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# Numerical exploration of heat transfer in a heat exchanger tube with cone shape inserts and $Al_2O_3$ and CuO nanofluids

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ABSTRACT

Offering the small disturbance in the flow without affecting the pumping force is one of the passive techniques of augmenting the heat transfer in the double tube heat exchanger (DTHE). The flow disturbance is offered by the inserts like twisted tape with different ration of twist to make the swirl in the flow in the tube. The properties of fluid medium also a contributing more in the heat transfer. Objective of this research work is to augment the heat transfer rate in DTHE with help of cone shape inserts and Nanofluids. The cone shape insert create turbulence in the flow and the nano sized (<100 nm) powder which suspended on the fluid increase the thermal conductivity of the fluid. The $Al_2O_3$ and CuO Nano particles 1% of volume fraction, suspended in the water to improve the heat transfer rate in the effective of are used for this analysis and 3D model of cone and plain tube generated in solid works software and CFD Analysis done in ANSYS-Fluent Software, in this results $Al_2O_3$ Nano particles get the higher heat transfer rate in with and without cone inserted.



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## 1. Introduction

Need of Eco-friendly, In the recent world energy transformation from one system to another system is more important, to improve the heat transfer rate in the heat exchangers, heat exchangers are mainly used in petrochemical, boiler and power plant industries [1]. Self-rotating inserted into a conventional smooth tube is an innovative way to increase the thermal performance of the heat exchangers. The numerical simulations and experimental studies have been conducted to investigate the characteristics of heat transfer rate and friction factor of the tube fitted with self-rotating inserts [2]. Heat transfer can be observed with plain smooth tube records were enhanced from 7% to 10% of the twisted tube heat transfer of fitted in the concentric circular tube. The data obtained from the experimental value of twisted tape friction factor was decreased 2% to 6% as compared to the plain tube [3]. The enhancement of tube side experimental Reynolds's number, heat transfer coefficient and heat transfer were found to be 120,180,240,300 LPH flow rate at 0.1% particle volume concentration respectively.

This is due to the better fluid mixing and higher effective thermal conductivity of Nano fluid. This secondary flow provides proper mixing to enhance heat transfer. Results enhanced heat transfer coefficient. It is observed the increasing trend of experimental Nusselt number for 0.1% particle volume concentration of Nano fluid [4]. Heat transfer is more in case of counter flow heat exchanger than parallel flow heat exchanger using water as a working medium. Heat transfer enhancement in double pipe heat exchanger increased as compared to tube without any insert or without extended surfaces. Friction factor and heat transfer coefficient is increased with the decreased of twist ratio of inserted tape. The provision of baffles in double pipe heat exchanger increases the pressure drop and hence thereby increasing the pumping power. This limitation is overcome by using fins, full length twisted tape, dimples etc. [5] the suspension of small size nanoparticles (Ag and $Al_2O_3$) in water-based Nanofluids has greater viscosity and thermal conductivity. It is found that there is certain limit of $\phi$ where size of nanoparticles start affecting. The size of Ag nanoparticles is affected only at small volume fraction ($\phi$) 0.01 as compared to $Al_2O_3$ nanoparticles; size has no effect at this volume

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**Nomenclature**

| | | | |
|---|---|---|---|
| $Al_2O_3$ | Alumina | Ag | Silver |
| CuO | Copper oxide | Nu | Nusselt number |
| $TiO_2$ | Titania | Re | Reynolds number |
| LPH | Litters per Hour | Pr | Prandtl number |
| DTHE | Double Tube Heat Exchanger | | |

fraction on velocity of Nano fluid but at high volume fraction 0.1, velocity of Nanofluids start affecting with increasing size of $Al_2O_3$ nanoparticles. On the other side for EG-based Nano fluids the size has no effect [6]. Pressure drop and constant wall heat flux heat transfer measurements in a 75-start spirally grooved tube with and without twisted tapes and with water as working fluid (Pr′ 5.4) have yielded highly non-linear behaviour of f and Nu with Reynolds number and twist ratio Y [7]. An experimental set-up was designed and fabricated to investigate the heat transfer performance of Nanofluids for a 4 mm thick steel plate with an initial temperature of 7000 °C. Multiple jet cooling characteristics of Nanofluids with 0.01, 0.03, 0.05 and 0.07% weight concentrations of alumina ($Al_2O_3$) and titania ($TiO_2$) nanoparticles were extensively examined for DI water with three different nozzle tip to plate distances of 120, 180 and 240 mm, respectively [8]. Recommended 1% concentration of $Al_2O_3$ nanoparticles with base fluid for minimum pumping loss.

## 2. Materials and methods

### 2.1. Mathematical modelling

Single phase model in which both the fluid phase and the particles are in thermal equilibrium and flow with the same local velocity is considered. The three dimensional model are considered to describe the turbulent flow and heat transfer behaviour of Nanofluids in a horizontal circular pipe under uniform heat flux boundary condition, whereas a multi-phase model is carried out in Saha and Paul. Computational geometry consists of a pipe with length L of 1.5 m and a circular section with inner diameter (Di), of 0.058 m.

