Study of Heat Transfer Performance and Pumping Power Improvement of Nanofluid Through a Rough Circular Tube

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Abstract -*The higher thermal conductivity of solid nanoparticles is utilized in enhancing the overall thermal and hydrodynamic behavior of working fluid for optimum design of compact heat exchangers. The addition of nanoparticles in base fluid within a preferred volume fraction so called nanofluid provides superior thermo-physical properties compared to that of base fluid which finally assists in improving the overall heat transfer characteristics. In the present work,* Al_2O_3*-water nanofluid is employed in a rough circular tube subjected to constant heat flux to investigate the forced convection heat transfer characteristics and pumping power under turbulent flow condition. The study is performed for a wide range of Reynolds number- 10,000 to 30,000 with different volume fraction of nanoparticles (1% to 5%) and different relative roughness of pipe wall (0.001, 0.002 and 0.003). SST k-ω turbulence model for single phase analysis is adopted and finite volume method is employed for solving the transport equations (mass, momentum and energy) and turbulence quantities. The heat transfer rate is substantially enhanced by the implementation of nanofluid in rough tube compared to that of smooth tube with an increase of Reynolds number and volume fraction. Finally the optimum volume fraction of nanoparticles is determined for which nanofluid requires lower amount of pumping power compared to water. The reduction of mass flow rate for nanofluid is also calculated which results better thermal performance.*

Key Words:**Nanofluid, heat transfer, pumping power, relative roughness, turbulent**

1.INTRODUCTION

This In recent years, in the field of sustainable energy, the implementation of nanofluid has been exploited tremendously in heat and mass transfer applications because of its extreme demand as an effective and suitable heat transfer fluid to reduce the size and material cost for designing compact heat exchangers. Nanofluid refers to the mechanism of stable and uniform suspension of solid metallic nanoparticles of size less than 100 nm in conventional base fluid like water, oil or ethylene glycol etc. The solid nanoparticles have higher thermal conductivity which potentially influences in enhancing the overall thermal and hydrodynamic characteristics of the working fluid [1-4]. To accomplish equivalent heat transfer rate, nanoparticles are dispersed in base fluid with low volume concentration which significantly change the thermo-physical properties of working fluid and eventually enhances the heat transfer rate [5, 6]. Numerous researches have been carried out with nanofluids employed in different geometries to enhance the heat transfer performance in many aspects of heat and mass transfer applications.

Maiga et al. [7,8] investigated the forced convection heat transfer enhancement utilizing γ-Al_2O_3-water and γ-Al_2O_3-ethylene glycol nanofluids inside a uniformly heated circular tube under constant heat flux for both laminar and turbulent flow conditions. Heat transfer coefficients were augmented by the increase of Reynolds number and volume fraction. Finally a correlation was provided for mean Nusselt number in terms of Reynolds number and Prandtl number. Behzadmehr et al. [9] performed numerical investigation on turbulent heat transfer in a circular tube subjected to constant heat flux using Cu-water nanofluid with 1% volume fraction and multiphase approach was employed. Improvement of heat transfer coefficients was shown and results were compared with single phase. Zeinali et al. [10] experimented to observe the enhancement of heat transfer rate for laminar flow using Al_2O_3-water nanofluid through a circular pipe for a wide range of Peclet number 2,000 to 6,000 and volume fraction of 0.2% to 2.5%. Bianco et al. [11] reported improvement of Nusselt number of Al_2O_3 -water nanofluid in a circular tube with an increase of Reynolds number and volume fraction adopting both single and multiphase analysis. Laminar flow was considered for a range of Reynolds number 200 to 1200 and volume fraction 1% to 4%. Izadi et al. [12] studied numerically laminar forced convection in an annulus using Al_2O_3-water nanofluid to observe the heat transfer enhancement as well as the thermal and hydrodynamic behavior of fluid flow. Yurong et al. [13] worked on the heat transfer intensification of TiO_2-water nanofluid through straight circular tube adopting both single phase and combined Euler and Lagrange method for laminar flow condition. Fotukian and Nasr [14] reported significant improvement of heat transfer and studied the pressure drop of CuO-water nanofluid through a circular tube for a range of Reynolds number of 5,000-33,000. Heat transfer coefficients were increased by 25% with a penalty of pressure drop 20%

