International Journal of Science and Research (IJSR)
ISSN: 2319-7064
Index Copernicus Value (2016): 79.57 | Impact Factor (2017): 7.296

Heat Transfer and Flow Behavior Analysis of Double Pipe Heat Exchanger

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Abstract: ***Various researchers around the world are working upon the enhancement of effectiveness through the identification of various working fluids. In addition, micro-heat exchangers developed till date employ twisted tape inserts, baffles and fins to increase the surface area of the heat exchangers still keeping the effective dimensions of the heat exchangers to be low. In the present work, a double pipe heat exchanger with corrugated pipes is proposed to enhance the thermohydraulic performance of the heat exchangers. Investigation is completed in the present work to find the pressure drop and heat transfer in the double pipe heat exchanger (DPHE) using Computational Fluid Dynamics (CFD). Mass flow rates considered for the analysis are according to the practical conditions employed in industrial refrigeration systems. Further, due to the corrugations in the pipes, the increase in pressure drop expected. This pressure drop in the DPHE would contribute to the performance of the compressor in the refrigeration system thereby reducing the Coefficient of Performance (COP) of the system. Friction factors and Nusselt Numbers applicable to corrugated Double Pipe Heat Exchangers (DPHE) are investigated.***

Keywords: Double Pipe Heat Exchangers, Heat Transfer, Thermo-hydraulic, Computational Fluid Dynamics, Coefficient of Performance

1. Introduction

Heat exchangers are being employed in heat transfer applications like food and chemical processing industries, manufacturing industries, refrigeration and air conditioning industries, space applications etc. To enhance the heat transfer rates diverse methods have discovered [1-8] till date which includes active and passive methods. From the survey, it has been noticed that there are numerous studies available in which methods have been adopted to increase the heat transfer rates in circular double pipe heat exchangers. Very few studies are available in literature where corrugated geometry has been employed in order to investigate the heat transfer rates. Few studies involved nano-fluids those have been used to increase the rate of heat transfer [9-12].

As no work has been reported where the corrugation geometry (inner as well as outer pipe) has been considered in the development of double pipe heat exchangers, therefore in the present study an attempt has been made in order to investigate the effect of corrugation on the velocities and temperature profile of the fluid flow inside double heat exchanger. The prime objective of the present study is to investigate the flow behavior such as pressure drop and friction factor in outer corrugated pipe along with the heat transfer between fluids. It is necessary to investigate heat transfer through both the fluids for further calculations of heat exchanger thermal efficiency. A comparison between the heat transfer and flow characteristics have been made for a corrugated and smooth pipe flow.

2. Computational Analysis

2.1 Modeling of double pipe heat exchanger with corrugation

The corrugated fluid domain for inner and outer flow in DPHE is generated in ANSYS module. The specifications of these fluid domains are enlisted in Table **1**. Inner corrugated fluid domain represents the flow of hot water where it enters from the left hand side and exiting at the right side as shown in Fig-1. The outer corrugated fluid domain indicates the flow of cold water which enters from the right side and exiting from the left side.

Table 1: Specifications of the DPHE fluid domain

Length *(mm)*	Inner pipe Dia*(mm)*	Outer pipe Dia*(mm)*	Corrugation Pitch *(mm)*	Corrugation Depth *(mm)*
518	32	46	10	1

2.2 Fluid Properties

In the FLUENT interface, first of all there is a need to define the fluid properties for the inner and outer corrugated fluid domain of DPHE. As in the inner domain hot water at 350K is entering therefore the properties at 350K of water are inserted in the interface. In case of outer fluid domain, the properties of water at 293K are inserted in the interface.

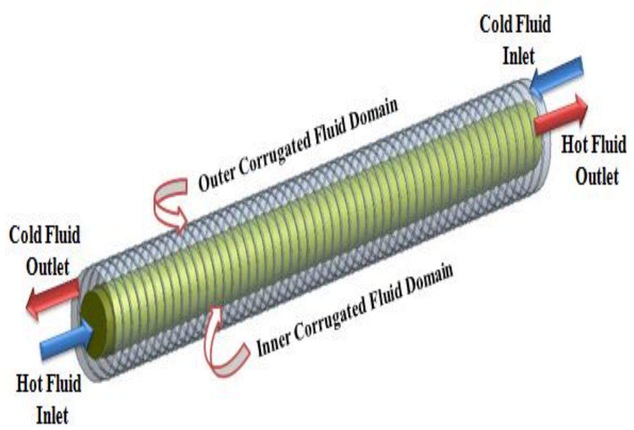

Figure 1: Corrugated Fluid domains of hot and cold water

Figure 2: Meshed Geometry with Medium Size

2.3 Mathematical Models

In order to perform Thermohydraulic Analysis of Double Pipe Heat Exchanger with Corrugated Tubes, generally five equations are need to be solve and these are continuity equation, x-momentum, y-momentum, z-momentum equations and energy equation. These equations can be written in the mathematical form on the page:

In order to investigate heat transfer and flow analysis in corrugated DPHE, these equations is required to solve. To solve these equations, Realizable 2 equation k-epsilon viscous model has been used.

