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Numerical Study of a Concentric Tube Heat Exchanger Using Dimpled Tubes with Al_2o_3 NanoFluid

[1]K.Senthil kumar, [2]P.Palanisamy, [3]R.Vinoth, [4]S.Perumal

[1]Associate Professor, [3] Asst Professor, Department of Mechanical Engineering, Adhiyamaan College of Engineering, Hosur, Tamil Nadu, INDIA
[2]Professor, Department of Mechanical Engineering, Dr.Navalar College of Engineering, Thiruchirapalli, Tamil Nadu, INDIA
[4]PG Student, Department of Mechanical Engineering, Adhiyamaan College of Engineering, Hosur, Tamil Nadu, INDIA

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ABSTRACT

The heat exchanger device such as concentric tube heat exchanger faces challenges to obtain the best thermal performance of heat exchanger devices plays important role in the performance of cooling system. The conventional coolant offer low thermal conductivity. The numerical study of dimpled tube with Nanofluids is important to improve the thermal performance system in heat exchange. In the present work, a concentric tube heat exchanger has been numerically modeled for heat transfer and a pressure drop characteristic under different flow rates are investigated. Reynolds numbers for the inner tube ranged from 1000 to 5000. The overall heat transfers were calculated in parallel heat exchanger. CFD Analyses were carried out with flue gas in the annulus side and Al_2O_3 Nanofluid in the inner tube side. The three-dimensional governing equations for momentum, continuity, and heat transfer were solved using a finite volume based computational fluid dynamics (CFD) code. The numerical results indicated that the Nusselt number for the spherical dimpled tube and ellipsoidal dimpled tube are 35.7% and 63.59% higher than that for the smooth tube in same Reynolds number. The friction factors in the dimpled tube increase by 52.7% and 87.27% for ellipsoidal and spherical dimples compared to the smooth tube. Heat transfer rate of spherical dimpled tube with Al_2O_3 Nanofluid is 40% better than in the plain tube and ellipsoidal dimpled tube with Al_2O_3 Nanofluid is 46% better than in the plain tube with Al_2O_3 nanofluid. Thus, the ellipsoidal dimpled tube could enhance heat transfer more efficiently than smooth and spherical dimpled tube with Al_2O_3 Nanofluid.

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INTRODUCTION

Effective utilization of available energy has become the need of the hour today. This obviously requires effective devising. When it concerns with heat energy the devices are heat exchanger. Heat exchangers are used in a variety of applications like process industries, thermal power plants, air conditioning equipments, refrigerators, radiators for space vehicles, automobiles etc. Increase in heat exchanger performance can lead to more economical design of the heat exchanger which can help to move energy, material and cost savings related to a heat exchange process. The present investigation augmented surface has been achieved with dimples strategically located in a tube to increase the thermal performance of heat exchanger. These techniques can be divided into two groups viz, passive and active. Active techniques which need the external power source and passive techniques which do not need the external power source. Some examples of passive techniques are surface modification, use of inserts and use of dimples etc. conventional heat transfer fluids like water and ethylene glycol are not capable of achieving designed heat transfer rate due to their low thermal conductivity. The uses of Nanofluids in replace of conventional fluids have some significant enhancement in heat transfer.

David J. Kukulka (2011) has been conducted experimental studies on the heat transfer co efficient in enhanced and smooth tubes. They stated that Increases in heat transfer for most Enhancement tubes are in excess of 120% over smooth tubes and minimize the fouling rate. Chinaruk Thianpong (2009) have been studied an experimental work of the dimpled tube with a twisted tape and plain tube in turbulent flow. Their results show the Heat transfer rate and friction factor dimpled tube with twisted tape, are respectively 1.66 to 3.03 and 5 to 6.31 times of those in the plain tube. Pedro G.Vicente (2002) studied the Heat transfer rate of ten helically dimpled tubes with different height and pitch ratios in turbulent flow condition. Their results show the Heat