compared to base fluid water. Syam and Sharma [15] performed numerical study on convective heat transfer coefficient and friction factor in a circular tube fitted with twisted tape inserts with Al_2O_3-water nanofluid. Heat transfer enhancement was shown for a range of Reynolds number 700 to 2200 and volume concentration up to 0.5% and a correlation was developed for Nusselt number and friction factor based on the experimental evaluations. Nasiri et al. [16] found enhancement of heat transfer with Al_2O_3 and TiO_2 nanoparticles dispersed into water through an annular duct under constant wall temperature for a range of Reynolds number 4,000 to 12,000 and volume fraction 0.1% to 1.5%. Sebastien et al. [17] studied heat transfer and hydraulic behavior of SiO_2-water nanofluid through a horizontal tube with imposed wall temperature and maximum 60% improvement of heat transfer coefficient compared to pure water were shown. Heyhat et al. [18, 19] conducted experiments to observe the enhancement of heat transfer and pressure drop of Al_2O_3-water nanofluid in a circular tube under constant wall temperature for both laminar and turbulent flow condition. Javad and Amir [20] analyzed numerically the thermal performance and pressure drop in a circular tube under constant heat flux using Al_2O_3 nanoparticles dispersed in EG/water mixture with different volume concentrations (1% to 10%) and Reynolds number ranging from 1,500 to 10,000. Ahmad et al. [21] experimented with SiO_2, Al_2O_3, TiO_2 nanoparticles suspended in de-ionized water to obtain considerable augmentation of laminar convective heat transfer and pressure drop through a straight tube under constant heat flux. Ehsan et al. [22] evaluated the cooling performance of three different water based nanofluids (Al_2O_3, TiO_2 and CeO_2) through a circular tube for laminar forced convection. Esmaeilzadeh et al. [23] experimented to investigate the hydrodynamics and laminar heat transfer characteristics of γ-Al_2O_3-water nanofluid through a horizontal tube under constant heat flux. Average heat transfer coefficient was increased by 19.1% for 1% volume concentration and finally a correlation for Nusselt number based on experimental values was proposed. Adi T. Utomo et al. [24] studied the effect of water based alumina, titania and carbon nanotube nanofluids on the enhancement of laminar convective heat transfer coefficient in a horizontal pipe with constant heat flux. Hemmat et al. [25] conducted experiment to evaluate thermo-physical properties (Thermal conductivity and viscosity) of MgO-water nanofluid and observed significant improvement of heat transfer in a circular tube for turbulent flow condition. A correlation was developed for the dynamic viscosity based on the experimental results. Similarly a lot of numerical and experimental studies were performed employing different geometries with corrugation applied at the wall boundary as well as the implementation of nanofluid in order to obtain better heat transfer performance [26-30]. In our previous work [31], heat transfer enhancement was shown with aluminum oxide (Al_2O_3), Copper (Cu) and titanium dioxide (TiO_2) nanoparticles employed in three different corrugated channels (are sine-shape, V-shape and rectangular shape). The maximum enhancement of heat transfer coefficient was 50.15% with Al_2O_3-water nanofluid compared to water in sine shaped channel. In another work of ours [32], heat transfer as well as pumping power was studied in V-shaped corrugated tube with Al_2O_3-water nanofluid for a range of Reynolds number 4,000-20,000 and particle volume fraction 1%-5%. The requirement for minimum pumping power was found for 3% volume fraction of nanoparticles which was 40% lower compared to base fluid water.

From the literature review, it may be mentioned that most of the scientific articles emphasized on the heat transfer augmentation using nanofluid in different heat transfer applications but the justification of implementing nanofluid in terms of increased pumping power due to the improvement of thermo-physical properties has not been studied elaborately. Moreover, in most of the geometries, the wall of the fluid domain was considered as smooth. So, the scope of our present work is to investigate numerically the enhancement of turbulent forced convective heat transfer and reduction of pumping power through a rough circular tube under constant heat flux. The simulation is performed for a range of Reynolds number of 10,000 to 30,000 with a range of volume fraction of 1% to 5% and rough circular tubes with three different values of relative roughness of 0.001, 0.002 and 0.003. A significant enhancement of heat transfer properties like convective heat transfer coefficient and Nusselt number were observed with an increase of Reynolds number and volume fraction and the results are compared with smooth tube. The optimum volume fraction is calculated for which nanofluid showed better performance in terms of reduced pumping power and mass flow rate.

2. NOMENCLATURE

Ac	Cross-sectional area of circular tube
Aw	surface area of circular tube
Cp	Specific heat at constant pressure
Dh	Hydraulic diameter
dp	Particle diameter
f	Friction factor
hc	Average heat transfer coefficient
k	Turbulence kinetic energy
kt	Thermal conductivity
M	Molecular weight of the base fluid molecule
W	Pumping power per unit length
ΔP	Pressure drop
N	Avogadro number
Q	Heat transfer
T	Temperature
T0	Reference temperature, 273K

U Velocity of flow at inlet
KB Boltzmann's constant
Greek symbols
ϕ Volume fraction
μ Dynamic viscosity
ν Kinematic viscosity
ρ Mass density
ε Average height of roughness
ω Specific dissipation rate
Subscripts
av Average value
i Value at inlet
o Value at outlet
nf Nanofluid
bf Basefluid
p Nanoparticle
w Value at wall
Dimensionless parameter
Nu Nusselt number
Re Reynolds number
Pr Prandl number

3. GOVERNING EQUATIONS

The two dimensional transport equations for continuity, momentum and energy equation for forced convection under turbulent flow and steady state condition are expressed as follows [33]

Continuity equation:

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

Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[?\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right] + \frac{\partial}{\partial x_j}(-\rho\overline{u_i' u_j'})$$

Energy equation:

$$\frac{\partial}{\partial x_i}(-\rho u_i T) = \frac{\partial}{\partial x_j}((\Gamma + \Gamma_t)\frac{\partial T}{\partial x_j})$$

Here Γ is molecular thermal diffusivity and Γ_t is turbulent thermal diffusivity which can be expressed as:

$$\Gamma = \frac{?}{Pr'}$$

$$\Gamma_t = \frac{?_t}{Pr_t'}$$

The SST k-ω model has a similar form to the standard k-ω model. The transport equations for the SST k-ω model are as follows [33]:

The Reynolds number for the flow of nanofluid is expressed as

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

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

$$Q_{nf} = \dot{m}_{nf} C_{P_{nf}} (T_{bo} - T_{bi})_{nf}$$

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

$$T_b = \frac{\int_0^{A_c} uT\, dA_C}{\int_0^{A_c} u\, dA_C}$$

A_c is cross-sectional area of the circular section. 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, the logarithmic temperature difference between the wall and the fluid which is calculated as

$$\Delta T_m = \frac{(T_W - T_i) - (T_W - T_o)}{\ln[(T_W - T_i) - (T_W - T_o)]}$$