### 2.2. Governing equations

The dimensional steady-state governing equations of fluid flow and heat transfer for the single phase model have been presented and the following assumptions are considered:

i. Fluid flow is incompressible, Newtonian and turbulent,
ii. The Bossiness approximation is negligible as the pipe is placed horizontally,
iii. Fluid phase and nanoparticles phase are in thermal equilibrium and no-slip between them and they flow with the same local velocity,
iv. Nanoparticles are spherical and uniform in size and shape,
v. Radiation effects and viscous dissipation are negligible. Under the above assumptions, the dimensional steady state governing equations for the fluid flow and heat transfer for the single phase model can be expressed as

Continuity equation:

$$\frac{\partial_{V_x}}{\partial x}+\frac{\partial_{V_r}}{\partial r}+\frac{V_r}{r}=0$$

x-Momentum equation:

$$\frac{1}{r}\frac{\partial}{\partial x}(r v_x v_x)+\frac{1}{r}\frac{\partial}{\partial x}(r v_r v_x)+\frac{\mu}{\rho}\frac{\partial}{\partial r}\left[\left(\frac{\partial v_x}{\partial r}+\frac{\partial v_r}{\partial x}\right)\right]$$
$$=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\frac{\partial}{\partial x}\left[\left(2\frac{\partial v_x}{\partial x}-\frac{2}{3}\left(\nabla . V\overrightarrow{V v}\right)\right)\right]$$

r-Momentum equation:

$$\frac{1}{r}\frac{\partial}{\partial x}(r v_x v_r)+\frac{1}{r}\frac{\partial}{\partial x}(r v_r v_r)=-\frac{1}{\rho}\frac{\partial p}{\partial r}+\frac{\mu}{\rho}\frac{\partial}{\partial x}\left[\left(\frac{\partial v_r}{\partial x}+\frac{\partial v_x}{\partial r}\right)\right]+\frac{\mu}{\rho}$$
$$\times\frac{\partial}{\partial r}\left[\left(2\frac{\partial v_r}{\partial r}-\frac{2}{3}\left(\nabla .\overrightarrow{v}\right)\right)\right]-2\frac{\mu}{\rho}\frac{v_r}{r^2}$$
$$+\frac{2}{3}\frac{1}{r}\frac{\mu}{\rho}\left(\nabla .\overrightarrow{v}\right)$$

where $\left(\nabla .\overrightarrow{v}\right)=\frac{\partial v_x}{\partial x}+\frac{\partial v_r}{\partial r}+\frac{v_r}{r}$

Energy equation:

$$\frac{\partial(v_v T)}{\partial x}+\frac{\partial(v_r T)}{\partial r}=\frac{1}{\rho}\left(\frac{\partial}{\partial x}\left(\Gamma\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial r}(\Gamma\frac{\partial T}{\partial r})\right)$$

where x and r are the axial and radial coordinates respectively, vx and vr are the respective axial and radial velocity, T is the temperature, Γ is the exchange coefficient for general transport, ρ is the density, p is the pressure and μ is the dynamic viscosity of Nanofluid. For turbulent flow regime, both the terms Γ and μ are replaced by their effective values and defined as

$$\mu_{\text{eff}}=\mu+\mu_{\text{t}}$$

$$\Gamma_{\text{eff}}=\frac{\mu}{pr}+\frac{\mu_t}{\sigma_t}$$

Respectively, where $\mu_t$ is the turbulent molecular viscosity, $\sigma_t$ is the constant of turbulent Prandtl number and $P_r$ is the Prandtl number of Nanofluid. It is expected that heat transfer coefficient (Nusselt number) of the Nano fluid depends upon a number of factors such as thermal conductivity and heat capacity of both the base fluid and the ultrafine particles, the flow pattern, the viscosity of the Nanofluid, the volume fraction of the suspended particles, the dimensions and the shape of these particles as well as on the flow structure. Therefore, the general form of the Nusselt number yields $Nu_{nf}$ = f (Re, Pr, ks /kf, $(\rho Cp)_p$, ϕ, dimensions and shape of particles, flow structure).

Three main parameters involved in calculating heat transfer rate of the Nano fluid are heat capacity, viscosity, and thermal conductivity, which may be quite different from those of the original pure fluid. The average Nusselt number and friction factor basis of inner diameter of tube can be expressed as below:

$$\text{Nu}=\frac{h\ dh}{k}$$

$$\text{f}=\frac{\Delta P}{1\ /2\ \rho u2\ \frac{L}{dh}}$$

The convective heat transfer coefficient which is defined as:

$$h = \frac{Qave}{Ai(Tw - Tb)}$$

where Tw is the average wall temperature that estimates from the attached thermocouples over inner tube and Tb is average (bulk) temperature of fluid. The average heat flux is calculated as:

$$Q_c = m_c Cp_c\,(T_{c,\ out} - T_{c,\ in}).$$

where $m_c$ is the mass flow rate of cold fluid; $Cp_c$ is the specific heat of cold fluid; $T_{c,in}$ and $T_{c,out}$ are the inlet and outlet cold fluid temperatures, respectively. $Q_h$ is the heat transferred from hot fluid in outer tube that is expressed as:

$$Q_h = m_h Cp_h\,(T_{h,\ out} - T_{h,\ in}).$$

where $m_h$ is the hot water mass flow rate; $T_{h,in}$ and $T_{h,out}$ are the inlet and outlet hot fluid temperatures, respectively. The thermal and physical properties of Nanofluids such as density, viscosity, specific heat thermal conductivity of the Nanofluid were calculated as given below. The density of Nanofluid is evaluated using the general formula for the mixture:

The density of the Nanofluid is defined as:

$$\rho_{nf} = \varphi\rho_{np} + (1-\varphi)\rho_f$$

Einstein equation for the effective viscosity of a fluid in volume concentrations less than 5% is given by:

$$\mu_{nf} = \mu_f(1 + 2.5\varphi)$$

From Xuan and Roetzel's equation for the specific heat of Nanofluid is given as:

$$(\rho Cp)_{nf} = (1-\varphi)(\rho Cp)_f + \varphi(\rho Cp)_{np}$$

The thermal conductivity of the Nanofluid is defined as:

$$\frac{k_{nf}}{k_f} = \frac{k_p + (n-1)k_f - (n-1)(k_f - k_p)\varphi}{k_p + (n-1)k_f + (k_f - k_p)\varphi}$$

where the empirical shape factor given by $n = 3/\Psi$. where "Ψ" sphericity is the, defined as the ratio of the surface area of a sphere with a volume equal to that of the particle to the surface area of the particle.

Thermal performance factor is defined as follows:

$$\eta = \frac{(Nu_t/Nu_p)}{(f_t/f_p)^{1/3}}$$

### 2.3. Computational domain and meshes

The problem under consideration consists of steady, forced turbulent convection flows and heat transfer of a Nanofluid flowing inside a circular pipe with conical insert. The computational model geometry and the grid were generated using SOLIDWORKS, the pre-processing module for the FLUENT software. The CFD domain consists of a circular pipe with length L of 1500 mm with diameter D is 50.8 mm. The pipe has appropriate length in order to obtain fully developed velocity and temperature profiles at the outlet. The condition of the constant wall heat flux is considered in this study. The nomenclature of the twisted tape and its 3D model and insert placed on the tube are depicted in the Fig. 1, Fig. 2 and the Fig. 3 respectively.

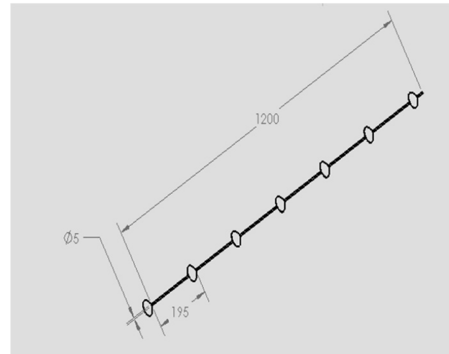


**Fig. 1.** Cone inserts profile.

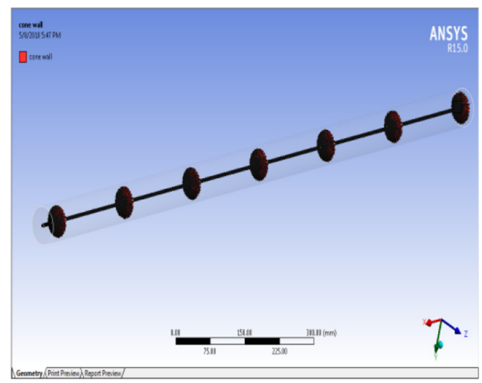


**Fig. 2.** Mesh image of Tube with insert.

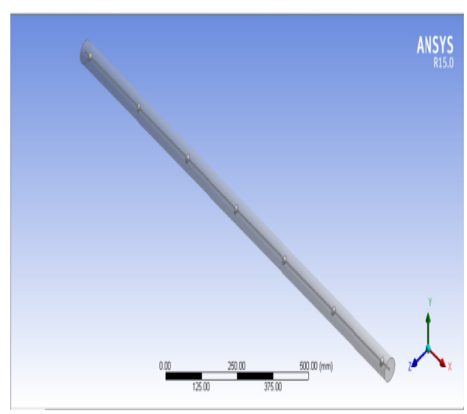


**Fig. 3.** Insert in tube assembly.

### 2.4. Mesh generation sensitivity and turbulence model

The accuracy of finite volume method is directly related to the quality of the discretization used. In this study, structured hexahedral meshes are used which are known to provide higher accuracy and reduce the CFD computational effort. A comprehensive mesh sensitivity study was done to check on the influence of the mesh resolution on the results and to minimize numerical influences introduced by the size of meshes and their distributions. For mesh sensitivity analysis, four meshes of differing size were used ranging from 0.32 to 0.65 million. The mesh with 0.65 appears to be satisfactory to ensure the accuracy of numerical results as well as their independency with respect to the number of nodes used.