Corresponding Author: K.Senthil kumar, Associate Professor, Asst Professor, Department of Mechanical Engineering, Adhiyamaan College of Engineering, Hosur, Tamil Nadu, INDIA.
Tel: +919894333716 E-mail: senthilace@gmail.com

transfer increases from 20% to110% and total area can be reduced from 80% to 20% of those in the smooth tube. Juin Chen (2001) have done experimental studies of six copper dimple tubes of varying geometry's were used for comparison with a standard smooth tube. Best dimple tube six which had the higher heat transfer coefficient increases from 1.25 and 2.37 times than those for the smooth tube due to large geometry dimensions. S.Suresh (2011) studied the heat transfer and friction factor of helically dimple tube with different volume concentrations of Cuo/water Nanofluid under turbulent flow. The heat transfer results is 19%, 27% and 39% (0.1%, 0.2% and 0.3% volume concentrations of nanoparticles in a fluid) and Friction factor are 2–10% higher than the plain tube. Sharma (2009) has done an experimental study using twisted tapes in the transition flow of low volume concentration Al_2o_3 nano fluid and reported considerable enhancement of convective heat transfer compared to flow with water. Pattern along the tube of the concentric tube heat exchanger with the increased area on the tube side is known as dimpled tube. Heat transfer enhancement of tubular heat exchangers with dimpled tubes by Kalinin (1991), and dimpled and helical tubes for compact heat exchangers by Giovannini (1991) Corrugated tubes have been studied by Marto (1979) and cope (1945) was the first to study about three dimensional roughness. He analyzed roughness produced by knurling the inner tube surface. Dipprey (2005) studied that the tube surfaces has a close-packed sand grain type roughness, which resembles natural roughness because of its three dimensional nature and random shape of the roughness elements. As a result, the main concern of this study is to model the helically finned and corrugated tubes in the FLUENT–GAMBIT CFD program Fluent release (2009) and then determine the friction factors, convective heat transfer coefficient and pressure drop by means of the FLUENT CFD program Zdaniuk (2008) and test the models reliability with several sets of basic heat transfer data Suresh (2011) belonging to the flow in smooth and enhanced tubes. The dimple tube dimensions are shown in table 1.

In the present work investigate the heat transfer and pressure characteristics of the concentric tube heat exchanger in the parallel flow regime using a dimple tube with Al_2O_3 Nanofluid .The different dimple tube geometry's (ellipsoidal and spherical) are compared with plain tube with Al_2o_3 Nanofluid. The Flue gas is flow through the outer tube and Nanofluid is flow through the inner tube.

MATERIALS AND METHODS

a. Nanofluid Preparation and Evaluation of Properties:

Nanofluid is preparing by two step method. The specifications of the Nano Particles are as follows: Alumina (Al_2O_3), Nano Dur 99.5%, Average particle size= 40–50 nm, Surface area = 32–40 m^2/g, Molecular weight = 101.96. Nanofluid with a required volume concentration of 0.1% was prepared by dispersing specified amounts of alumina Nano particles in de ionized with ethylene glycol. The alumina particles were characterized by X-ray diffraction technique and the average particle size were calculated by using J.C.Maxwell equation (1954). To make the Nano Particles more stable and remain more dispersed in ethylene glycol.

The thermo physical properties of nanofluid for a volume concentration of (ϕ) were calculated at the average bulk temperature of the nanofluid using the regression correlations widely used in the literature.

The density of Al_2O_3 Nanofluid (ρ_{nf}) was determined using Suresh's equation (2011),

$$\rho_{nf} = \phi\rho_s + (1-\phi)\rho \tag{1}$$

The effective thermal conductivity of (k_{nf}) nanofluid can also be evaluated using the J.C.Maxwell equation (1954), for nanofluid with volume fraction less than unity. Maxwell equation is given by,

$$k_{nf=} \frac{k_s+2k+2\phi(k_s-k)}{k_s+2k-\phi(k_s-k)} \tag{2}$$

The specific heat of the nanofluid (cp_{nf}) is calculated using Suresh's equation (14),

$$(\rho C_p)_{nf} = (1-\phi)\,(\rho C_p) + \phi(\rho C_p)s \tag{3}$$

b. CFD Modeling:

Geometries for the three different concentric tube heat exchangers were created in Pro-E and exported as stereo lithography files. The outer tube specifications of concentric tube heat exchangers are 76mm of outer diameter and 74mm inner tube diameter and made up of galvanized iron (GI). The Inner tube is made up of copper material. Because it's having higher thermal conductivity. The inner tube specifications of the concentric tube heat exchanger with ellipsoidal and spherical dimple geometries sketches are shown in Fig.1a and Fig. 1b and where ϕ stands for the projected diameter of spherical dimples, ϕ_1 and ϕ_2 represent the projected major and minor axis of ellipsoidal dimples respectively, p is the longitudinal space, h is the dimple height; and θ is the angle between two dimples. The geometrical parameters of the spherical and ellipsoidal dimpled tubes are shown in Table 1. GAMBIT was used to plot and mesh the model of the test tubes. The problem under investigation is a two-dimensional (axis symmetric), parallel flow of water flowing inside smooth, spherical and ellipsoidal dimpled tubes, all having an outer diameter of 10 mm and a length of 2000 mm. The concentric tube heat exchangers were imported into FLUENT 6.3.26. Commercial computational fluid dynamics software based on a control volume-finite difference formulation. A cylindrical coordinate system was used with a mesh size of 40 x 80 in the horizontal and vertical directions, respectively. Approximately 779662 cells for smooth tube, 783140 cells in the spherical dimpled tube and 785539 cells for ellipsoidal dimpled tube were used in the numerical solution. There is a difference between the mesh structures of spherical and ellipsoidal dimpled tubes. This difference has been caused by the high initial number and the spherical angle of the spherical dimpled tube. A high initial number and a high ellipse angle cause problems in forming ellipsoidal dimpled tube mesh structures. FLUENT (2005), uses the control volume technique to convert the Governing equations to algebraic equations so that they can be solved numerically. The control volume technique works by integrating the governing equations for each control volume, and then generates discretization of the equations, which conserves each quantity based on control volume. The fluid enters the circular tube with uniform axial velocity and temperature. The flow and thermal fields and pressure fields are assumed to be axisymmetric with respect to the horizontal plane that is parallel to the x-axis, as shown in Fig .2. and Fig.3.

Table 1: Dimensional configurations of the inner tubes.

Dimension	Smooth tube	Spherical tube	Ellipsoidal tube
Length of tube, L (mm)	2000	2000	2000
inner diameter of tube, d_i (mm)	9	9	9
outer diameter of tube, d_o (mm)	10	10	10
Thickness of wall, e (mm)	1	1	1
Angle between two dimples, θ (·)	-	90	90
Longitudinal space of dimple, p (mm)	-	20	20
Height of dimple, h (mm)	-	1.3	1.3
Projected diameter of spherical dimple, φ (mm)	-	3.5	-
Projected major axis of ellipsoidal dimple, φ_1(mm)	-	-	5
Projected minor axis of ellipsoidal dimple, φ_2 (mm)	-	-	3
Dimple mode	-	Spherical	Ellipsoidal

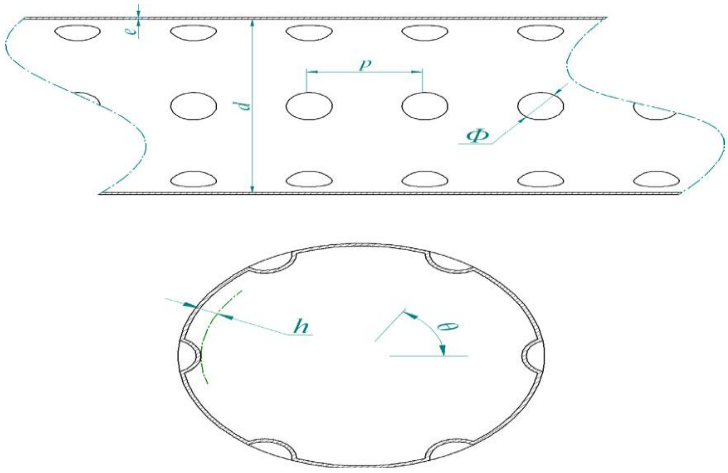


Fig. 1(a): Sketch of longitudinal and transverse cross-section of spherical dimpled tube.