2.4 Boundary Conditions

The boundary conditions those are used in this analysis are available in Table 2. The operating pressure is 700kPa and for pressure outlet boundary condition, the gauge pressure is 0Pa.

3. Results and Discussions

Equation 1 to Equation 5 represents the mathematical expressions those have been used in order to evaluate friction factor, Reynold's number, pressure drop, pumping power and Nusselt number. Using the post- processing module of the FLUENT, the shear stresses (τ_{wall}) at the walls of the corrugated steel pipe are retrieved. These values are used in the calculation of friction factors at different flow rates for each type of the corrugated steel pipe using the Equation 1.

$$f = \frac{8\tau_{wall}}{\rho v_{avg}^{2}} \quad (1)$$

Reynolds Number can be estimated as

$$\mathrm{Re} = \frac{D_h V_{avg} \rho}{\mu} \quad (2)$$

Where V_{avg} = Average velocity of flow in the pipe (*m/s*) and ρ =Density (*kg/m*^{3}). Friction factors values calculated from Equation 3 are used to estimate the pressure drop at different flow rates for each of the case using

$$\Delta p = \frac{fLv^2}{2D_h\rho} \quad (3)$$

Where Δp =Pressure Drop (*Pa*), f =Friction Factor, L = Length of the HTS cable (*m*), D_h = Hydraulic Diameter (*m*). The pumping power needed to pump the water through the corrugated steel pipe of HTS cable is calculated using the pressure drop values estimated from Equation 4.

$$W = \Delta p \dot{V} \quad (4)$$

Where $V = A \times v_{avg}$ is volume flow rate (*L/min*) and A is the area of cross-section of the pipe (*m*^{2}).
Now, Nusselt number, a dimensionless form of the temperature gradient in the fluid at the heat transfer surface, is calculated from Equation 5.

$$Nu = \frac{h_c L}{k_t} \quad (5)$$

Where, h_c = Heat Transfer coefficient (*W/m*^{2}*-K*), k_t = thermal conductivity (*W/m-K*). The unknown value, the heat transfer coefficient is estimated from Equation 6 at different flow rates for each corrugation pitch of fixed depth

$$q'' = h_c \Delta T \quad (6)$$

Where, q'' is the heat flux (*W/m*^{2}), $\Delta T = T_w - T_{bulk}$ is the temperature difference (*K*), T_w = wall temperature and T_{bulk} = bulk temperature of the fluid. The different values of wall temperatures are obtained from the numerical solution.

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0$$

Momentum Equation

$$\rho\frac{\partial u}{\partial t} + \rho u\frac{\partial u}{\partial x} + \rho v\frac{\partial u}{\partial y} + \rho w\frac{\partial u}{\partial z} = -\frac{\partial \hat{p}}{\partial x} + \mu\left[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right]$$

$$\rho\frac{\partial v}{\partial t} + \rho u\frac{\partial v}{\partial x} + \rho v\frac{\partial v}{\partial y} + \rho w\frac{\partial v}{\partial z} = -\frac{\partial \hat{p}}{\partial y} + \mu\left[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right]$$

$$\underbrace{\rho\frac{\partial w}{\partial t}}_{\text{Local Acceleration}}+\underbrace{\rho u\frac{\partial w}{\partial x}+\rho v\frac{\partial w}{\partial y}+\rho w\frac{\partial w}{\partial z}}_{\text{Convection}}=\underbrace{-\frac{\partial p}{\partial z}}_{\text{Pressure gradient}}+\underbrace{\mu\left[\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right]}_{\text{Viscous terms}}$$