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**Table 1**
Properties of HT fluids.

| Material Properties | Density (kg/m$^3$) | Specific Heat (j/kg.K) | Thermal conductivity (w/m-K) | Viscosity (kg/m-s) |
|---|---|---|---|---|
| $Al_2O_3$ | 3970 | 765 | 40 | 0.001254 |
| CuO | 6320 | 531 | 32.9 | 0.001254 |
| Water | 998 | 4179 | 0.605 | 0.001003 |

*2.4.1. Boundary conditions*

The fluid enters with uniform temperature of $T_0 = 300$ K and velocity $V_0$ at the inlet. Different inlet uniform velocities are applied. In order to validate the CFD model Reynolds number and thermal boundary condition were chosen to match the available correlations. At the outlet of the computational model a relative average pressure equalling zero was defined. A constant heat of h = 600 W/K m$^2$ is specified for the wall (wall surface).

*2.4.2. Numerical method*

The modeled cases were solved using ANSYS FLUENT software. A segregated, implicit solver option was used to solve the governing equations. The first order upwind discrimination scheme was employed for the terms in energy, momentum, and turbulence parameters. A second order pressure interpolation scheme and COUPLED pressure-velocity coupling were implemented. A residual root-mean-square (RMS) target value of $10^{-5}$ ($10^{-8}$ for energy equation) was defined for the CFD simulations.

*2.5. Thermo physical properties of the nano fluids*

The single-phase approach is chosen to calculate the thermo physical properties of Nano fluids as it is widely used in the literature [9–12]. In this model the homogenous mixture is assumed prior to solving the governing equations of continuity, momentum, and energy for the single phase fluid flow that the presence of nanoparticles is realized by modifying physical properties of the mixture fluid. It is assumed that there is no velocity difference between fluids and the particles, and the fluids and the particles are in thermal equilibrium [9–11].

This assumption implies that all the convective heat transfer correlations available in the literature for single-phase flows can be extended to nanoparticle suspensions, providing that the thermo physical properties appearing in them are the Nano fluid effective properties calculated at the reference temperature. Note that most Nano fluids used in practical applications usually comprise the oxide particles finer than 40 nm. In the current CFD study the considered Nano fluid is a mixture of water and particles of $Al_2O_3$ and CuO (Table 1).

## 3. Experimentation

The numerical experimentations were carried out with various the flow conditions individually for each heat transfer fluid like plain water, alumina water nanofluid and copper oxide nanofluid. The Fig. 4 illustrates the observations.

The Fig. 4 illustrates the observations of pressure counters when experimenting with plain water at various flow conditions. The Fig. 5 depicts the temperature counters when experimenting with plain water at various flow conditions. The Fig. 6 shows the Turbulence kinetic Energy profile when experimenting with plain