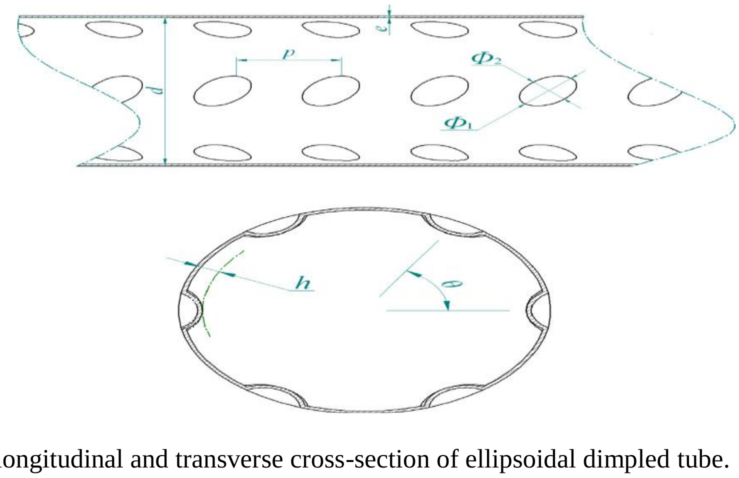


Fig. 1(b): Sketch of longitudinal and transverse cross-section of ellipsoidal dimpled tube.

c. Calculation of Heat Transfer:

The overall heat transfer coefficient is calculated with equation as follows:

$$U = \frac{Q}{A\Delta T_{lm}} = \frac{m_h C_{ph}(T_{hi} - T_{ho})}{A\Delta T_{lm}} \tag{4}$$

Where Q is the actual heat transfer rate in the test section, ΔT_{lm} is the logarithmic mean temperature difference, T_{hi} and T_{ho} are the temperatures of the hot fluid inlet and outlet, respectively. A is the inside surface area of the test section were taken from Juin Chen (2001), and is calculated as follows,

$$A = N[(L - n)/p]\,[(\emptyset/2)^2 + e^2]\pi \tag{5}$$

where d is the inside diameter of the test tube, L is the length of the test tube, N is the no of longitudinal dimple columns, n is the no of dimples, P is the pitch of dimples, ϕ is the dia of dimples, e is the depth of dimples K is the thermal conductivity. The nusselt number is calculated as follows,

$$Nu = hd/k \tag{6}$$

d. Calculation of Pressure Drop:

The laminar sub layer inside the dimpled tube will be disturbed by eddies generated by the dimples, which will lead to an increase in pressure drop and decrease the friction factor. It is used to calculate the friction factor using the following equation,

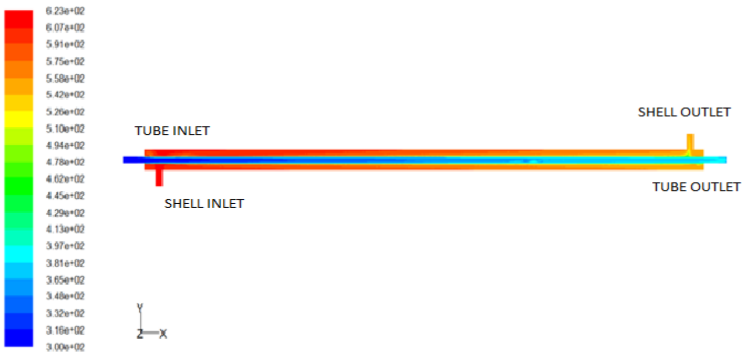
$$F = \Delta p/(\rho v^2/2)(d/l) \tag{7}$$

It can be observed that except density all other factors are independent of temperature. Hence, for comparison purpose, the pressure drop analyses were done under isothermal conditions. For nanofluid, the thermo-physical properties are calculated using Eqns. (1), (2) and (3).

RESULTS AND DISCUSSIONS

One can increase the heat transfer performance of heat exchangers by means of active and passive heat transfer enhancement techniques. Dimpled tubes create the rough surfaces used in passive heat transfer enhancement techniques and have been integrated into the air conditioning and refrigeration industries for effective tube-side performance. This implementation has been performed successfully because of the dimpled tubes ability to have a better heat transfer coefficient with only a moderate increase of the pressure drop penalty in comparison to the smooth tube.