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

$$Nu = \frac{h_c D_h}{k_{tnf}}$$

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

$$W = \frac{\frac{\pi}{4} {D_h}^2 U_{av} \Delta P}{L}$$

Where ΔP is differential pressure differerce

$$\Delta P = \frac{f L \rho {U_{av}}^2}{2 D_h}$$

4. THERMOPHYSICAL PROPERTIES

The equation of density of nanofluid considering water as base fluid is given by the following equation taken from the paper of Pak and Cho [34]

$$\rho_{nf} = \rho_p \phi + \rho_{bf}(1 - \phi)$$

Specific heat of nanofluid is calculated using the following equation [34]

$$C_{nf} = (1 - \phi)C_w + \phi C_p$$

The dynamic viscosity of nanofluid is given by the following empirical correlation derived by Corcione [35] with 1.84% of standard deviation.

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

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

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

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. An arbitrary size of nanoparticle of 50 nm is considered for the synthesis of the Al_2O_3-water nanofluid.

Thermal conductivity of the nanofluid is obtained using the correlation proposed by Ko and Kleinstreuer [36] which is given in equation 14 to consider the effect of temperature of the nanofluid and the Brownian motion of the solid particles.

Thermal conductivity,

$$k_{tnf} = \frac{k_{tp}+2k_{tbf}+2(k_{tp}-k_{tbf})\phi}{k_{tp}+2k_{tbf}-(k_{tp}-k_{tbf})\phi}k_{tbf} + 5\times 10^5\beta\phi\rho_p C_p\sqrt{\frac{K_B T}{\rho_p D}}f(T,\phi)$$

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

5. NUMERICAL MODEL AND BOUNDARY CONDITION

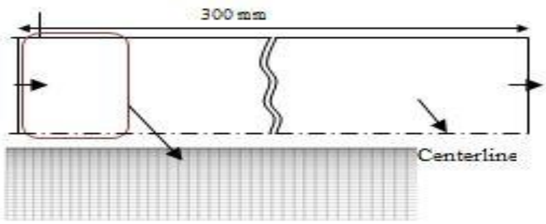

Fig-1: Physical model of the solution domain with a snapshot of the mesh

The solution domain of the present numerical study is a simple pipe which is shown in figure 1. The commercial CFD package- ANSYS Fluent 12.0 has been employed to solve the governing equations and a 2D simulation with axisymmetric model has been carried out which simplifies the computational process. The diameter of the pipe is taken as 3 mm. A constant heat flux of 5000 W/m2 is applied on the wall of the pipe. A turbulent flow, with a uniform velocity and a temperature of 300 K at the inlet, assuming no slip condition at the wall, is considered through both smooth and rough pipe for the numerical simulation. The wall temperature and the bulk temperature are measured at a distance of 275 mm from the inlet which ensures the readings at a fully developed flow. The pressure drop is measured in the region of 275 mm to 290 mm from the inlet of the pipe. SST k-ω turbulent model has been selected considering roughness at the wall boundary of the tube for more accurate formulation at the boundary layer. Turbulent intensity is taken as 5% at the inlet and pressure outlet is considered. The SIMPLE algorithm and second order upwind discretization for pressure, momentum, turbulent kinetic energy, specific dissipation rate, and energy are used to set the solution criteria.

6. GRID INDEPENDENCE AND CODE VALIDATION

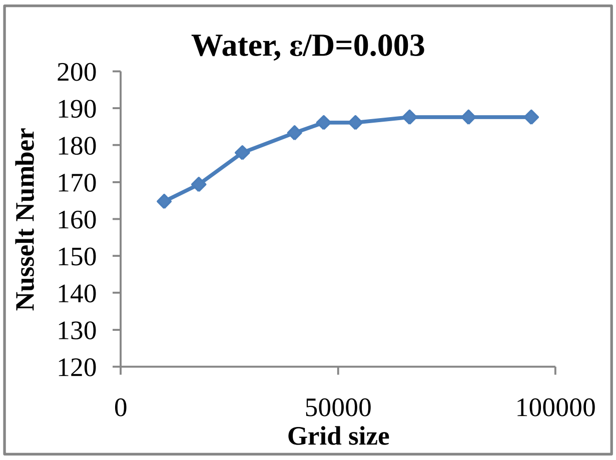

Fig-2: Effect of different grid size on the Nusselt number for relative roughness of 0.003 with water flowing through the tube

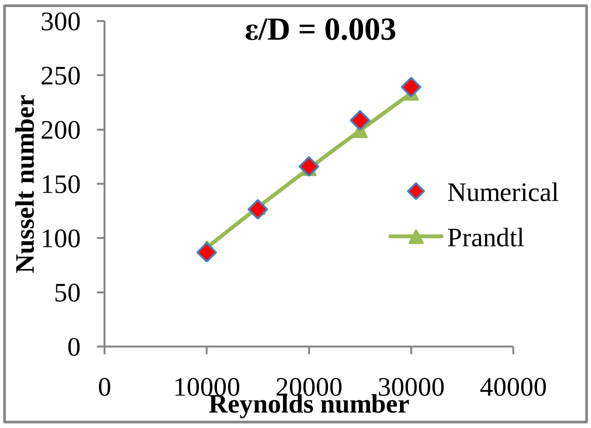

Fig-3: Validation of Nusselt number obtained with Prandtl correlation [39]

