Comparative Study for Thermal and Fluid Flow Peculiarities in Cascading Spiral Inner Tube Heat Exchanger with or without Diverse Inserts over Spiral Tube

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Abstract. Present work deals with experimental analysis in which various observation takes place related to performance characteristics like HTR, effectiveness and fluid flow peculiarities like Nu, Re, f, NTU, TPF etc in a cascaded spiral concentric tube type heat exchanger consisting of a GI shell and a cascaded spiral copper inner tube having conical shape inserts, hemispherical insert and plane profile unit for parallel flow. Five spiral cascades with three turns in each are used for the inner side copper tube unit. Water enters at 304°K from the innermost turn along the cascaded spiral tube, and leaves at the outermost turn. The hot fluid inlet for heat exchanger is from bottom of one end of the shell at 333°K, flowing radically across the cascades, and then leaving the exchanger from the top of another end. This paper based on comparative experimental study of the behavior of such system for parallel flow with different shape of inserts, and then compared with the simulation results for same. In this analysis comparative approach is considered for evaluating optimum fluid flow and thermal properties on the basis of geometrical parameters. The results predicted that out of three profiles of insert, plain, conical and half cut elliptical profile, which located over inner copper tube of exchanger, half cut elliptical profile insert shows best results on fluid flow and thermal peculiarities, for mass flow rate range 0.012 to 0.049 Kg/s and Reynolds number range 4236 to 18540, heat transfer rate varies from 1128.06 to 2374.77, Nusselt number varies from 75.15 to 144.05, Friction factor varies from 0.00376 to 0.00305 and effectiveness varies from 0.77 to 0.40 and Number of transfer unit varies from 2.08 to 0.68 for half cut elliptical profile.

1. Introduction

As per previous literature it has been observed that phenomenon of heat transfer plays an crucial role in day to day appliances such as light weight equipments like laptops for effective cooling purpose, heavy components used in industrial sectors like heat exchangers for waste heat recovery, solar based equipments like solar air heater, solar water heater etc regarding effective utilization of non conventional energy resource like sun, that why research e have keen interest on such type of investigation in which effect of various types of insert, micro channel, baffles etc[11,12,13,14,15,18], along with diverse fluid [16,17] on thermal and fluid flow peculiarities involved and responsible for thermal performance enhancement in solar based heat transfer or exchangers type devices. Also, present ongoing researches and demands, it is necessary to find the behavior of differently augmented heat exchangers and heat transfer surface due to several engineering implementation and insinuation dealt with it. More efficacious and veracious heat exchanger type devices can be made through augmentation and economic reconsideration regarding heat transfer. A decrement in heat exchanger
size has been seen, with its increased performance [1], poor performance of tube type heat exchangers have been noticed while handling high viscosity fluids in laminar flow due to very little heat transfer because of inefficient thermally boundary layer. Control of heat transfer is governed by the extent of flourished boundary layer and thermal conductivity associated with tube material. The effect of curved tube within a CTHE has been analyzed by many researchers in near future. Garimella et al. [2] investigated forced convection heat transfer and determined average heat transfer coefficients of transition flows and laminar flow for coiled type annular ducts. Prabhanjan et al. [3] studied HCHE and compared its result with straight tube specifically for heat transfer rate. Ho et al. [4–6] derived the correlations of HTC for tube-side and air-side. Bhola et al. [7–8] conducted simulation for finding the effect of rectangular insert in parallel flow and counter flow arrangements of CTHE. Analyzing above literatures shows that either spirals, helical or inserts have been used to maximize the heat transfer of a CTHE, which enables us to use a cascaded system for our research.

2. Governing Equations:

The governing equations for investigating the thermal and fluid flow peculiarities in the experimental test set up was carried out by following steps, in first step discharge and velocity is evaluated experimentally for determining the mass flow rate of hot and cold fluids, the governing equation is given as,

\[ Q_d = \frac{V}{t} \]  
\[ Q_d = A \times V \]  
\[ m = \rho \times A \times V \]

Here, the term \( m \) is the mass flow rate of hot and cold fluid which is the major term used to evaluate heat transfer rate of hot fluid and cold fluid which is second step. \( Q \) is calculated using the equation given below.

\[ Q_h = m_h \times c_{ph} \times (T_{in} - t_{out}) \]  
\[ Q_c = m_c \times c_{pc} \times (T_{out} - t_{in}) \]

Where, \( T_{in}, T_{out} \) and \( t_{in}, t_{out} \) is the inlet temperature of shell side heat exchanger, outlet temperature of shell side heat exchanger and inlet temperature of spiral tube side heat exchanger, outlet temperature of spiral tube side heat exchanger.