This analysis study is conducted to calculate the heat transfer and pressure drop analyses for parallel flow for three different tubes. It carried out numerical studies for each tube, covering a wide range of Reynolds numbers starting from 1000 and going as far as 5000. The numerical model was permitted to run for a necessary time to create certain steady-state conditions. The effects of the spherical angle on the spherical tube and the ellipse angle on the ellipsoidal tube on the temperature difference between the Al_2O_3 nano fluid and tube inner wall across the tubes, heat flux, pressure drop penalty, and convective heat transfer coefficient and friction factor were numerically investigated. The thermodynamics and transport properties of the Al_2O_3 nano fluid were provided by the Fluent CFD Program (2005).



(A) PLAIN TUBE

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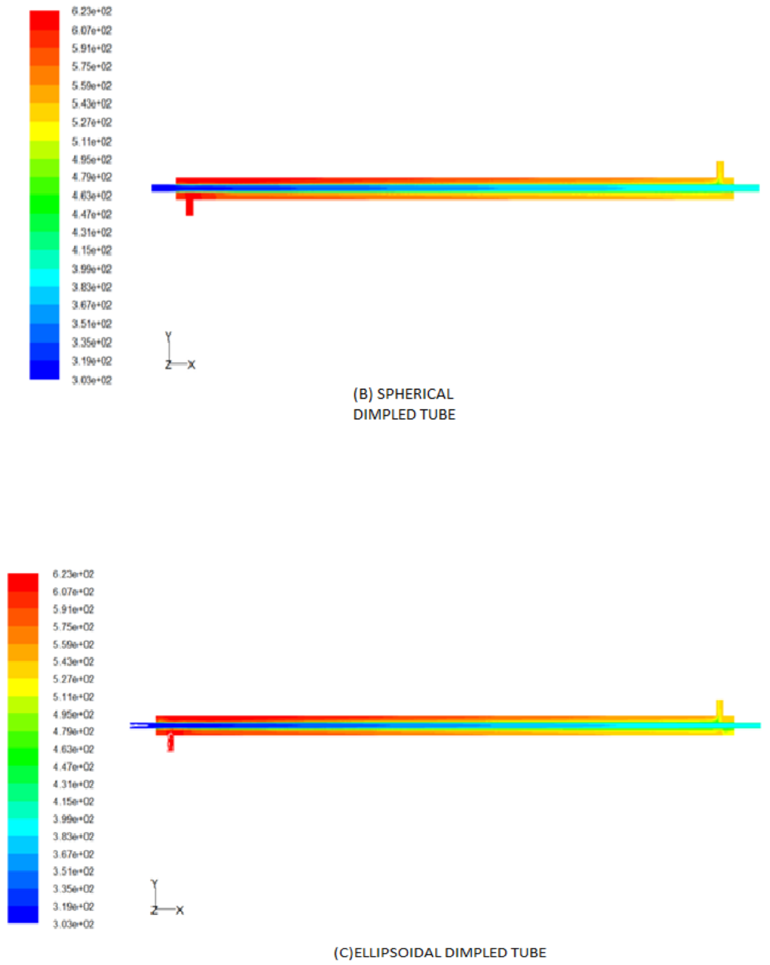
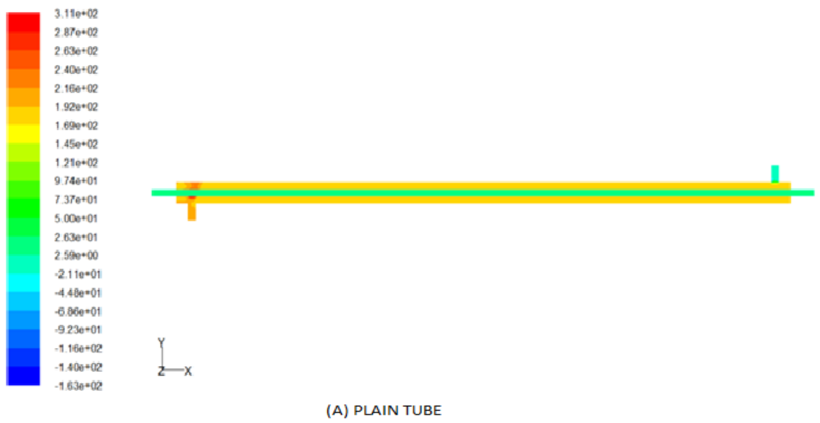
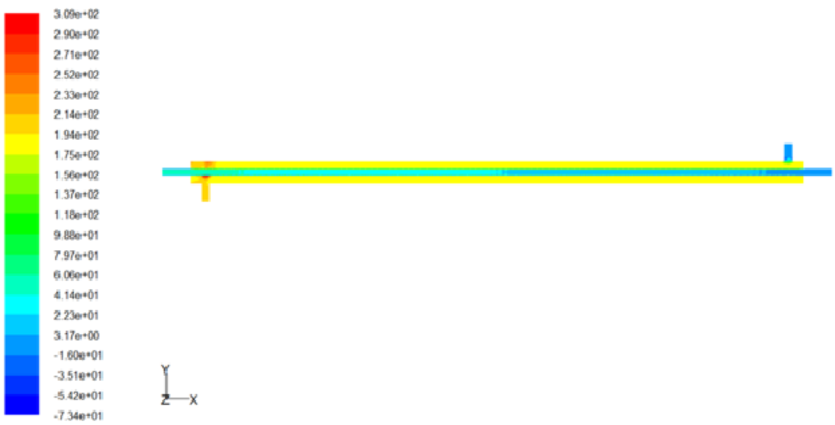
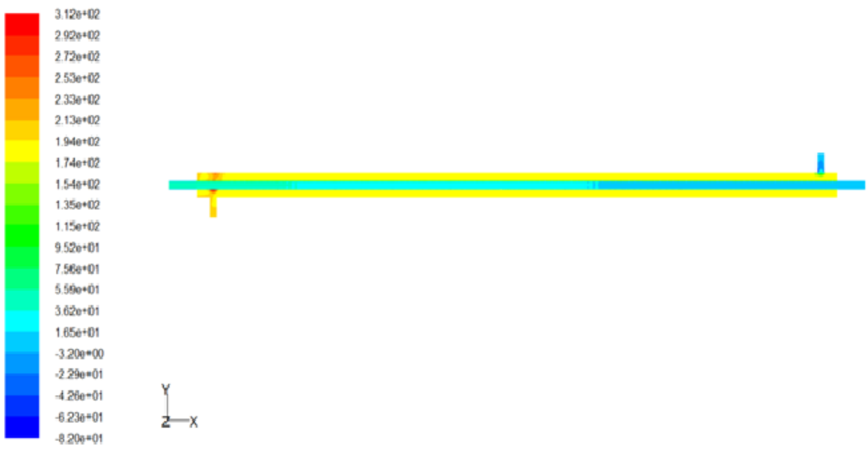


