanofluid requires increased pumping power and the justification of using nanofluid in terms of pumping power is not studied in most of the research articles. So the scope of our present work is to study turbulent forced convection heat transfer and pumping power through a V-shaped corrugated tube subjected to uniform heat flux using Al₂O₃-water nanofluid. The study is performed for a wide range of Reynolds number 4,000-20,000 for particles volume fraction 1% to 5%. The optimum volume fraction for which nanofluid provides lesser amount of pumping power compared to base fluid is analyzed along with the improvement of heat transfer coefficients with an increase of Reynolds number and volume fraction.

Nomenclature

A_c	Cross-sectional area of tube
A_w	surface area of tube
C_p	Specific heat at constant pressure
D	Diameter of the tube
h_c	Average heat transfer coefficient
k	Thermal conductivity
ΔP	Differential pressure
Q	Heat transfer
q	Heat flux
T	Temperature
U	Velocity of flow at inlet
W	Pumping power

Greek symbols

ϕ	Volume fraction
μ	Dynamic viscosity
ν	Kinematic viscosity
ρ	Mass density

Subscripts

av	Average value
i	Value at inlet
o	Value at outlet
nf	Nanofluid
bf	Basefluid
p	Nanoparticle
w	Value at wall

max Maximum value
min Minimum value

Dimensionless parameter

Nu Nusselt number
Re Reynolds number
Pr Prandl number

Governing Equations

The Reynolds number for the flow of nanofluid is expressed as

$$Re = \frac{\rho_{nf} U_{av} D}{\mu_{nf}}$$

The rate of heat transfer Q_{nf} to tube wall is assumed to be totally dissipated to nanofluid flowing through a corrugated tube, raising its temperature from inlet fluid bulk temperature T_{bi} to exit fluid bulk temperature T_{bo} . Thus,

$$Q_{nf} = \dot{m}_{nf} C_{P,nf} (T_{bo} - T_{bi})_{nf} \quad (1)$$

Where \dot{m}_{nf} is the mass flow rate of nanofluid, $C_{P,nf}$ is the specific heat of nanofluid at constant pressure. The definition of bulk temperature T_b is given by

$$T_b = \frac{\int_0^{A_c} u T dA_c}{\int_0^{A_c} u dA_c} \quad (2)$$

The average heat transfer coefficient, h_c is given by

$$h_c = \frac{Q_{nf}}{A_w (\Delta T_m)}$$

Where A_w is the surface area of circular tube and ΔT_m is the difference between the wall temperature and the average bulk temperature of the fluid

The average wall temperature T_w is computed by

$$T_w = \frac{1}{\sigma} \int_0^{\sigma} T_{w,x} dx$$

So the expression of average Nusselt number is defined as follows

$$Nu = \frac{h_c D}{k_{nf}}$$

The pumping power per unit length in turbulent flow is given by

$$\dot{W} = \frac{\pi D h^2 U_{av} \Delta P}{L} \quad (3)$$

Where ΔP is pressure difference

$$\Delta P = \frac{f L \rho U_{av}^2}{2D} \quad (4)$$

Thermo-physical Properties of Nanofluid

The density and specific heat of Al_2O_3 -water nanofluid is found from equation (6) and equation (7) [11]

$$\text{Density, } \rho_{nf} = \rho_p \phi + \rho_{bf} (1 - \phi) \quad (6)$$

$$\text{Specific heat, } C_{nf} = (1 - \phi) C_{bf} + \phi C_p \quad (7)$$

The thermal conductivity of the nanofluid is calculated using the correlation proposed by Koo and Kleinstreuer [12]. In equation (8), to calculate the effective thermal conductivity, the diameter of the solid nanoparticles, volume fraction and temperature of the nanofluid, properties of the base fluid and nanoparticles and the Brownian motion of the solid particles within the base fluid have been considered.