The third step using LMTD method for evaluating HTC and OHTC which is governed by the following equation:

\[ \eta \]  

| Nomenclature | Description |
|--------------|-------------|
| \( Q_d \)   | discharge   |
| \( Q_c \)   | heat transfer rate of hot water |
| \( m_c \)   | mass flow rate of hot water |
| \( c_{pc} \) | mass flow rate of hot water |
| \( k_h \)   | heat transfer coefficient |
| \( Q_d \)   | helical coil heat exchanger |
| \( Q_c \)   | heat transfer coefficient |
| \( m_c \)   | heat transfer coefficient |
| \( c_{pc} \) | helical coil heat exchanger |
| \( k_h \)   | heat transfer coefficient |
| \( \mu_h \) | viscosity of hot fluid |
| \( \mu_c \) | viscosity of cold fluid |
| \( \rho_h \) | density of hot fluid |
| \( \rho_c \) | density of cold fluid |
| \( \rho_h \) | density of hot fluid |
| \( \rho_c \) | density of cold fluid |
| \( \eta \)  | thermal performance factor |
\[ Q = h \cdot A \cdot \Delta T \]  
\[ \Delta T = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2) \]  
\[ \Delta T_1 = (T_{\text{out}} - t_{\text{in}}) \]  
\[ \Delta T_2 = (T_{\text{in}} - t_{\text{out}}) \]  
\[ U = 1 / (1 / h_i + 1 / h_o) \]

After determining above values forth step is used to evaluate the dimensionless properties of fluid like Nusselt number, Reynolds number and Friction factor that was governed by following equation.

\[ \text{Nu} = h \cdot D / k = 0.023 \; \text{Re}^{0.8} \text{Pr}^{0.4} \]  
\[ \text{Re} = \rho \cdot V \cdot D / \mu \]  
\[ f = \Delta P / (L / D)^{5} (\rho \cdot V^{2} / 2) \]

At last after determining above values for dimensional less parameters fifth step is used to evaluate the effectiveness and number of transfer units through NTU method of CSTHE where, effectiveness is the ratio of actual to maximum possible heat transfer that was governed by following equation.

\[ \varepsilon = Q_{\text{actual}} / Q_{\text{max}} \]  
\[ Q = m_{h} \cdot c_{ph} (T_{\text{out}} - t_{\text{in}}) = m_{c} \cdot c_{pc} (T_{\text{in}} - T_{\text{out}}) \]  
\[ Q_{\text{max}} = C_{\text{min}} \cdot (T_{\text{in}} - t_{\text{in}}) \]  
\[ T_{\text{out}} = T_{\text{in}} - Q_{\text{max}} / C_{\text{max}} \]  
\[ t_{\text{out}} = t_{\text{in}} + Q_{\text{max}} / C_{\text{min}} \]  
\[ \text{NTU} = U \cdot A / C_{\text{min}} \]

Where, \( C_{\text{min}} = m \cdot c_{p} \)

\[ \text{NTU}_{\text{parallel}} = - \left[ \ln \left( 1 - (1-c)/(1+c) \right) \right]^{0.33} \]

\[ \eta = (\text{Nu} / \text{Nu}_{i}) / (f/f_{i})^{0.3} \]