Fig. 2: Temperature distribution of plain and different dimple geometry tubes





(B) SPHERICAL DIMPLED TUBE



(C) ELLIPSOIDAL DIMPLED TUBE

Fig. 3: Pressure distribution of plain and different dimple geometry tubes.

a. Heat Transfer Study:

The data obtained from CFD analysis conducted using plain tube dimpled tubes and were used to increase the heat transfer in parallel flow regime. Fig, 4 shows the variation of Reynolds number (Re) with Heat transfer (Q) for plain tube and dimpled tubes. The use of Al_2O_3 nanoparticles as the dispersed phase in ethylene glycol significantly enhances the convective heat transfer in the parallel flow. Heat transfer rate of spherical dimpled tube with Al_2O_3 nano fluid is 40 % better than in the plain tube and ellipsoidal dimpled tube with Al_2O_3 nanofluid 46% better than in the plain tube. Fig, 5 shows the Nu vs. Re for both the smooth tube and dimpled tube with different dimple geometry's. The results indicate that the three tubes are much different from each other, with the increase of Reynolds number. The computed results indicated that the Nusselt number for the spherical dimpled tube and ellipsoidal dimpled tube are 35.7% and 63.59% higher than that of the smooth tube. It is worth to note that those of the two dimpled tubes are much different from each other. In order to characterize the test tubes, Nusselt number of the two dimpled tubes and plain tube can be expressed as a function of Reynolds number. Nusselt number equations were taken from yuwang (2010),

$$Nu\ (\text{plain}) = 0.671 Re^{0.348} Pr^{1/3} \quad (1000 < Re < 5000) \qquad (8)$$

$$Nu\ (\text{spherical}) = 0.098 Re^{0.655} Pr^{1/3} \quad (1000 < Re < 5000) \qquad (9)$$

$$Nu\ (\text{ellipsoidal}) = 0.245 Re^{0.571} Pr^{1/3} \quad (1000 < Re < 5000) \qquad (10)$$