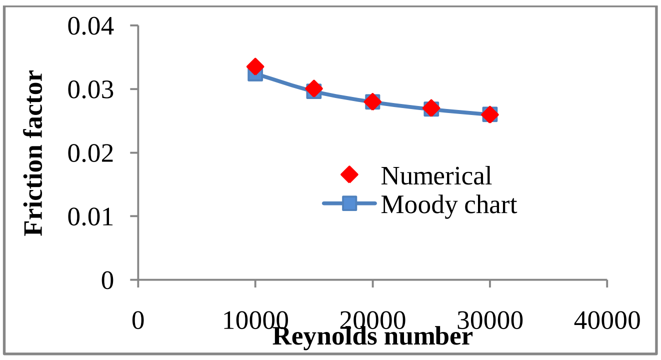

Fig-4: Comparison of friction factor from numerical study with Moody chart.

Grid independence study is carried out in order to find the optimum grid size without compromising the accuracy of the results. Nusselt number has been calculated for different grid sizes with water flowing through a tube of relative wall surface roughness of 0.003. It may be noted that the Nusselt number for water increases with the increase in the grid size of the solution domain and becomes constant beyond the grid size of 950*65. However with the change in the value of ε/D, the optimum grid size also changes. For a tube with smooth wall, the grid size beyond which there is no change in the value of Nusselt number is 700*40. This value is 800*50 for ε/D=0.001 and 900*60 for ε/D=0.002.
The Nusselt number and the friction factor have been calculated to validate the numerically found data with the well-established correlations. For smooth tube, the calculated Nusselt number is compared with the correlation developed by Notter and Sleicher [37] and the friction factor is validated with that obtained from Moody chart [38] which is used for validating friction factor for rough tube as well. However the Nusselt number obtained for the tube with wall surface roughness is validated with the correlation developed by Prandtl which is shown in figure 3 [39].

6. RESULTS AND DISCUSSION

6.1 Effect of roughness on heat transfer

Wall surface roughness contributes to the enhancement of the heat transfer by affecting the near wall flow behavior. In this numerical study, the heat transfer rate increases with the increase in the wall surface roughness. From Fig. 5, it is observed that relative roughness in the range of 0.001 to 0.003 results in 11.5% to 32.35% enhancement of heat transfer rate at a Reynolds number of 25000 compared to smooth tube. The extent to which this improvement takes place is greater in case of a higher Reynolds number. Similar trends have been observed for nanofluids of all volume fractions.

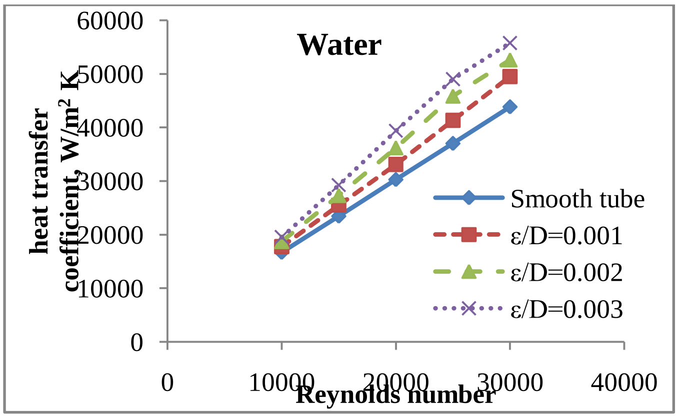

Fig-5: Comparison of heat transfer coefficient with Reynolds number for different rough surfaces.

6.2 Effect of roughness on pumping power

Surface roughness changes the near wall turbulence behavior which increases the coefficient of heat transfer. However this improvement in the heat transfer rate is obtained at the cost of pumping power. Fig. 6 shows the graph of pumping power per unit length of the tube vs. Reynolds number for different values of relative roughness of the wall surface. An increase in the relative roughness of the wall results higher pumping power requirement.

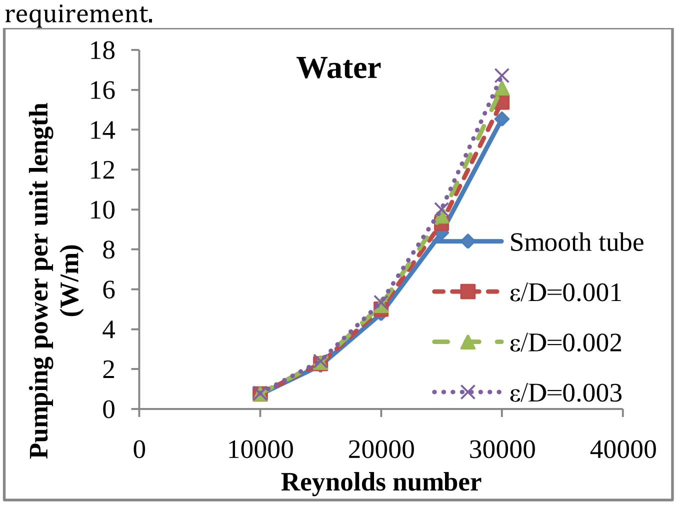

Fig-6: Pumping power requirement vs. Reynolds number for different surface roughness

