Heat Transfer and Friction Factor Characteristics of Turbulent Flow through a Conical Heat Exchanger

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Abstract: In this present energy demand scenario, the use of a waste heat recovery device like regenerator, recuperator, reheaters in the industrial processes is gaining much more importance due to improvement in thermal efficiency, and consequently reduction in thermal pollution. In the power plant, the waste heat carried by hot flue gases through the chimney remains untapped which can be recovered through the use of a conical heat exchanger at the inner portion of the chimney. Also, few customized designs of the conical coil heat exchanger can have the potential to be used to recover the waste heat in other industrial processes like chemical reactors, electrical contractors, HVAC applications, condenser, boiler, and evaporator, etc. In the present study, different arrangements of the conical coil heat exchanger are analyzed using a numerical technique. A conical coil of different geometrical configurations viz. at three different cone angles 45°, 90° and 135°, and at three different pitches 20 mm, 25 mm, and 30 mm are considered. The results for heat transfer and friction factor are presented for the flow of fluid of different Reynolds numbers. The impact of geometrical parameters as well as flow parameters on pressure drop and heat transfer under constant wall heat flux conditions are presented. The average heat transfer and pressure drop values are observed to be 175 to 225% more than the smooth tube values with increased pressure drop. Also, the results indicate that there is a nonlinear increase in both Nusselt number and friction factor with the increase in cone angle and pitch of conical coil.

Keywords: heat recovery; heat transfer; conical heat exchanger; turbulent flow.

1. Introduction
Now a day’s waste heat recovery in industrial processes like combustion is gaining much more importance. Waste heat recovery not only helps in improving thermal efficiency but also helps in lowering the thermal pollution caused because of industrial processes. In power plants regenerators, recuperators, reheaters serve the purpose of waste heat recovery. But the heat carried by flue gases through the chimney remains untapped. So, there is scope for heat recovery through the chimney by designing a conical shape of the heat exchanger. The same design of conical coil heat exchanger can also be used for waste heat recovery in other industrial processes like chemical reactors, electrical contractors, HVAC applications, condenser, boiler, and evaporator, etc.

2. Literature Survey
Most of the experimental and numerical studies has been conducted towards the development of simple helical coil heat exchangers, whereas conical coil heat exchanger has not gained so much attention. This may be because of its complex design and construction over the helical coil heat exchanger. Purandare et al. [1] carried out an experimental study on conical coil heat exchangers for
thermal analysis. The observations were reported for thermohydraulic characteristics at a different flow rate of cold and hot fluid. The various fluid flow and geometrical parameters like Nusselt number, friction factor, and effectiveness are estimated for the heat exchanger. Joshi et al. [2] Conjectured a primitive design of one conical heat exchanger with the purpose of exhaust gas heat recovery of twin-cylinder diesel engine. The experimentation brought out the relationship between temperature rise in water for various flow rates. A simple program developed for prediction of overall heat transfer coefficient is found to underpredict the values than actual leaving ample scope for characterizing heat transfer coefficient. Waghmare et al. [3] Conducted comparative numerical analysis of straight and conical coil heat exchangers. The results show enhanced heat transfer for the conical coil as compared to the straight coil heat exchanger. Prabhanjan et al. [4] reported an experimental study on fluid-to-fluid heat exchangers and compared the heat transfer rates between a helically coiled heat exchanger and a straight tube heat exchanger. Experimental results showed that the heat transfer coefficient is a function of geometrical parameters and the temperature of the water bath containing the heat exchanger. It is observed from the literature that additional secondary flow is developed within the mainstream flow. This additional flow causes the overall increase in turbulence level of the fluid which primarily responsible for the enhancement of heat transfer. The present study focuses on a simulation study of the conical heat exchanger of three different cone angles 45°, 90° and 135°, and at three different pitches 20 mm, 25 mm, and 30 mm. The results of the present study are presented for heat transfer and friction factors in the conical heat exchanger at different inlet flow conditions.

3. Numerical Model
3.1. Geometry
The geometry and different geometrical parameters of the conical heat exchanger used in the present study are shown in Figure 1 and Figure 2 shows 3D view of the conical coil

![Diagram of conical heat exchanger](image)

\[ \theta \text{ - Cone Angle} \quad D1 \text{ – Minimum Diameter} \quad D2 \text{ – Maximum Diameter} \]

\[ d \text{ – Tube Diameter} \quad Dm \text{- Mean Diameter} \]

**Figure 1.** A sectional view of conical coil
Figure 2. 3-Dimensional view of a conical coil (45° cone angle and 15 mm pitch)

Table 1. Geometric elements of a conical heat exchanger

| Parameter                      | Details     |
|--------------------------------|-------------|
| Coil Mean Diameter             | 200 mm      |
| Total Number of coils          | 9           |
| Coil Pitch (mm)                | 15, 20, 25  |
| Cone Angle (θ)                 | 45°, 90°, 135° |
| Tube Size ID x OD (mm x mm)    | 10 x 12     |
| Tube Length (mm)               | 3000        |