Thermal conductivity,

$$k_{nf} = \frac{k_p + 2k_{bf} + 2(k_p - k_{bf})\phi}{k_p + 2k_{bf} - (k_p - k_{bf})\phi} k_{bf} + 5 \times 10^5 \beta \phi \rho_p C_p \sqrt{\frac{k_{bf} T}{\rho_p D}} f(T, \phi) \quad (8)$$

Where, $f(T, \phi) = (-6.04\phi + 0.4705)T + (1722.3\phi - 134.63)$

The dynamic viscosity of nanofluid is calculated by using the following empirical correlation developed by Corcione [13] with a 1.84% of standard deviation.

$$\frac{\mu_{nf}}{\mu_{bf}} = \frac{1}{1 - 34.87 \left(\frac{d_p}{d_{bf}}\right)^{-0.3} \phi^{1.03}} \quad (9)$$

Where d_{bf} is the equivalent diameter of the base fluid molecule, and is given by

$$d_{bf} = \left[\frac{6M}{N\pi\rho_{fo}} \right]^{1/3} \quad (10)$$

Where M is the molecular weight of the base fluid, N is the Avogadro number, and ρ_{fo} is the mass density of the base fluid calculated at temperature $T_o = 293$ K. In the present study the nanoparticles are assumed to have a diameter of 50 nm in order to calculate the effective thermal conductivity and viscosity.

Physical Model and Boundary Condition

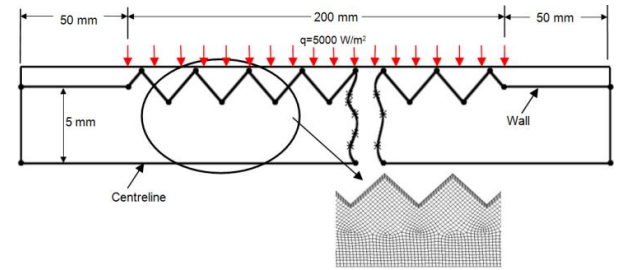


Figure 1. A 2D model of the circular corrugated pipe and the corresponding mesh showing inflation at the wall (not to scale)

A numerical investigation of turbulent flow through a circular, plain and a V-shaped corrugated pipe with constant heat flux of 5000 W/m² applied on its surface has been considered in this study. The diameter of both of the pipes is taken as 10 mm. In case of the circular pipe, the calculation has been done at a fully developed section, to ensure which, 1 m long pipe is taken where the heat transfer coefficient was measured at a distance of 800 mm from the inlet. The pressure has been measured at two sections- 800 mm and 900 mm from the inlet respectively to measure the pressure drop. A 2D simulation with axisymmetric model has been done. For both of the pipes, fluid is allowed to flow with a uniform velocity and uniform temperature of 300 K at the inlet of the tube with an assumption of no slip condition. In case of the corrugated tube, which is shown in figure 1, the pressure and bulk temperature of the fluid are measured at the beginning and end of the corrugated section. The corrugation has an amplitude of 1 mm and wavelength of 4 mm. Realizable k-ε turbulent model is chosen with enhanced wall treatment for the single phase analysis. Turbulent intensity is taken as 5% at the inlet and at the outlet boundary "pressure outlet" is considered. Heat transfer parameters like convective heat transfer coefficient and average Nusselt number and fluid dynamic parameters like pressure drop, friction factor and pumping power are calculated. With an increase of Reynolds number and particle volume fraction heat transfer characteristics, fluid flow behaviour and pumping power are investigated. By the addition of nanoparticles with a low volume concentration in base fluid, the enhancement of heat transfer, the increase of frictional losses, and the reduction of pumping power in comparison with water are studied under the prescribed flow condition.

Code Validation and Grid Independence

To study the heat transfer behaviour and pumping power requirement numerically for the flow of fluid through the circular plain and corrugated pipe, commercial computational fluid dynamics software – ANSYS Fluent [15] has been employed in the present work. Water is considered as the working substance and Nusselt number measured by numerical method at the fully developed region is compared with that obtained using the correlation proposed by Notter and Sleicher [14].

The correlation developed by Notter and Sleicher is as follows:

$$Nu = 5 + 0.016 Re^a Pr^b \quad (11)$$

Where, $a = 0.88 - \frac{0.24}{4+Pr}$ and $b = 0.33 + 0.5e^{(-0.6Pr)}$

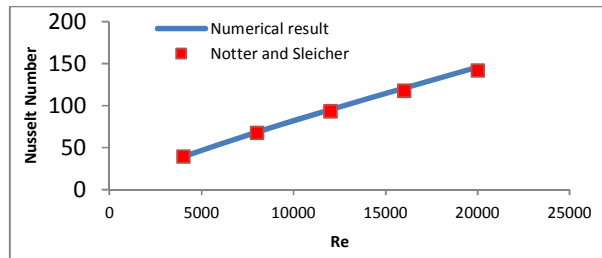


Figure 2. Comparison of Nusselt number between the numerical results from present work and Notter and Sleicher correlation [14]

From figure 2 it is obvious that the numerical results of the present study are in good agreement with the Nusselt number obtained using Notter and Sleicher correlation.