6.3Effect of nanofluid on heat transfer

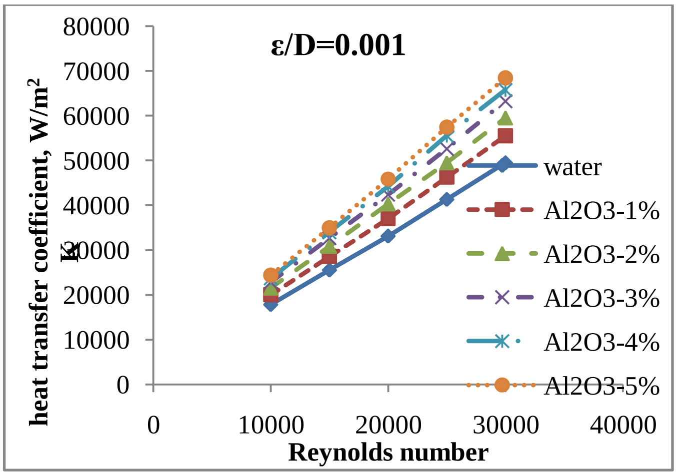

Fig-7: Coefficient of heat transfer with Reynolds number flowing through a tube of relative roughness of 0.001.

Using nanofluid instead of water enhances the heat transfer rate due to the change in thermophysical properties of the working fluid. This enhancement is shown in fig. 7.At a Reynolds number of 25000, the value of heat transfer coefficient improves by 12.03% to 39.08% for Al¬2O3-water nanofluid of volume fractions ranging from 1% to 5% compared to that of water.

6.4 Effect of nanofluid on pumping power

Addition of nanoparticles in the base fluid water increases the viscosity which results in the pressure drop in the test section. The increased pressure drop corresponds to the increased pumping power requirement for the flow per unit length. These results are shown in figure 8. Higher Reynolds number contributes to the pumping power requirement remarkably for water as well as for all volume fractions of nanofluid. The increase in the volume fraction of nanoparticles in the base fluid increases the pumping power. At a Reynolds number of 25000, the pumping power increases by 17.4% to 203% for the nanofluid of volume fractions ranging from 1% to 5% compared to that of water.

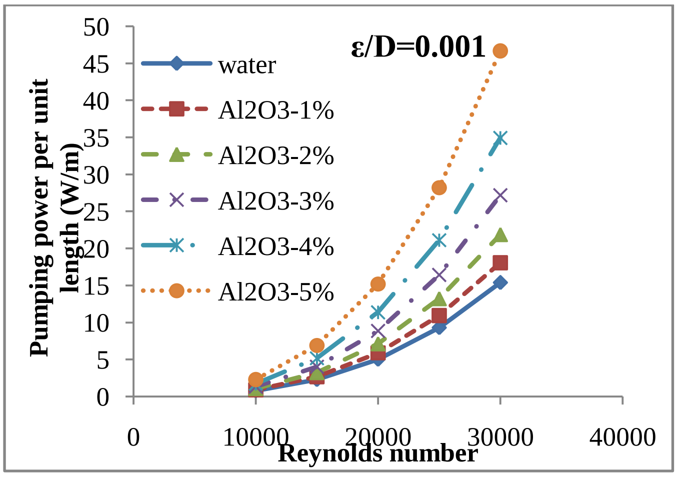

Fig-8: Pumping power requirement vs. Reynolds number for different working fluids

6.5 Optimum volume fraction of nanofluid in terms of pumping power

Both nanofluids and roughness at the wall boundary increases the heat transfer rate, but at the cost of the pumping power required for the fluid flow. However, this penalty in the pumping power may be justified if it results in a much higher value of heat transfer coefficient. Figure 9 shows the pumping power requirement in the range of heat transfer coefficient of 30000 W/m²K to 40000 W/m²K. The improvement of pumping power is achieved up to 3% volume fraction of the nanofluid for heat transfer coefficient ranging from 30000 W/m² K to 40000 W/m² K. The minimum pumping power requirement for the flow is obtained for the nanofluid of volume fraction of 2%. Al_2O_3-water nanofluid of 5% volume fraction always requires more pumping power than water for any value of heat transfer coefficient. The nanofluid of volume fraction of 4% needs slightly less pumping power than water at higher heat transfer coefficient. For smooth and rough tube of all three values of relative roughness, the minimum pumping power corresponding occurs at 2% volume fraction of nanofluid.

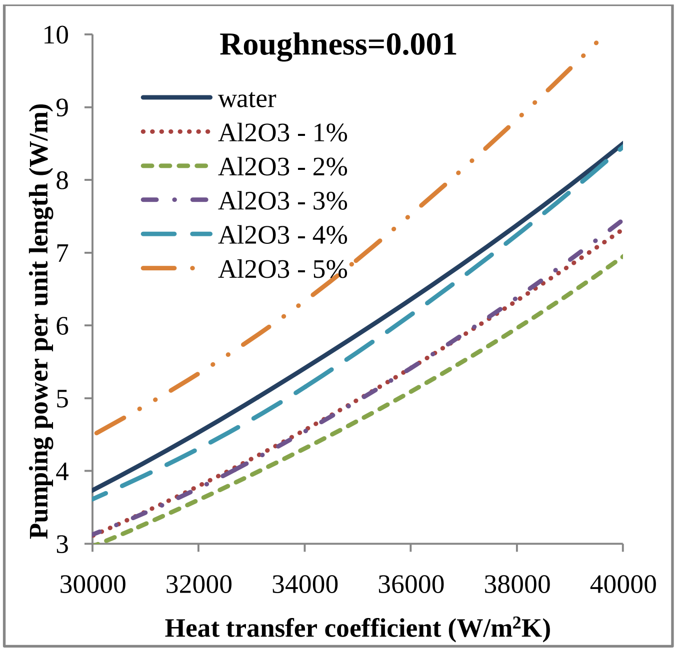

Fig-9: Variation of pumping power and corresponding heat transfer coefficient in the range of h=30000 W/m² K to 40000 W/m² K for a relative roughness of 0.001.