3.2. Assumption and Boundary Condition

The flow of the fluid through the conical coil is assumed to be three dimensional, incompressible, the steady-state flow of water. The Reynolds number at the inlet of the tube is varied from 4000 to 12000 in the steps of 2000. The boundary/zone conditions considered for analysis are listed below:

- Solver: Pressure Based Steady state
- Model: Viscous K- Epsilon Turbulence Model
- Fluid Domain: Water – Solid
- Solid Domain / Tube Material: Copper
- Tube / Coil inlet: Velocity Inlet type with Temperature Ti (=300K)
- Tube Coil outlet: Outflow Type
- Wall Boundary Condition: uniform wall heat flux at the exterior wall (such that tube outlet temperature remains To = 330K)
- All other walls: Stationary No Slip

3.3. Governing Equations

The rate equations for the cylindrical coordinate system are mass conservation, momentum equations, and energy equation.

3.3.1 Conservation of Mass:

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

3.3.2 Momentum Equation for turbulent flow:
\[
\frac{\partial (\rho u)}{\partial t} + \frac{\partial (\rho u u_i)}{\partial x_i} = -\frac{\partial (P)}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \mu \left( \frac{\partial (u)}{\partial x_j} + \frac{\partial (u)}{\partial x_i} \right) - \frac{2}{3} \mu \left( \frac{\partial (u)}{\partial x_i} \right) \delta_{ij} \right)
\]

3.3.3 Conservation of Energy:

\[
\frac{\partial}{\partial x_i} \left( \rho u_i C_v T - k \frac{\partial T}{\partial x_i} \right) = u_i \frac{\partial P}{\partial x_i} + \left[ \mu \left( \frac{\partial (u)}{\partial x_j} + \frac{\partial (u)}{\partial x_i} \right) - \frac{2}{3} \mu \left( \frac{\partial (u)}{\partial x_i} \right) \delta_{ij} \right]
\]

3.4. Model Validation and Grid Independency

Before the simulation for conical heat exchanger, a validity of simulation model is carried out by modeling the flow through a straight tube of length equal to 3000 mm. The flow through the straight tube is simulated at different values of Reynolds number. Figure 3 shows the comparison of the values of the present simulation study with the analytical results obtained using the Dittus-Boelter equation for the smooth tube. It is observed that the variation of the average Nusselt number at different Reynolds numbers is less than 5%. This ensures the correctness of the simulation methodology used in the present study.

![Figure 3. Validation of Nusselt number, Nu with Reynolds number, Re](image)

It is reported in the literature that the K-epsilon model overpredicts the analytical results by around 7 to 8% which is well within the acceptable range. The grid independence test for pressure drop is carried out. The results as shown in Figure 4 are plotted against the number of elements as other parameters (velocity and outlet temperature) remain almost constant for any number of elements. It is found that for the number of elements more than 75,95183 (for which the number of divisions in edge sizing is 60) the pressure drop almost remains constant and it is independent of the number of elements.
Figure 4. Grid independence check for pressure drop

3.5. Meshing
The meshing of the geometric model is done in ANSYS 2019 R3 academic version. Figure 5 shows the mesh model of 45 cone angle coil divisions. For meshing of geometry, edge sizing of the faces is done and this face mesh is swept all along the length of the tube to ensure the smooth flow of the element all over the mesh (Figure 6).

Figure 5. Mesh model of 45 cone angle coil divisions

Figure 6. Face sizing with 60 number of divisions

4. Data Reduction Method
The heat transfer and pressure drop parameters viz. Nusselt Number and friction factor and flow parameter like Reynolds Number, are evaluated using following well-established relations.

Tube Reynolds number,

\[ \text{Re} = \frac{\rho V d}{\mu} \]  (1)

Average Nusselt number,

\[ Nu = \frac{h d}{K_w} \]  (2)
Friction factor,

\[ f_c = \frac{\Delta P}{\frac{d}{2 \rho V^2 L}} \]  

(3)

5. Results and Discussion

5.1. Temperature and Pressure Contours:

The pressure and temperature contour of the conical coil at the outer wall are shown as follows. It is observed from Figure 7 that the pressure values are almost constant at a given cross-section. However, as it is a case of pressure-driven flow, it is observed that the value of absolute pressure reduces gradually from the inlet to the exit of the pipe. Therefore, it can be concluded that the flow through a conical pipe is associated with a large pressure drop. The contours of pressure are mentioned in Figure 8, shows that along the conical path of the fluid the surface temperature of the tube increases from inlet to the exit of the conical pipe. It is also observed that there exists a variation of the surface temperature of the tube in the circumferential direction.

