Experimental and Numerical Analysis for Augmentation of Heat Transfer by Cascading Spiral Inner Tube in a Concentric Tube Heat Exchanger

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Abstract- The study deals with the experimental investigation on heat transfer rate by cascading spiral tube inside a Heat Exchanger for varying mass flow rate from .01 to .05kg/s. In total there are five cascades over a length of 1.25m exchanger at a distance of 0.2365m with 3.25 turns in each. For fabrication of inner tube, a straight copper tube with total length 6.67m and diameter 0.00675m is bent into a cascaded spiral-coil of three turns. Water is used as working fluid for the experiment. The cold water enters from the innermost turn along the cascaded spirally coiled tube, and exits from the outermost turn. The hot water enters the heat exchanger from bottom of one end of shell and flows radially across spiral tubes to the periphery and exits from the top of another end. Present work is based on experimental analysis as well as computational fluid dynamic (CFD) using ANSYS 14.5, which includes the variation of mass flow rate of cold fluid and its effects on heat exchanger and heat transfer coefficient. The experimental result shows that Nusselt number and head loss increases with increase in mass flow rate whereas friction factor and effectiveness decreases with increase in mass flow rate, the experimental results are then compared with CFD simulation for similar concentric tube arrangement for validating the experimental results. In this analysis parallel flow arrangement is considered.

Key Words- *Spiral inner pipe, Cascading, Heat Transfer enhancement, CFD Analysis (ANSYS)*

I. INTRODUCTION

Recently many researches have executed their researches for enhancing the heat transfer characteristics of such systems by using passive, active or compound technique for heat transportation. Passive technique is used for enhancing the heat transfer in present research because of economical, space and enhanced heat rate of heat transfer advantages. Turbulent flow is used for experiment because viscous flow shows poor heat transfer characteristics in laminar flow. Literature survey for present work includes Garimella et al.[18] probed average heat transfer coefficient for coiled annular ducts considering laminar and transition flows in forced convection. Bolinder and Sunden [3] interpreted the parabolized Navier–Stokes and energy equations by utilizing a FVM .Zhenget al. [1] enforced control-volume finite difference method with second-order accuracy for resolving the 3-D fluid domain to analyze thermal radiation within a helical pipe for laminar flow. Acharya et al. [11] numerically deciphered heat transfer rate for coiled-tube heat exchangers. Lin and Ebadian [16] utilized the standard k–ε model for 3-D turbulent convectivetype heat transfer in helical tubes. Sillekens et al. [9] engaged the finite difference discretization scheme to fix the parabolized Navier–Stokes and energy equations for fluid domain. Rindt et al. [2] reviewed the axial varying wall temperature and compared the result with constant wall temperature boundary conditions for heat exchangers. Lemenand and Peerhossaini [5] interpreted the Navier–Stokes and energy equations as a thermal model to predict heat transfer relationship using a twisted pipe heat exchanger. Ho et al. [4–6] derived the correlations of heat transfer coefficients for tube-side and air-side in a concentric tube heat exchanger.Naphon and Wongwises [12] suggested some correlation for the average heat transfer coefficient in spiral coil heat exchanger under dehumidifying conditions. In their other two papers Naphon and Wongwises [13,14] mathematically modeled heat transfer characteristics of spirally coiled finned tube heat exchangers under wet-surface conditions and dry-surface conditions. As mentioned above, Compared to the numerous investigations in the helically coiled tubes, there are few researches on the heat transfer and flow characteristics in the spirally coiled tube in open literature. This enables us to perform our preset research.

II. EXPERIMENTAL SETUP AND FLUID DOMAIN

Fig. 1 and Fig. 2 shows experimental test rig and fluid domain for simulation respectively. The test rig consists of a concentric tube heat exchanger with cascaded spiral inner tube and data procuring system. Water is the active fluid. Whole set up is fabricated such that its components can be altered or overhauled easily. In addition to the loop components, a 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.

Fig. 1: Experimental setup

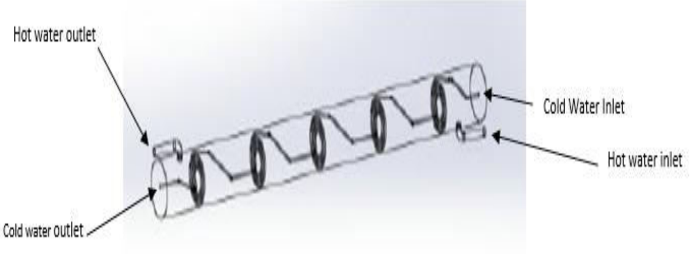

Fig. 2 Fluid domain for Simulation

Table 1: Dimensions of Cascaded Spiral Tube Heat Exchanger

Parameters	**Dimension**
Outer diameter of the tube, mm	6.75
Inner diameter of the tube, mm	4.2
Innermost diameter of spiral coil, mm	40
Outermost diameter of spiral coil, mm	100
Total coil turns	3
Total cascaded spiral coils	5
Distance between cascaded spiral coils, mm	263.3
Outer diameter of shell, mm	110
Inner diameter of shell, mm	104.12
Length of the shell, mm	1250
Diameter of hole at inlet, outlet of hot water, mm	15