Taking water as the working substance, for the circular plain pipe, an optimum grid size of 600×40 has been used with a bias factor of 10 towards the inlet and outlet as well as the wall of the pipe. Beyond this size of the grid, there is not any change in the velocity profile. For the circular corrugated pipe, the velocity profile at $x=0.15$ m has been found for the grid sizes – 9600, 14200, 22300, 31200, 46000, 61600 and it has been observed that beyond a grid size of 46000, there is not much change in the velocity profile. Therefore for the present study, a grid size of 46000 has been taken for the fluid domain.

Result and Discussion

Effect of Volume Fraction of Nanofluid on the Heat Transfer Through Circular Plain Tube

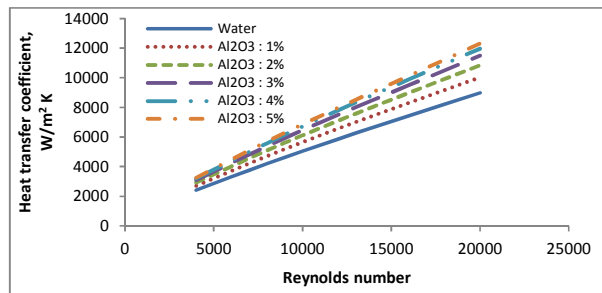


Figure 3. Heat transfer enhancement using nanofluid through circular plain pipe

From figure 3 it is seen that the addition of nanoparticle in the base fluid water has an effect of increasing the heat transfer rate which is higher at a greater volume fraction of the nanofluid. At a Reynolds number of 12000, the heat transfer coefficient obtained using Al_2O_3 -water nanofluid of 5% volume fraction is 36.3% higher than that obtained from water. This trend is obtained in case of corrugated pipe also.

Effect of Using Corrugated Pipe on the Heat Transfer Rate for Water:

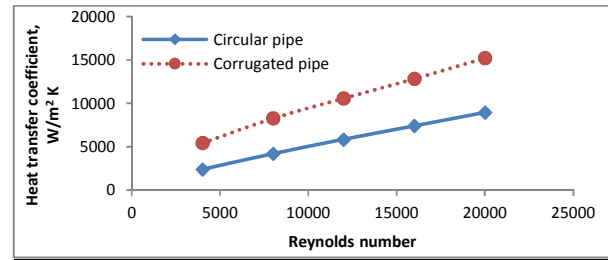


Figure 4. Heat transfer enhancement using corrugated tube

The use of corrugated pipe contributes in the enhancement of the heat transfer coefficient comparing to the circular plain pipe. From figure 4, at a Reynolds number of 12000, the corrugated pipe enhances the heat transfer rate by 81%.

Effect of Using Corrugated Pipe on the Pumping Power for Water

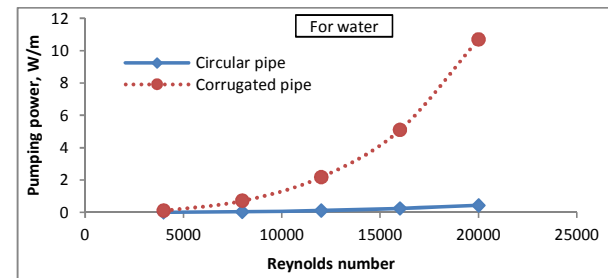


Figure 5. Comparison between the pumping power obtained using plain and corrugated pipe for water as the flowing fluid

From figure 5, the required pumping power for corrugated pipe is higher than that of plain pipe. With the increase in the Reynolds number, the pumping power requirement as well as the difference between the two increases. Similar trend is seen for the nanofluid of various volume fractions.