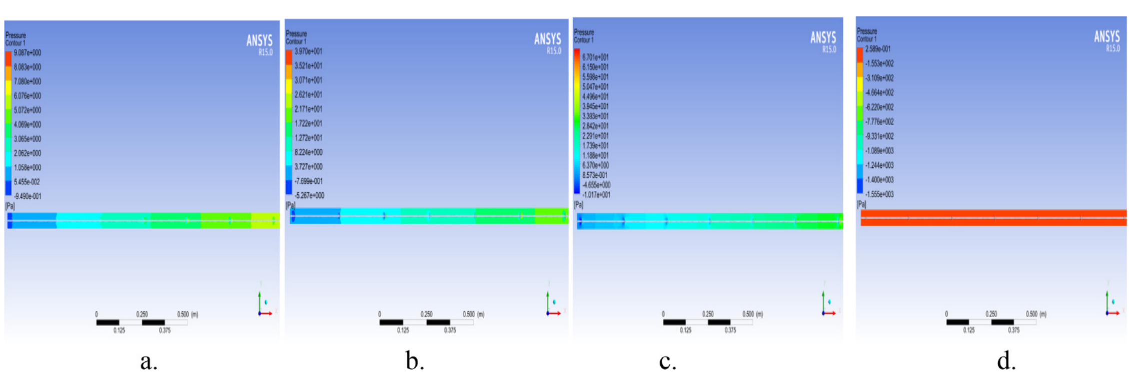

**Fig. 4.** Pressure Contour with Plain Water at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

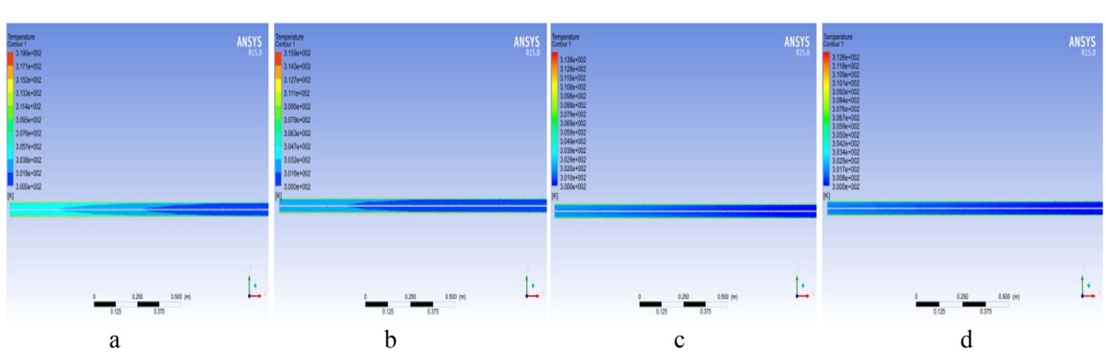

**Fig. 5.** Temperature gradient with plain water at Re of (a) 2000 (b) 5000 (c) 7000 (d) 10,000.

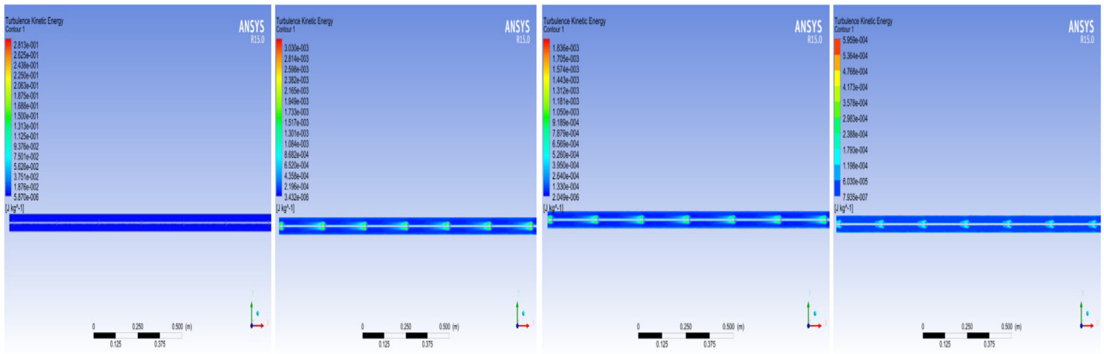

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**Fig. 6.** Turbulence kinetic Energy with plain water at Re of (a) 2000 (b) 5000 (c) 7000 (d) 10,000.

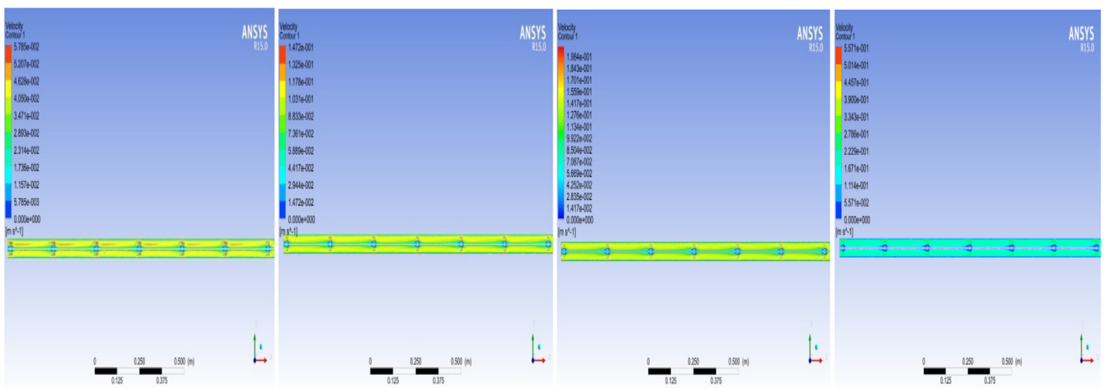

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**Fig. 7.** Velocity contour with plain water at Re of (a) 2000 (b) 5000 (c) 7000 (d) 10,000.

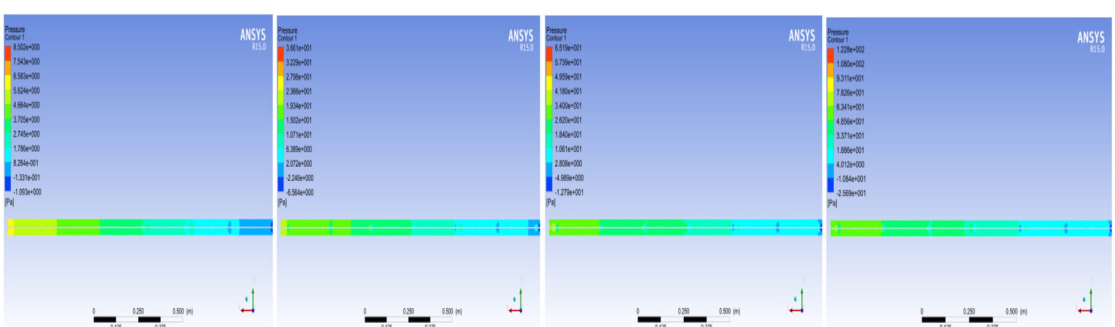

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**Fig. 8.** Pressure Contour with $Al_2O_3$-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