![Figure 7. Pressure Contour](image1)

![Figure 8. Temperature Contour](image2)

5.2. Variation of Nusselt number and friction factor

The change in average Nusselt number in terms of Reynolds number at the tube inlet is shown in Figure 9 for the conical coil having cone angle equal to 45° at three different pitches viz 15mm, 20 mm, and 25mm at the configuration of the coil. It is observed that the Nusselt number rises with the increase in Reynolds number. The values of the Nusselt number at a particular Reynolds number are observed to be very close for all three pitches. Hence, it can be concluded that the variation of the pitch does not affect heat transfer for conical coil configuration of cone angle equal to 45°. The generation of swirl flow for three different pitches is observed to be uniform maintaining the same heat transfer enhancement.
Figure 9. Deviation of Nusselt number, $Nu$ with Reynolds number for 45° cone angle coil

Similarly, the change of average friction factor with tube inlet Reynolds number is shown in Figure 10 for the configuration of the coil of cone angle 45° at three different pitches viz 15mm, 20 mm, and 25mm. It is observed that the average friction factor gradually reduces with an increase in the value of the Reynolds number. The higher values of friction factor are observed at low values of Reynolds number while the variations in these values are observed to be less at higher Reynolds number. Another interesting feature can be interpreted from Figure 9 that the effect on friction factor for three different pitches is found negligible as these values are observed to be very close at a particular Reynolds number.

Figure 10. Deviation of friction factor with respect to Reynolds number for 45° coil cone angle
5.3. Effect of Cone angle

The change of average friction factor with the Reynolds number is shown in Figure 11 for the configuration of a coil having a pitch of 15 mm at three different cone angles 45°, 90°, and 135°. It is to be noted that the values of the average friction factor shows slightly higher values at a higher cone angle. However, this is valid only for lower values of Reynolds number till 6000. At higher values of Reynolds number more than Re=6000, a negligible variation is observed with the cone angles. It can be concluded that the variation of cone angle does not change the turbulence level within the tube, at higher values of Reynolds number, resulting in the same values of average friction factor at three different cone angles. The higher values of the average friction factor are observed at low Reynolds numbers while these values are observed significantly low at higher flow rates.

![Figure 11. Variation of Friction factor with cone angle for 15 mm pitch coils](image1)

![Figure 12. Variation of Nusselt number with cone angle for 15 mm pitch coil](image2)
The average Nusselt number with varying Reynolds numbers is shown in Figure 12 for a 15 mm configuration coil at three different cone angles 45°, 90°, and 135°. It is to be noted that the average Nusselt number shows negligible variation with the cone angles. It can be concluded that the variation of cone angle does not affect the flow characteristics significantly, therefore produces the same values of average heat transfer at three different cone angles. The higher values of heat transfer are observed at Reynolds number equal to 12000 while lower values are observed at Reynolds number of 4000.

5.4. Effect of Pitch

Figure 13 shows the variation of average friction factor for the configuration of the conical coil of cone angle 90° of three different pitches viz 15mm, 20 mm, and 25mm at same Reynolds number. It is noticed that the average friction factor almost remains constant with an increase in the value of coil pitch. This is observed at all the values of Re studied. It also can be confirmed from Figure 13 that higher values of friction factor are produced at low values of Reynolds number. These results bring the important conclusion that the effect of the pitch of a conical coil is minimum at a particular Reynolds number value.

Similarly, the variation of the average Nusselt number for the configuration of the conical coil of cone angle 90° of three different pitches at the same Re is shown in Figure 14. It is also observed that the configurations of different pitches provide uniform heat transfer enhancement at different coil pitches. This is observed for all flow conditions concerning different values of Re studied. It also can be confirmed from figure 13 that higher heat transfer is possible at higher values of Reynolds number and vice versa.
It can also be concluded from Figures 13, and 14 that the higher values of Reynolds number produce higher heat transfer at the expense of low values of friction factor. Thus, it is recommended that the flow rates through the conical coil can be maintained higher as lower flow rates produce less heat transfer at the expense of relatively high-pressure drops.

6. Conclusion
The various configurations of a conical heat exchanger are studied using a numerical technique. A conical coil heat exchanger of three different cone angles 45°, 90°, and 135°, and at three different pitches 20 mm, 25 mm, and 30 mm are presented for the study of heat transfer and friction factor characteristics at different flow rates. The influence of geometrical and flow parameters on pressure drop and heat transfer is presented for heat exchanger tube exhibited to constant wall heat flux condition. The average heat transfer values are observed to be 175 to 225% more than the corresponding smooth tube values with increased pressure drop. The numerical results reveal that a nonlinear increase in both Nusselt number and friction factor with the increase in cone angle and pitch of the conical coil.

Nomenclature

| Symbol | Description |
|--------|-------------|
| $A$    | surface area of coil, (m$^2$) |
| $C_p$  | Specific heat, (kJ/kg.K) |
| $D$    | Mean Coil Diameter, (mm) |
| $d_o$  | Outside tube diameter, (mm) |
Inside tube diameter, (mm)

Heat transfer coefficient, (W/m²K)

Length of wire coil, (mm)

Thermal conductivity of water, (W/m.K)

Number of turns

Pitch of the coil, (mm)

Pressure drop, (Pa)

Heat transfer, (W)

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