3. Experimental Test Set Up Layout and Instrumentation:

Figure 3.1 shows a simplistic diagram of experimental set up and Figure 3.2 shows the simplistic diagram of plane geometrical profile of spiral tube and spiral tube with different geometrical shapes inserts like triangular insert, hemispherical insert located over spiral tube. The test rig consists of a CTHE with cascaded spiral inner tube and data procuring system. Water is the active fluid. Whole set up is fabricated in such a way that its components can be altered or overhauled easily. In addition to the loop component lively set of apparatus for measuring and controlling the temperature and flow rate of all fluids is installed at all important points in the circuit. In the experimental set up six thermocouples T1, T2, T3, T4, T5 and T6 are installed at inlet and outlet of shell side GI pipe, inlet and out let section of spiral copper tube, hot water tank and cold water tank for evaluating the fluid temperature of hot water and cold water, hot fluid moves from hot water tank to inlet section of shell side GI Pipe to outlet section and cold fluid moves from cold water tank to inlet section of spiral copper tube side to outlet section. Flow rate measurement devices were also installed at inlet and out let section of tube and pipe for determining the pressure drop and mass flow rate. During experimentation readings were noted with the help of flow meters, pressure meters and digital temperature indicator device where all thermocouple are attached. Table number 3.1 shows all the geometrical dimensions of the cascaded spiral tube heat exchanger which were used to fabricate the set up and Table number 3.2 shows all the operating condition used during experimentation.
Figure Number 3.1: Schematic diagram of the experimental setup

(a) Inbuilt Heater Tank (b) Hot Water Tank (c) Thermocouple (d) Pressure Gauge (e) Cold Water in

Figure Number 3.2: (a) Side view of conical shape insert located over spiral tube. (b) Isometric view of conical shape insert located over spiral tube. (c) Side view of plain profile of spiral tube. (d) Front view of plain profile of spiral tube. (e) Side, Front, Isometric view of hemispherical shape insert located over spiral tube.
Table 3.1: Geometrical Dimensions of cascaded spiral tube heat exchanger

| Parameters                                      | Dimensions |
|------------------------------------------------|------------|
| Outer diameter of the tube, mm                  | 20         |
| Inner diameter of the tube, mm                  | 16         |
| Inner most diameter of the spiral coil tube, mm | 10.6       |
| Outer most diameter of the spiral coil tube, mm  | 12.6       |
| Number of coil turns                            | 3          |
| Number of spiral coils                          | 6          |
| Distance between spiral coils, mm               | 11.4       |
| Outer diameter of shell, mm                     | 150        |
| Inner diameter of shell, mm                     | 144.12     |
| Length of the shell, mm                         | 850        |
| Diameter of inlet and outlet hole of hot water, mm | 15         |

Table 3.2: Experimental conditions of cascaded spiral tube heat exchanger

| Parameters                                      | Hot Water | Cold Water |
|------------------------------------------------|-----------|------------|
| Inlet water temperature, (°C)                  | 60        | 31         |
| Mass flow rate, (kg/s)                         | 0.067     | 0.012-0.049 |
| Specific heat c_p, (J/kg. °C)                  | 4185      | 4178       |
| Prandtl number (Pr)                            | 2.99      | 5.3        |
| Thermal conductivity K, (W/m². °C)             | 653       | 619        |
| Viscosity µ, (Ns/m²)                           | .000474   | .000780    |
| Density ρ, (kg/m³)                             | 983       | 995        |

4. Results and Discussions:

It has been observed from the figure number 4.1 that the Reynolds number range from 4236 to 18540 as shown in figure, friction factor shows a decrement for different profiles and inserts and its values varies from 0.00588 to 0.00513 for plane profile of spiral shape copper tube having no inserts, 0.00495 to 0.00427 for conical shape inserts located over spiral type inner copper tube profile and 0.00376 to 0.00305 for spiral type copper tube profile having hemispherical shape inserts, where as the simulated value of friction factor for exchanger also shows an decrement for different profiles and inserts and its values varies from 0.00940 to 0.00404 for plane profile of spiral shape copper tube having no inserts, 0.00491 to 0.00325 for conical shape inserts located over spiral type inner copper tube profile and 0.00320 to 0.00203 for spiral type copper tube profile having hemispherical shape inserts. On the other hand experimental values of Nusselt number shows an enhancement for different shape of inserts and its values varies from 35.05 to 98.52 for plane profile of spiral shape tube, 75.15 to 144.05 for conical shape inserts located over spiral copper tube profile and 45.15 to 123.95 for spiral tube profile having hemispherical shape inserts. On comparing the experimental values of these three profiles hemispherical shape insert shows best and most promising results among other profiles, because of good turbulence in fluid flow due to the reason of insert shape and available surface area, the best and optimum Nusselt number is 144.05 and friction factor is 0.00305 for best suitable mass flow rate is 0.049 Kg/s, where as the simulated value of Nusselt number in exchanger also showed an increment for different profiles and inserts and its values varies from 40.25 to 120.25 for plane profile of spiral shape copper tube having no inserts, 48.15 to 146.85 for conical shape inserts located over spiral type inner copper tube profile and 88.15 to 165.85 for spiral type copper tube profile having hemispherical shape inserts. Thus this shows a close agreement and validates the experimental and simulation values of Nusselt number and friction factor. Further the experimental values were also validated for plain spiral profile for Nusselt number and friction factor from the following governing equations as given below.