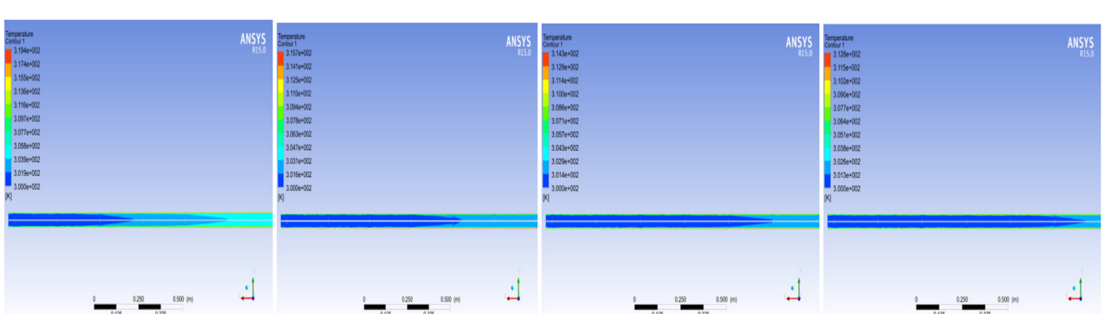

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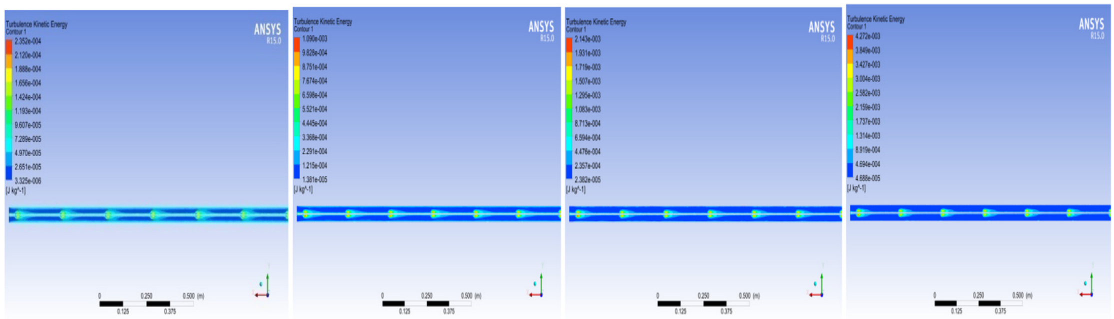
**Fig. 9.** Temperature Contour with $Al_2O_3$-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.



a b c d

**Fig.10.** Turbulence KE Contour with $Al_2O_3$-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

water at various flow conditions and the Fig. 7 demonstrates the velocity contours when experimenting with plain water at various flow conditions.

The setup is now simulated with $Al_2O_3$-Water nanofluid and observations like pressure counters, temperature counters, Turbulence kinetic Energy profile and velocity counters were observed. Similarly the Fig. 8 illustrates the observations of pressure counters when experimenting with $Al_2O_3$-Water nanofluid at various flow conditions. The Fig. 9 depicts the temperature counters when experimenting with $Al_2O_3$-Water nanofluid at various flow conditions. The Fig. 10 shows the Turbulence kinetic Energy profile when experimenting with $Al_2O_3$-Water nanofluid at various flow conditions and the Fig. 11 demonstrates the velocity contours when experimenting with $Al_2O_3$-Water nanofluid at various flow conditions.

Finally the model was simulated with CuO-Water nanofluid and observations like pressure counters, temperature counters, Turbulence kinetic Energy profile and velocity counters were observed. The Fig. 12 illustrates the observations of pressure counters when experimenting with CuO-Water nanofluid at various flow conditions. The Fig. 13 depicts the temperature counters when experimenting with CuO-Water nanofluid at various flow conditions. The Fig. 14 shows the Turbulence kinetic Energy profile when experimenting with CuO-Water nanofluid at various flow conditions and the Fig. 15 demonstrates the velocity contours when experimenting with CuO-Water nanofluid at various flow conditions.

## 4. Results and discussion

The heat transfer of a Nano fluid is expected to depend, apart from the flow configuration, on a number of material factors, such as thermal conductivity and heat capacitance of pure fluid and particles, volume fraction, viscosity, etc.