Energy Equation

$$\frac{\partial(E_t)}{\partial t}+\frac{\partial(uE_t)}{\partial x}+\frac{\partial(vE_t)}{\partial y}+\frac{\partial(wE_t)}{\partial z}=-\frac{\partial(up)}{\partial x}-\frac{\partial(vp)}{\partial y}-\frac{\partial(wp)}{\partial z}-\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\left[\frac{\partial q_x}{\partial x}-\frac{\partial q_y}{\partial y}-\frac{\partial q_z}{\partial z}\right]$$

$$+\frac{1}{\mathrm{Re}}\left[\frac{\partial}{\partial x}\left(u\tau_{xx}+v\tau_{xy}+w\tau_{xz}\right)+\frac{\partial}{\partial y}\left(u\tau_{xy}+v\tau_{yy}+w\tau_{yz}\right)+\frac{\partial}{\partial z}\left(u\tau_{xz}+v\tau_{yz}+w\tau_{zz}\right)\right]$$

Table 2: Different inlet and outlet boundary conditions for hot and cold fluid

Water	Inlet Mass Flow Rate (kg/s)					Temperature (K)	Outlet
Hot	0.05	0.06	0.07	0.08	0.09	350	Pressure Outlet
Cold	0.03	0.035	0.045	0.055	0.065	293	Pressure Outlet

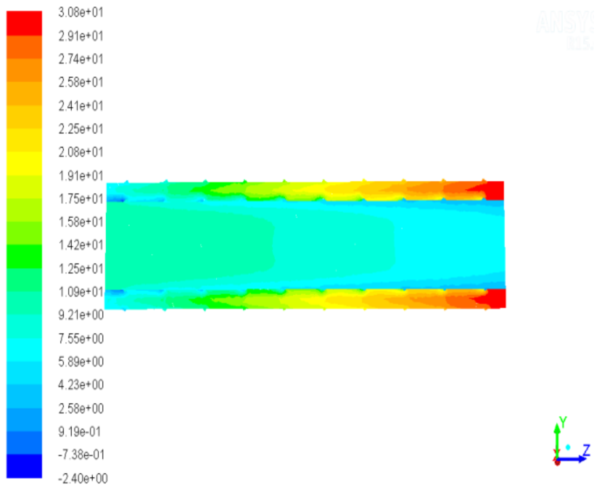

Figure 3: Contours plot of total pressure in DPHE

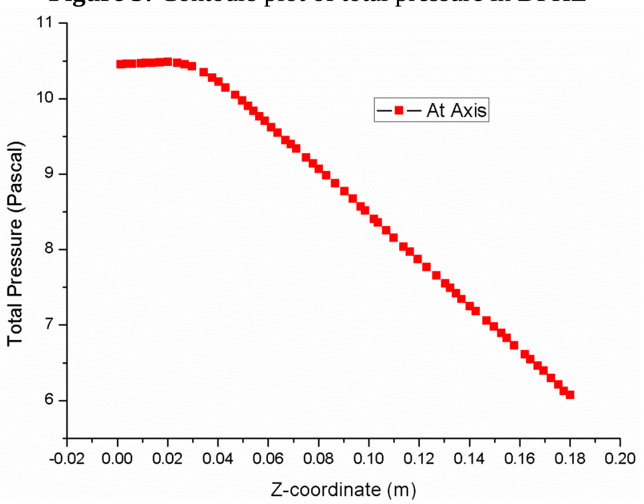

Figure 4: Pressure drop plot at central axis in hot water pipe

3.1 Pressure Drop and pumping power Analysis

In the present study, pressure drop analysis has been done in order to evaluate the turbulent flow effects when hot water is flowing in the inner corrugated pipe of double pipe heat exchanger. Fig-3 shows the contour plot for total pressure in DPHE.

Fig-4 shows the pressure drop along the axis of the double pipe heat exchanger. It can be noticed that for a length of 518mm there is a 4-5Pa pressure drop. Fig-5 and Fig-6 shows pumping power required to pump the hot and cold fluid. It can be noticed that the pumping power found to increase with the increase in mass flow rate for both hot and cold fluid. Also, the magnitude of pumping power is found to be higher for inner and outer corrugated pipe flow than the flow through a smooth pipe.