Effect of using corrugation on the pumping power requirement for the same heat transfer coefficient

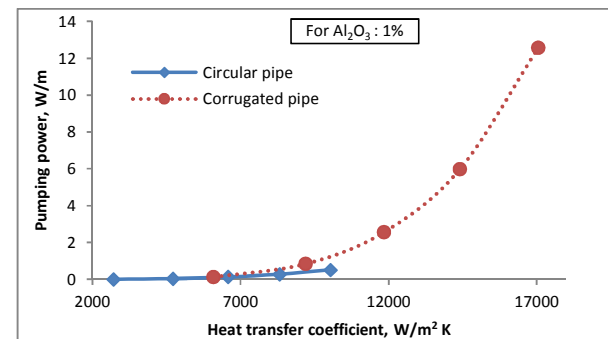


Figure 6. Comparison between the pumping power for corrugated pipe and plain pipe with the change in the heat transfer coefficient

The requirement of pumping power is always higher for corrugated pipe than the plain pipe in case of 1% Al_2O_3 -water nanofluid. At $h=8000$ W/m^2 K, the pumping power for corrugated pipe is 0.495 W/m and that for plain pipe is 0.26 W/m. The same results are obtained for water and nanofluid of other volume fractions as well.

Effect of Using Nanofluid on the Pumping Power for Circular Plain Pipe Compared to Water

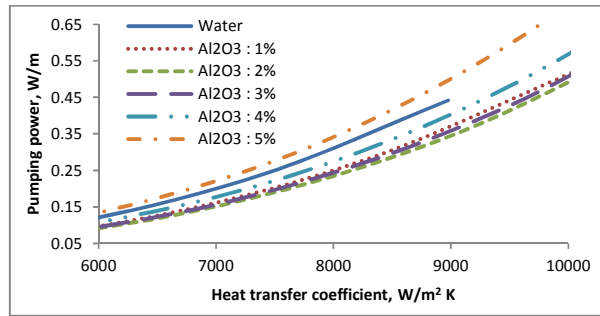


Figure 7. Comparison of pumping power for nanofluid with that for water at the same heat transfer coefficient (zoomed from $h=6000$ W/m^2K to $h=10000$ W/m^2K)

As the volume fraction of the nanofluid increases the requirement of pumping power reduces for the same heat transfer rate. This pumping power is the lowest at a volume fraction of 2% beyond which the pumping power increases again. And at a volume fraction of 5%, the pumping power requirement is more than that of water when same heat transfer rate is required [see figure 7].

Effect of Using Nanofluid on the Pumping Power for Corrugated Pipe Compared to Water

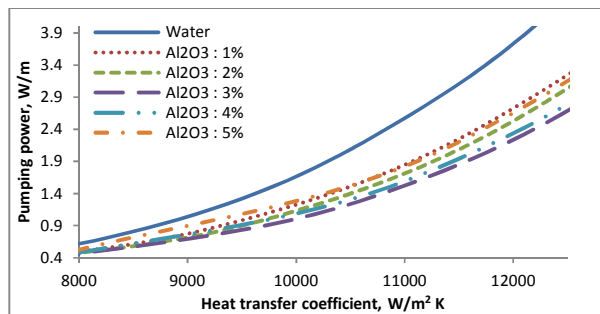


Figure 8. Comparison of pumping power for nanofluid and water for corrugated pipe (zoomed from $h=8000$ W/m^2K to $h=12500$ W/m^2K)

The use of nanofluid results in the reduction in the pumping power requirement compared to water. In figure 8, it is seen that with the increase in the volume fraction of the nanofluid, the pumping power requirement decreases compared to water. At a volume fraction of 3%, pumping power is the lowest. Unlike plain pipe, here at a volume fraction of 5%, the pumping power requirement is still lower than that of water.

Conclusions

Corrugated pipe and nanofluid both have been used to enhance the heat transfer rate compared to system of water flowing through a simple plain pipe. However, this enhancement is achieved at the cost of pumping power required for the flow. This increase in the pumping power is much higher when corrugated pipe is used. Addition of nanofluid reduces this pumping power requirement and the minimum occurs at a volume fraction of 3% in case of corrugated pipe. For circular plain pipe, at a volume fraction of 5%, the increase in heat transfer rate occurs at the penalty of pumping power which is more than that of water. But in case of corrugated pipe, the pumping power requirement for the flow of 5% Al_2O_3 -water nanofluid is much lower than that of water.

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