Fig. 10 illustrates the variation of requirement of pumping power per unit length with the change in heat transfer coefficient for different roughness of the wall surface. With the increase in the wall surface roughness, the pumping power reduces considering a constant value of heat transfer coefficient.

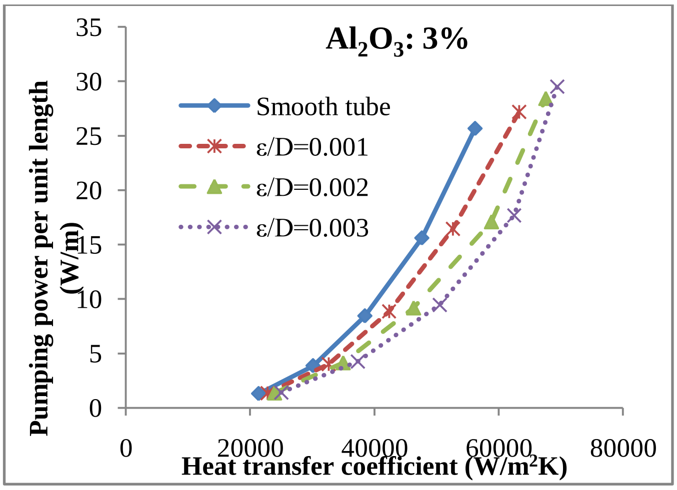

Fig-10: Pumping power vs. heat transfer coefficient for different roughness at wall boundary

6.6 Power advantage at optimum volume fraction

From Fig. 11, it is obvious that heat transfer coefficient increases when rough surface is used instead of smooth wall and nanofluid (ϕ=2%) is used instead of base fluid water. For extreme two cases, e.g. water flowing through smooth tube and nanofluid flowing through rough tube, an augmentation 33% in heat transfer coefficient is observed a Reynolds number of 20000.

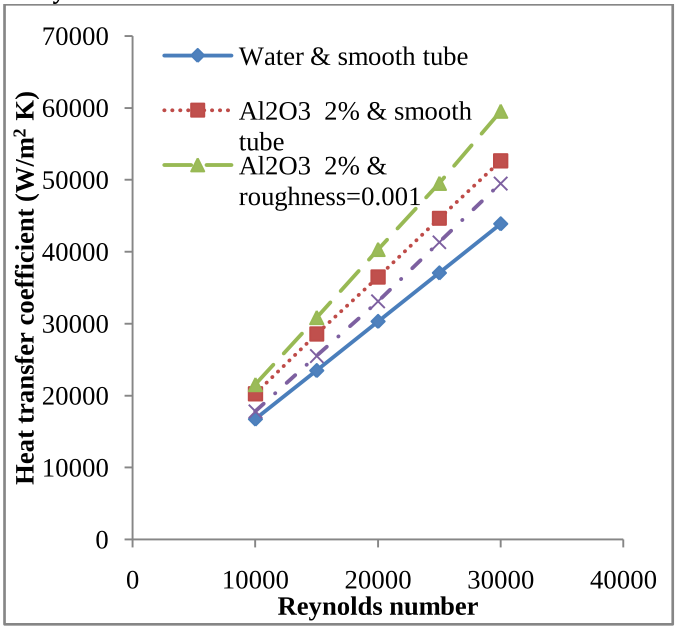

Fig-11: Comparison of heat transfer coefficient at different Reynolds number for four different working conditions.

Figure 12 shows the increase in pumping power requirement when nanofluid or roughness at the wall is used. The pumping power increases by 42% for "nanofluid and rough tube" condition compared to "water and smooth tube condition".

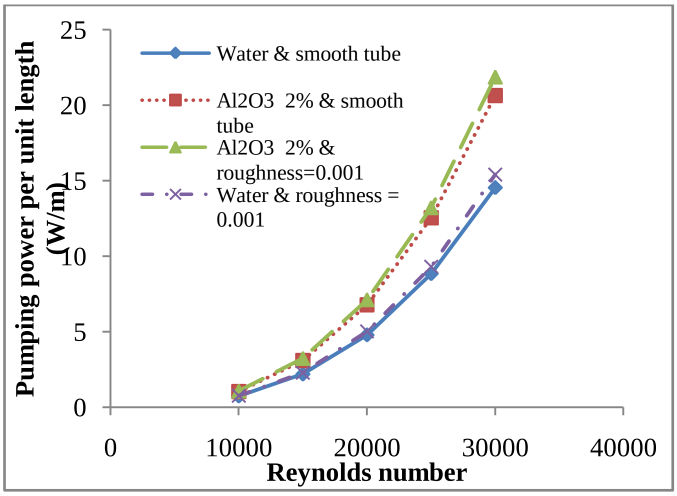

Fig-12: Comparison of pumping power requirement with Reynolds number for four working conditions.

CONCLUSION

The thermophysical properties are improved by using nanofluid which results in the enhancement of heat transfer coefficient compared to that of water. As the volume fraction of the nanofluid increases, this enhancement in the convective heat transfer coefficient also increases. The roughness at the wall surface of the tube also enhances the convective heat transfer rate. As the value of relative roughness increases, this heat transfer coefficient becomes higher.