Table 2: Experimental conditions

Parameter	**Hot water**	**Cold water**
Inlet water temperature, ^{o}K	333	304
Mass flow rate, kg/s	.067	.012 - .049
C_p, J/kg ^{o}C	4185	4178
Prandtl number (Pr)	2.99	5.3
Thermal conductivity (k), W/m^{2o}C	653	619
Viscosity (μ), N.s/m^2	.000474	.000780
Density (ρ), kg/m^3	983	995

The mass flow rate of cold water was varied using a variac while the hot water mass flow rate and temperature of both cold and hot water were kept constant during inlet. A centrifugal pump was used to control the flow rate of hot water. Steady state was achieved before recording the data.

III. MESHING OF THE COMPUTATIONAL DOMAIN

Table 3: Mesh setting for Fluid Domain

Parameter	**Heat exchanger**
Physics preference	CFD
Solver preference	Fluent
Smoothing	High
Minimum size	3.3295e-004m
Maximum size	3.3295 e-002m
Nodes	276659
Elements	1607898
Orthogonal quality	2.169555e+01

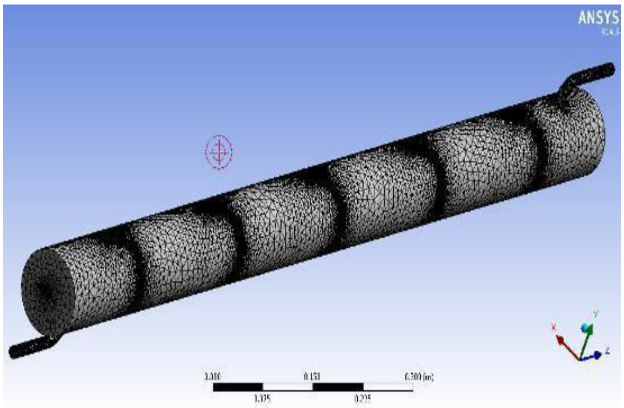

Fig. 3 Meshing of Fluid Domain

IV. Methodology

A. SIMULATION

a. CONTINUITY EQUATION

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho v_x) + \frac{\partial}{\partial y}(\rho v_y) + \frac{\partial}{\partial z}(\rho v_z) = 0$$

b. KAPPA-EPSILON MODEL

K – **Equation**

$$\rho[\bar{u}\frac{\partial k}{\partial x} + \bar{v}\frac{\partial k}{\partial r}] = \frac{\partial}{\partial x}[(\mu_l + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x}] + \frac{1}{r}\frac{\partial}{\partial r}[r(\mu_l + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial r}] + \rho g - \rho\varepsilon$$

Where, G is the production term and is given by

$$G = \mu_t[2\{(\frac{\partial \bar{v}}{\partial r})^2 + (\frac{\partial \bar{u}}{\partial x})^2 + (\frac{\bar{v}}{r})^2\} + (\frac{\partial \bar{u}}{\partial r} + \frac{\partial \bar{v}}{\partial x})^2]$$

ε **- Equation**

$$\rho[\bar{u}\frac{\partial \varepsilon}{\partial x} + \bar{v}\frac{\partial \varepsilon}{\partial r}] = \frac{\partial}{\partial x}[(\mu_l + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x}] + \frac{1}{r}\frac{\partial}{\partial r}(r\mu_l + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial r}] + C_{S1}G\frac{\varepsilon}{k} - C_{s2}\frac{\varepsilon^2}{k}$$

Here C_{s1} , C_{s2} , σ_k and σ_ε are the observational turbulent constant. Considering these valves according to the Launder *et al.,* 1974. The values of Cμ, C_{s1} , C_{s2} , σ_k and σ_ε are 0.09, 1.44, 1.92, 1.0 and 1.3 respectively.

c. ENERGY EQUATION

The conservative form of energy equation, in terms of total energy is written as follows

$$\rho_{Dt}^{DE} = -div(pu) + [\frac{\partial(u\tau_{xx})}{\partial x} + \frac{\partial(u\tau_{yx})}{\partial y} + \frac{\partial(u\tau_{zx})}{\partial z} + \frac{\partial(v\tau_{xy})}{\partial x} + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{zy})}{\partial z} + \frac{\partial(w\tau_{xz})}{\partial x} + \frac{\partial(w\tau_{yz})}{\partial y} + \frac{\partial(w\tau_{zz})}{\partial z} + div(k\ grad\ T) + S_E]$$

d. BOUNDARY CONDITION

For turbulent flow. The quantities *U, k* and *ε* are obtained using numerical calculations based on the *k-ε* model.