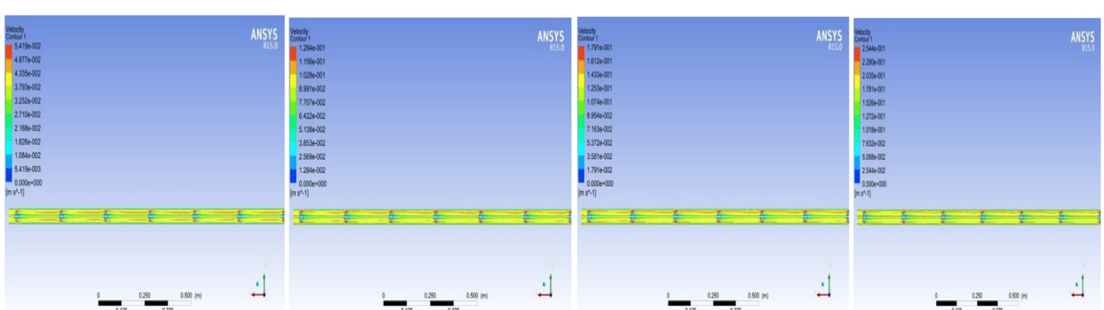


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**Fig. 11.** Turbulence Velocity Contour with $Al_2O_3$-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

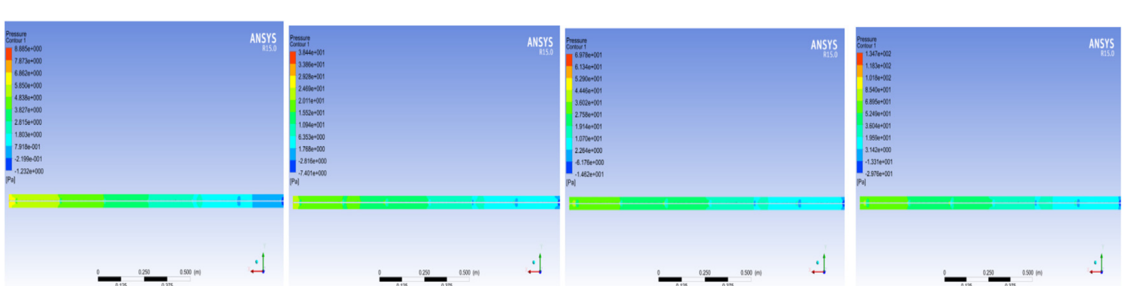

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**Fig. 12.** Pressure Contour with CuO-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

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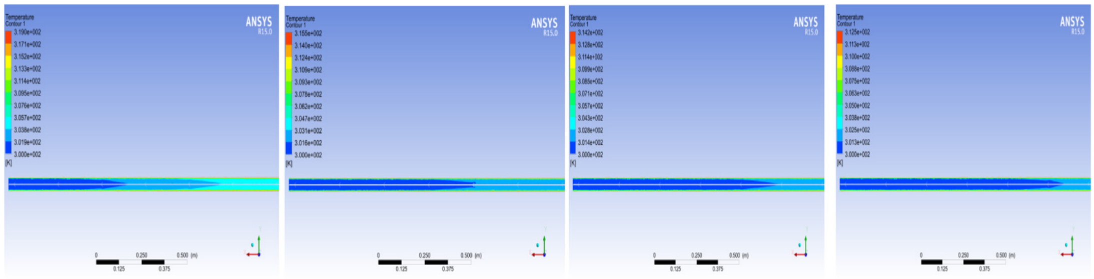


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**Fig. 13.** Temperature Contour with CuO-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

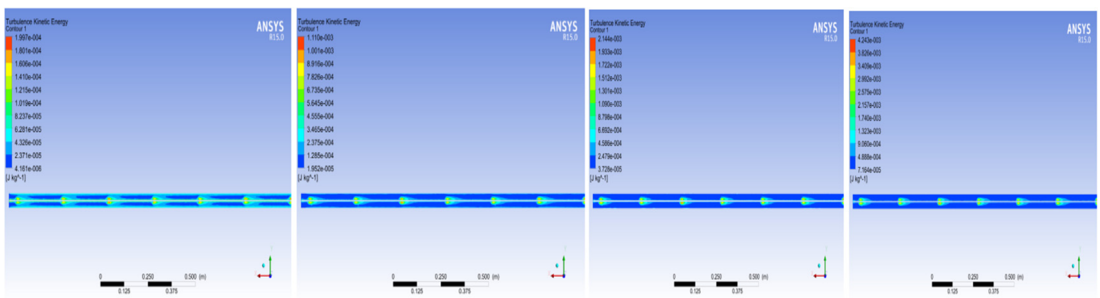


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**Fig. 14.** Turbulence KE Contour with CuO-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.