Requirement of pumping power per unit length for the flow of the working substance through the tube is increased by both using nanofluid and roughness at the wall. The pumping power shows an increasing trend for the increase in the volume fraction as well as the relative roughness. At 2% volume fraction of nanofluid, the pumping power requirement per unit length is the lowest when a constant value of convective heat transfer coefficient is considered. Pumping power requirement is also reduced for a constant value of heat transfer coefficient when roughness is used at the wall surface. However in this case with the increase in the value of relative roughness, pumping power reduction rate shows an increasing trend. A combined case of using nanofluid and wall surface roughness yields in a better enhancement of heat transfer rate and further reduction of mass flow rate in terms of lower requirement of pumping power per unit length compared to the case when nanofluid or roughness is used individually.

REFERENCES

[1] S. Lee, S.U.S. Choi, S. Li, J.A. Eastman, 1999, "Measuring thermal conductivity of fluids containing oxide nanoparticles", Trans. ASME Journal of Heat Transfer (121) pp. 280–289.

[2] J.A. Eastman, S.U.S. Choi, S. Li, W. Yu, L.J. Thompson, 2001, "Anomalously increased effective thermal conductivities of ethylene glycol-based nanofluids containing copper nanoparticles", Appl. Phys. Lett. (78) pp. 718–720.

[3] Y. Xuan, Q. Li, 2000, "Heat transfer enhancement of nanofluids", International Journal of Heat and Fluid Flow (21) pp. 58–64.

[4] H. Xie, J. Wang, T. Xi, F. Ai, 2002, "Thermal conductivity enhancement of suspensions containing nanosized alumina particles", Int. J. Appl. phys. (91) pp. 4568–4572.

[5] S.K. Das, N. Putra, P. Thiesen, W. Roetzel, 2003, "Temperature dependence of thermal conductivity enhancement of nanofluids", Journal of Heat Transfer (125) pp. 567–574.

[6] X. Wang, X. Xu, S.U.S. Choi, 1999, "Thermal conductivity of nanoparticle-fluid mixture", J. Thermophys. Heat Transfer (13) pp. 474–480.

[7] Sidi E. Becaye Maigaa, Cong Tam Nguyena, Nicolas Galanisb, Gilles Roya, 2004, "Heat transfer behaviors of nanofluids in a uniformly heated tube", Superlattices and Microstructures (35), pp. 543-553.

[8] Sidi E. Becaye Maigga, Samy Joseph Palm, Cong Tam Nguyen, Gilles Roy, Nicolas Galanis, 2005, "Heat transfer enhancement by using nanofluids in forced

convection flows", International Journal of Heat and Fluid Flow (26), pp. 530–546.

[9] A. Behzadmehr, M. Saffar-Avval, N. Galanis, "Prediction of turbulent forced convection of a nanofluid in a tube with uniform heat flux using a two phase approach", International Journal of Heat and Fluid Flow 28 (2007) 211–219.

[10] S. Zeinali Heris, M. Nasr Esfahany, S. Gh. Etemad, "Experimental investigation of convective heat transfer of Al_2O_3/water nanofluid in circular tube", International Journal of Heat and Fluid Flow 28 (2007) 203–210.

[11] V. Bianco, F. Chiacchio, O. Manca, S. Nardini, "Numerical investigation of nanofluids forced convection in circular tubes", Applied Thermal Engineering 29 (2009) 3632–3642.

[12] M. Izadi, A. Behzadmehr, D. Jalali-Vahida, "Numerical study of developing laminar forced convection of a nanofluid in an annulus", International Journal of Thermal Sciences 48 (2009) 2119–2129.

[13] Yurong He, Yubin Mena, Yunhua Zhao, Huilin Lu, Yulong Ding, "Numerical investigation into the convective heat transfer of TiO_2 nanofluids flowing through a straight tube under the laminar flow conditions", Applied Thermal Engineering 29 (2009) 1965–1972.

[14] S.M. Fotukian, M. Nasr Esfahany, "Experimental study of turbulent convective heat transfer and pressure drop of dilute CuO/water nanofluid inside a circular tube", International Communications in Heat and Mass Transfer 37 (2010) 214–219.

[15] L. Syam Sundar, K.V. Sharma, "Laminar convective heat transfer and friction factor of al2o3 nanofluid in circular tube fitted with twisted tape inserts", National Conference in Mechanical Engineering Research and Postgraduate Students, 26-27 May 2010, Malaysia; pp. 227-240, ISBN: 978-967-5080-9501.

[16] M. Nasiri, S. Gh. Etemad, R. Bagheri, "Experimental heat transfer of nanofluid through an annular duct", International Communications in Heat and Mass Transfer 38 (2011) 958–963.

[17] Sebastien Ferrouillat, Andre Bontemps, Joao-Paulo Ribeiro, Jean-Antoine Gruss, Olivier Soriano, "Hydraulic and heat transfer study of SiO_2/water nanofluids in horizontal tubes with imposed wall temperature boundary conditions", International Journal of Heat and Fluid Flow 32 (2011) 424–439.

[18] M.M. Heyhat, F. Kowsary, A.M. Rashidi, S. Alem Varzane Esfehani, A. Amrollahi, "Experimental investigation of turbulent flow and convective heat transfer characteristics of alumina water nanofluids in fully developed flow regime", International Communications in Heat and Mass Transfer 39 (2012) 1272–1278.