The enhanced structure for both the ellipsoidal and spherical dimple tubes could disturb, swirl, break the boundary layer developing and enhance the mixing of the hot and cold fluid and then improve the heat transfer of the tubes. It is interesting to note that the enhancements of the Nusselt number for the ellipsoidal dimpled tube are higher than that of the spherical dimpled tube at the different Reynolds number. The reason for the phenomenon can be attributed to the formation of longitudinal vortex which generated by ellipsoidal dimple.

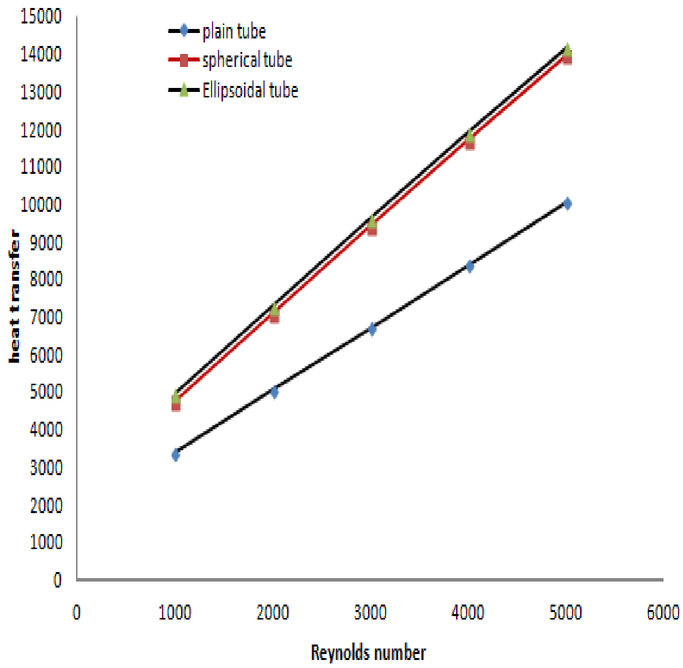


Fig. 4: Comparison of heat transfer as a function of Reynolds number for plain and different dimple geometry.

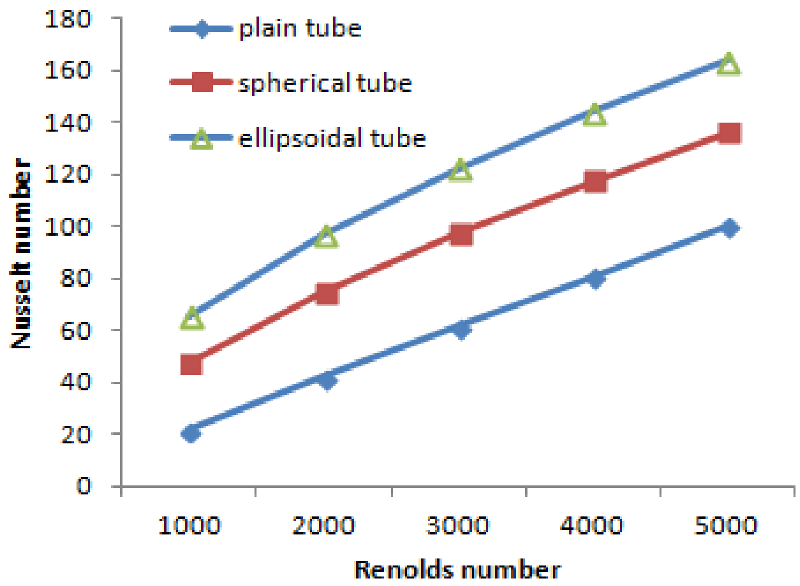


Fig. 5: Comparison of nusselt number as a function of Reynolds number for plain and different dimple geometry tubes.

b. Pressure Drops:

The pressure drop is an important parameter in the design of different tubes. Fig, 6 shows the friction factors vs. Reynolds numbers for both the dimpled tubes and smooth tube. It is clear that for all dimpled tubes, the friction factor varies with the increase of Reynolds Number in a manner similar to that of smooth tubes. Friction factor equations were taken from yuwang (2010).