Blasius equation for Friction factor [09]

\[ f_s = 0.316 \times Re^{-0.25} \]  

(22)
Dittus–Boelter equation for Nusselt number [10]
\[ \text{Nu}_x = 0.023 \cdot \text{Re}^{0.8} \cdot \text{Pr}^{0.4} \] (23)

Figure Number 4.1: Variation between Friction Factor and Nusselt Number for varying Reynolds Number.

After comparing both the experimental and simulated values of different profiles like of plane profile of spiral copper tube, spiral coil with conical inserts and spiral coil with hemispherical inserts etc, it has been observed that simulated values were greater than experimental values due to the reason of heat loss occurred from the experimental setup, but it was noted that minimum error percentage occurred between experimental and simulation values which varies from range 2% - 4% for all the properties like Friction factor, Nusselt number, effectiveness and heat transfer rate.

Figure Number 4.2: Variation between Number of Transfer Unit (NTU) and Mass Flow Rate for varying Shapes of Inserts and without Inserts.

Figure Number 4.3: Variation between Effectiveness and Heat Transfer Rate for varying Reynolds Number.
Thus, this condition of error percentage validated the experimental and simulated values of the experimentation. Now, from the figure number 4.2, it is evidently shown that as the mass flow rate increases from 0.012 to 0.049 Kg/s the value of NTU decreases with range of 1.27 to 0.41 for plane spiral copper tube, 1.67 to 0.56 for spiral tube having conical inserts and 2.08 to 0.68 for spiral copper tube having hemispherical shape insert. On comparing these values, it has been clearly observed that hemispherical inserts located over spiral copper tube showed best and optimum results as compared to other profile due to the reason that hemispherical insert have larger contact area than conical profile and it developed more turbulent region in fluid than other profile due to which number of transfer unit (NTU) occurred more in this shape only. From figure 4.4 and 4.5, it is evidently shown that as the range of Reynolds number increasing from 4236 to 18540 the Nu/Nu_s factor f/fs factor and thermal performance factor is decreasing but conical insert profile show best results as compared to other plain profile and half cut elliptical profile insert.

5. Conclusions:
The simulated and experimental results for the CCSICTHE shows close results and similarity, hence validating each other, in future the results of this cascaded structure can be compared using an analytical approach for similar problem, and also with different designs and shapes of inserts and micro channel over inner tube with diverse fluid and flow patterns in a CTHE. It has been concluded that out of three profiles, i.e. plain, conical and half cut elliptical profile, which located over inner copper tube of exchanger, half cut
elliptical profile shows best results on fluid flow and thermal peculiarities, for mass flow rate range 0.012 to 0.049 Kg/s and Reynolds number range 4236 to 18540, heat transfer rate varies from 1128.06 to 2374.77, Nusselt number varies from 75.15 to 144.05, friction factor varies from 0.00376 to 0.00305 and effectiveness varies from 0.77 to 0.40 and Number of transfer unit varies from 2.08 to 0.68 for half cut elliptical profile. The values which was occurred during experimentation, after optimized for evaluating best optimum values for implementation of this research in different areas and from such analysis it is concluded that hemispherical shape insert shows best and most promising as compared to other profiles, The optimum values for mass flow rate is 0.049 Kg/s, Nusselt number is 144.05, friction factor is 0.00305, heat transfer rate is 2374.77 and effectiveness is 0.77.

6. References

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