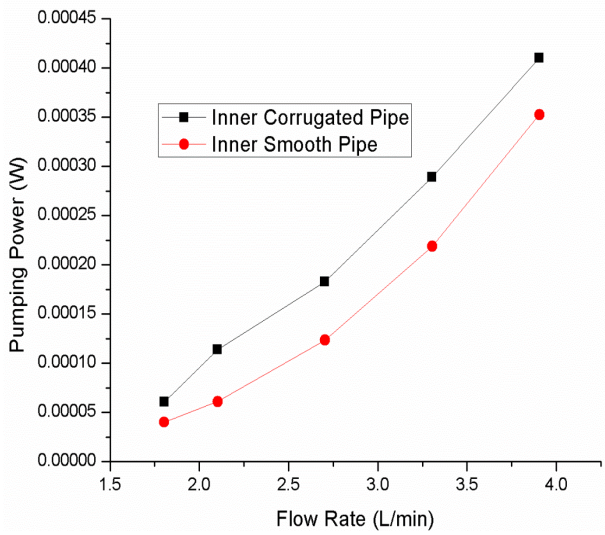

Figure 5: Pumping power vs. Mass flow rate for hot fluid

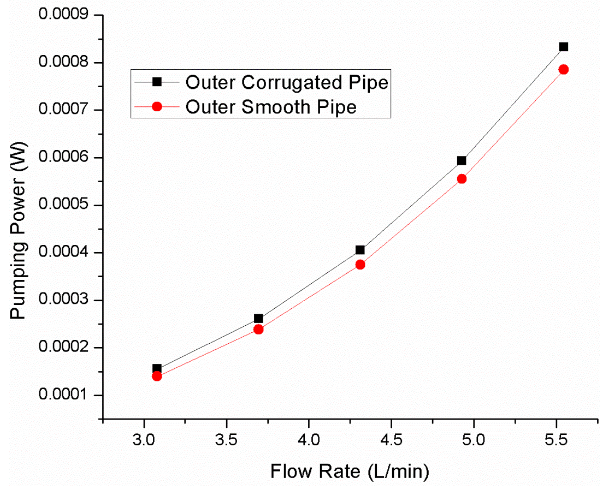

Figure 6: Pumping power vs. Mass flow rate for cold fluid

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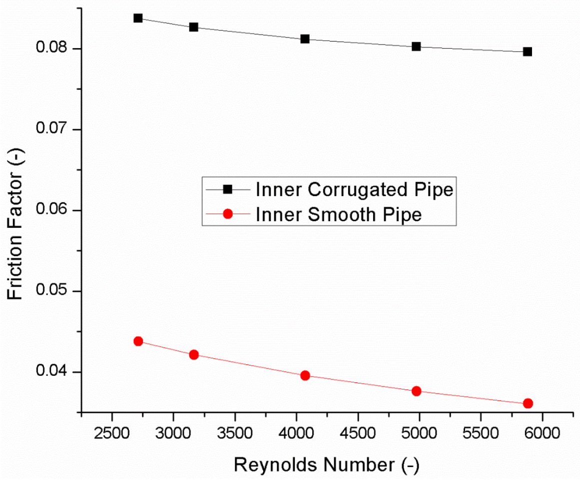

Figure 7: Friction factor vs. Reynolds Number variation for inner corrugated pipe

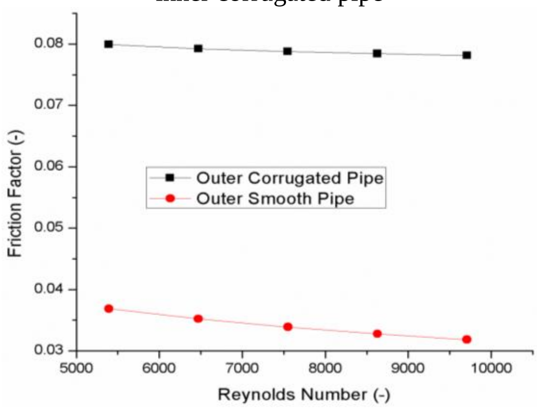

Figure 8: Friction factor vs. Reynolds Number variation for outer corrugated pipe

From Fig-7 and Fig-8, it can be noticed that with the increase in the Reynolds number the friction factor is found to be decrease for both hot and cold fluid flow through the smooth and corrugated pipe. This implies that the pressure drop will reduce with increase in the Reynolds number. Also, the friction factor for inner and outer corrugated pipe is found to be higher than the smooth pipe flow.