The reason for the increase of pressure drop can be attributed to the eddy generated by the dimples, which disturbed the flow field inside the dimpled tube. It can also be seen that the friction factors of dimpled tubes are dependent on geometrical and the arrangement of dimples. Friction factors for the two dimpled tubes and plain tube can be expressed as functions of Reynolds number.

F (plain) $= 1/(2.772Re^{-0.293} - 11.12\ 7)$ (1000 < Re < 5000) (11)
F (spherical) = $0.432Re^{-0.221}$ (1000 < Re < 5000) (12)
F (ellipsoidal) = $0{:}326Re^{-0{:}227}$ (1000 < Re < 5000) (13)

Friction factors of dimpled tubes in parallel flow were significantly higher than those of the smooth tube. The friction factors in the dimpled tube increase by 52.7% and 87.27% for ellipsoidal and spherical dimples compared to the smooth tube. It is very interesting that the friction factor of the ellipsoidal dimpled tube is lower than that of spherical dimpled tube while it has better performance on heat transfer.

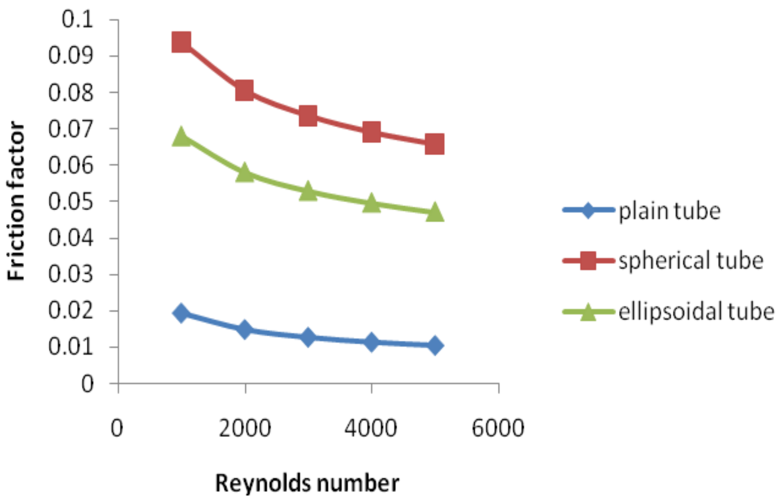


Fig. 6: Comparison of friction factor as a function of Reynolds number for plain and different dimple geometry tubes.

Conclusions:

In this paper, the heat transfer and pressure drop characteristics of dimpled tubes were measured analytically. The results can be summarized as follows:

(1) An analytical investigation for plain tube and both dimpled tubes was carried out. Dimpled tubes present a better performance than the smooth tube. The computed results indicated that the Nusselt number for the spherical dimpled tube and ellipsoidal dimpled tube are 35.7% and 63.59% higher than that for the smooth tube with Al_2O_3 nano fluid.

(2) The friction factors in the dimpled tube increase by 52.7% and 87.27% for ellipsoidal and spherical dimples compared with the smooth tube with Al_2O_3 nano fluid.

(3). Heat transfer rate of spherical dimpled tube with al_2o_3 nano fluid is 40 % times better than in the plain tube and ellipsoidal dimpled tube with Al_2O_3 Nanofluid is 46 % times better than in the plain tube with Al_2O_3 Nano fluid.

(4). The analytical results showed that the ellipsoidal dimpled tube has better performance on heat transfer than plain tube and spherical dimpled tube with Al_2O_3 Nanofluid. For heat exchanger application, ellipsoidal dimpled tubes can be used to improve the overall performance efficiency and reduce the size of the heat transfer system.

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