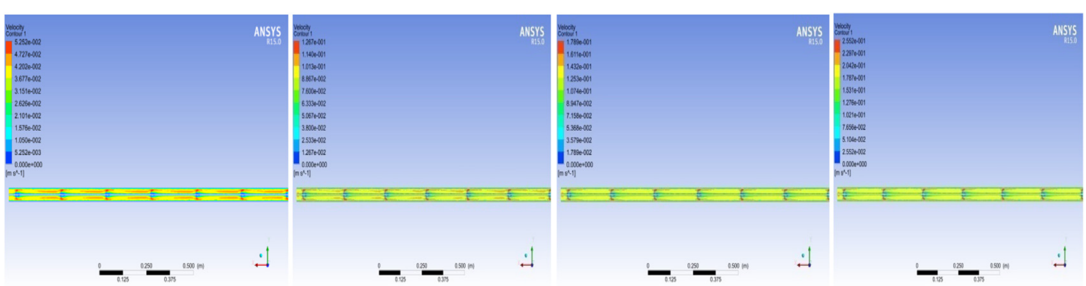


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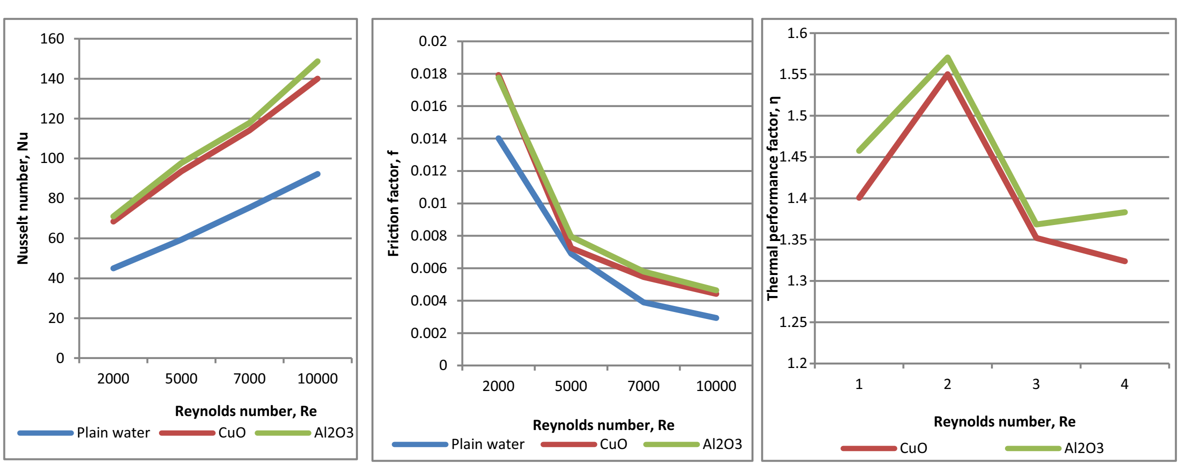
**Fig. 15.** Turbulence Velocity Contour with CuO-Water nanofluid at Re (a) 2000 (b) 5000 (c) 7000 and (d) 10,000.




a b c

**Fig. 16.** (a) Thermal performance (b) friction factor and (c) Thermal Performance factor.

*4.1. Thermal performance*

Fig. 16a represents the variation of Nusselt number with Reynolds number for tube equipped with conical insert. The Nusselt number increases with the increase in the Reynolds number. This leads to increase of turbulent intensity with increase of Reynolds number, leading to the amplification of heat transfer in the pipe. The heat transfer is highest with $Al_2O_3$ and CuO Nano fluid in comparison with plain water. This shows that the heat transfer can be effectively improved by using of Nano particles in water as the base fluid.

*4.2. Thermal performance and Reynolds's number*

As seen Fig. 16b, the thermal performance factor tends to decrease with the rise of Reynolds number, for all cases. The thermal performance factors of Nano fluids are consistently larger than unity and significantly higher with Nano fluids than plain water. This reveals the advantage of using Nano fluids over the plain water, in viewpoint of energy saving.

*4.3. Friction factor and Reynolds's number*

As seen Fig. 16c, the friction factor tends to decreases with rise of Reynolds's number for all the cases. Friction factor for water is comparatively lesser than the Nano fluids ($Al_2O_3$ and CuO). So, Nano fluids heat exchanger tubes require higher pump power. The aluminum oxide has a lesser friction factor than other Nano fluids.

## 5. Conclusion

Aluminum oxide $Al_2O_3$ and Copper oxide CuO Nano fluids into a cone shaped inserted heat exchanger tube is a smart way to improves heat transfer rate. The mathematical and numerical simulation was carried out by with the help of modeling software Solid Works and simulation software ANSYSFluent.

1. The heat transfer rate of tube enhances by introducing cone shaped inserts into a heat exchanger tube; this enhancement achieved by the presence of turbulence intensity of flow in tube.
2. Aluminum oxide ($Al_2O_3$) nano fluid gives the higher heat transfer rate than the copper Oxide nano fluid because of the nano layer and thermo physical properties of liquid and nanosolid particles: this nano layer acts as thermal bridge between liquid and solid phase. Strength of the thermal bridge is significantly higher in $Al_2O_3$ nano fluid. So this a key to enhances the thermal conductivity of $Al_2O_3$.
3. Pressure drop increase along the tube because of cone shaped insert and nano sized solid particles in presence of nano fluids; these nano sized solid particles are contacts with wall to produces friction .so, this will leads to increase pressure drop along the tube wall. The main objective for the researchers to achieve lowest pressure drop along the wall of the tube.

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