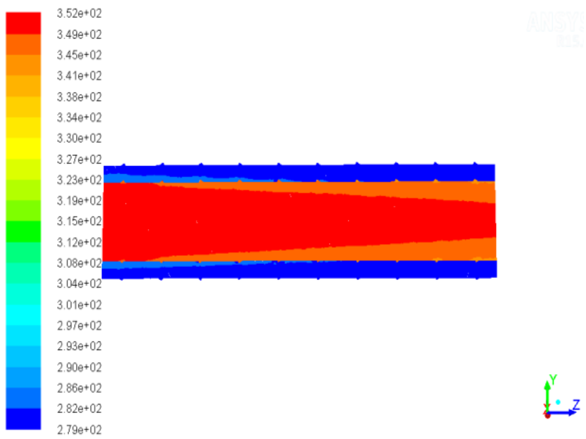

Figure 9: The contour map of total temperature inside the heat exchanger

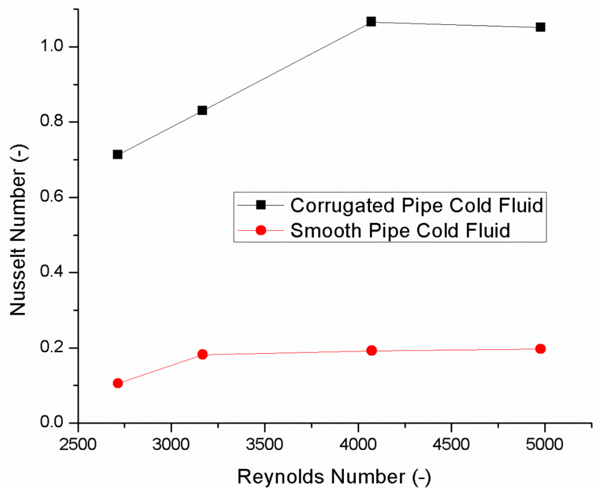

Figure 10: Nu vs. Re variation for cold water

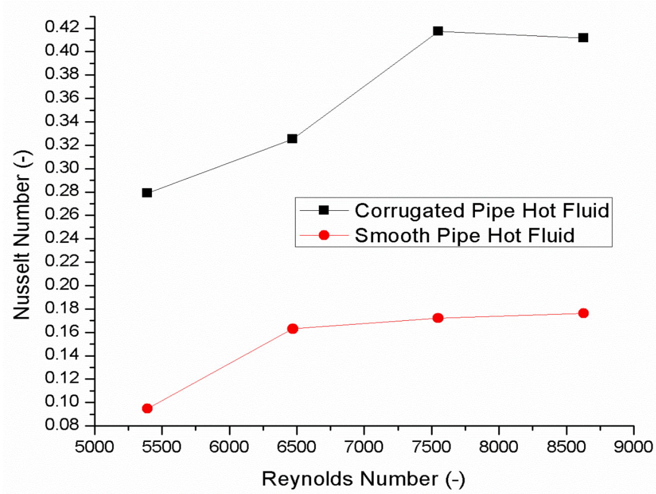

Figure 11: Nu vs. Re variation for hot water

3.2 Temperature and Turbulent Kinetic Energy Variations

Fig-9 shows the contour map of total temperature distribution inside the heat exchanger. Change in the contour color indicates the heat transfer between both the fluids. The minimum temperature found to be 279K and the maximum temperature goes up to the limit of 352K inside the heat exchanger. The heat transfer rate between two fluids can be calculated using empirical formulas. Fig-10 and Fig-11 shows the variation in Nusselt number with the Reynolds number for cold and hot water flowing through the outer and inner corrugated and smooth pipes of double pipe heat exchangers. It can be observed from the plots that the Nusselt number is found to increase with the increase in the Reynolds number. This may be due to the increase in the convective heat transfer coefficient which leads to increase in total heat transfer rates. Also, the Nusselt number for smooth pipe flow is found to lesser than the flow through the corrugated pipe.

4. Conclusions

Following conclusions have been drawn from the present study:

- It has been found that the pumping power is increases with the increase in mass flow rate for both hot and cold fluid. For smooth pipe the pumping power is found to less than that of corrugated pipe flow.

- Friction factor (Fig and Fig) is found to reduce with the increase in Reynolds number which results in lower pumping power which is required to pump the fluid. Also, the friction factor for corrugated pipe is found to higher than smooth pipe. This is due to the presence of corrugation in the double pipe heat exchanger.
- It has also been found that the heat exchanger length (518mm) is not sufficient in order to have a fully developed flow (**Error! Reference source not found.** and **Error! Reference source not found.**) in the inner corrugated pipe.

As Nusselt number is found to increase with Reynolds number therefore it can be concluded that the total heat transfer rates will increase with the increase in the convective heat transfer coefficients. Also, Nusselt number is found to be higher for corrugated DPHE than smooth pipe flow.

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