[19] M.M. Heyhat, F. Kowsary, A.M. Rashidi, M.H. Momenpour, A. Amrollahi, "Experimental investigation of laminar convective heat transfer and pressure drop of water-based Al_2O_3 nanofluids in fully developed flow regime", Experimental Thermal and Fluid Science 44 (2013) 483–489.

[20] Javad Bayat, Amir Hossein Nikseresht, "Thermal performance and pressure drop analysis of nanofluids in turbulent forced convective flows", International Journal of Thermal Sciences 60 (2012) 236-243.

[21] Ahmad Azari, Mansour Kalbasi1, Masoud Derakhshandeh, Masoud Rahimi, "An Experimental Study on Nanofluids Convective Heat Transfer Through a Straight Tube under Constant Heat Flux", Fluid dynamics and transport phenomena, Chinese Journal of Chemical Engineering, 21(10) 1082—1088 (2013),DOI: 10.1016/S1004-9541(13)60618-7.

[22] Ehsan B. Haghighi, Mohsin Saleemi, Nader Nikkam, Zahid Anwar, Itziar Lumbreras, Mohammadreza Behi, Seyed A. Mirmohammadi, Heiko Poth, Rahmatollah Khodabandeh, Muhammet S. Toprak , Mamoun Muhammed, Bjorn Palm, "Cooling performance of nanofluids in a small diameter tube", Experimental Thermal and Fluid Science 49 (2013) 114–122.

[23] E. Esmaeilzadeh, H. Almohammadi, Sh. Nasiri Vatan, A.N. Omrani, "Experimental investigation of hydrodynamics and heat transfer characteristics of γ-Al_2O_3/water under laminar flow inside a horizontal tube", International Journal of Thermal Sciences 63 (2013) 31-37.

[24] Adi T. Utomo , Ehsan B. Haghighi, Ashkan I.T. Zavareh, Morteza Ghanbarpourgeravi, Heiko Poth, Rahmatollah Khodabandeh, Bjorn Palm, Andrzej W. Pacek, "The effect of nanoparticles on laminar heat transfer in a horizontal tube", International Journal of Heat and Mass Transfer 69 (2014) 77–91.

[25] Mohammad Hemmat Esfe, Seyfolah Saedodin, Mostafa Mahmoodi, "Experimental studies on the convective heat transfer performance and thermophysical properties of MgO–water nanofluid under turbulent flow", Experimental Thermal and Fluid Science 52 (2014) 68–78.

[26] H. Heidary , M.J. Kermani, 2010, "Effect of nano-particles on forced convection in sinusoidal-wall channel", International Communications in Heat and Mass Transfer (37),pp. 1520–1527.

[27] M.A. Ahmed, M.Z. Yusoff , N.H. Shuaib, 2013, "Effects of geometrical parameters on the flow and heat transfer characteristics in trapezoidal-corrugated channel using nanofluid", International Communications in Heat and Mass Transfer (42), pp. 69–74.

[28] Seyed Alborz Manavi, Abas Ramiar, Ali Akbar Ranjbar, 2013, "Turbulent forced convection of nanofluid in a wavy channel using two phase model", Springer, Heat Mass Transfer, DOI 10.1007/s00231-013-1276-y.

[29] Ashkan vatani, H.A. Muhammad, 2013, "Turbulent nanofluid flow over periodic rib-grooved channels", Engineering Applications of Computational Fluid Mechanics, Volume-7, no-3, pp. 369-381.

[30] M.A. Ahmed, N.H. Shuaib, M.Z. Yusoff, A.H. Al-Falahi, "Numerical investigations of flow and heat transfer enhancement in a corrugated channel using nanofluid", International Communications in Heat and Mass Transfer 38 (2011) 1368–1375.

[31] Noor, S., Ehsan, M. M., Mayeed, M. S., & Islam, A.K.M. Sadrul (2014, November). Study of Convective Heat Transfer for Turbulent Flow of Nanofluids Through Corrugated Channels. In ASME 2014 International Mechanical Engineering Congress and Exposition (pp. V08AT10A035-V08AT10A035). American Society of Mechanical Engineers.

[32] Noor, S., Ehsan, M. M., Salehin, S., & Islam, A.K.M. Sadrul (2014). Heat Transfer and Pumping Power Using Nanofluid in a Corrugated Tube. Proceedings of AFMC 2014 conference. Heat Transfer, 8, 11.

[33] Fluent Theory guide, ANSYS.

[34] B.C. Pak, Y. I. Cho, 1998, "Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles", Experimental Heat Transfer (11), pp. 151-155.

[35] Massimo Corcione, "Heat transfer features of buoyancy-driven nanofluids inside rectangular enclosures differentially heated at the sidewalls", International Journal of Thermal Sciences 49 (2010) 1536-1546.

[36] Koo, Junemoo, and Clement Kleinstreuer. "A new thermal conductivity model for nanofluids." Journal of Nanoparticle Research 6, no. 6 (2004): 577-588.

[37] Notter, R. H., and C. A. Sleicher. "A solution to the turbulent Graetz problem—III Fully developed and entry region heat transfer rates." Chemical Engineering Science 27, no. 11 (1972): 2073-2093.

[38] Moody, Lewis F. "Friction factors for pipe flow." Trans. Asme 66, no. 8 (1944): 671-684.

[39] Ozisik, M. Necati. "Heat transfer: a basic approach." (1985).

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