1) At the inlet of the channel:

$$u = U_{in,} v = 0$$

$$k_{in} = 0.005U_{in}^{2}$$

$$\varepsilon_{in} = 0.1K_{in}^{2}$$

K_{in} represents the admission condition for turbulent kinetic energy and ε_{in} is the inlet condition for dissipation.

2) At the walls:

$$u = v = 0$$

$$k = \varepsilon = 0$$

3) At the exit:

$$P = P_{atm}$$

B. EXPERIMENT

Relation of total heat transfer is vital with the quantities like inlet and outlet fluid temperature, overall heat transfer and total surface area to be used for heat transfer for scheming and forecasting the conduct of a heat exchanger.

a. LMTD METHOD

If rate of total heat transfer between the hot and cold active fluid is q and neglecting the heat transfer between the device and its surroundings, application of the steady flow energy equations states that-

$$Q = \dot{m}c_p\Delta T$$

$$\Delta T_m = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)}$$

Total heat exchange in heat exchanger tube can be written as

$$Q = U_o A_o \Delta T_m = U_i A_i \Delta T_m$$

Where $U_i$andU_o refer to overall heat transfer coefficients with respect to inner and outer surface areasA_o andA_i respectively, and ΔT_m is the mean effective temperature difference between the fluids. h_i and h_o represent respective convective heat transfer coefficient and $r_i$and r_o represent respective radii, then

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i}$$

$$\frac{1}{U} = \frac{1}{h_o A_o} + \frac{\ln(r_o/r_i)}{2\pi k l} + \frac{1}{h_i A_i}$$

b. ε- NTU METHOD

This method is used to determining the capacity of the heat exchanger. When the exchanger is given and one wants to know the heat exchanger capacity, effectiveness-NTU method is used.

- $NTU = \frac{UA}{(\dot{m}c_p)_{min}}$
- Heat capacity ratio – $C = \frac{C_{min}}{C_{max}}$
- Effectiveness -

$$\varepsilon = \frac{q}{q_{max}} = \frac{1-\exp[-NTU(1+C)]}{1+C}$$

- $Q_{max} = \dot{m}c_{p,min}(T_{h,i} - T_{c,i})$
- Outlet temperatures:

For hot fluid, $T_{h,o} = 333 - \frac{Q}{C_{max}}$

For cold fluid, $T_{c,o} = \frac{Q}{C_{min}} - 304$

V. RESULT

In present work experiments and simulations were carried out with variable mass Reynolds number for the geometry considered here for heat exchanger. The results obtained from the experiment and counterfeit analyses are compared in form of graphs below.

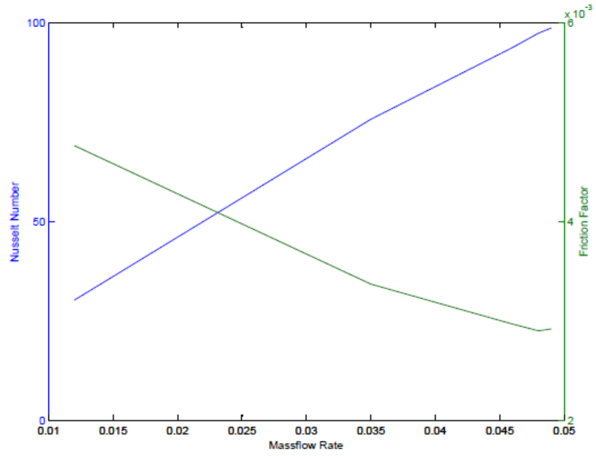

Fig. 4 Variation of Nusselt Number and Friction Factor w.r.t mass flow rate for experimental work

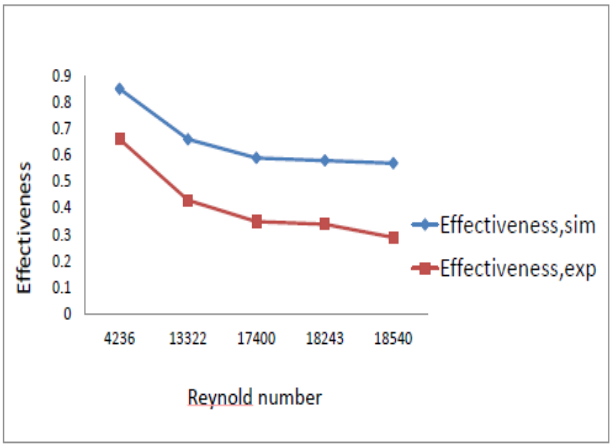

Fig. 5 Comparison of effectiveness w.r.t Reynolds number for simulation and experimental work

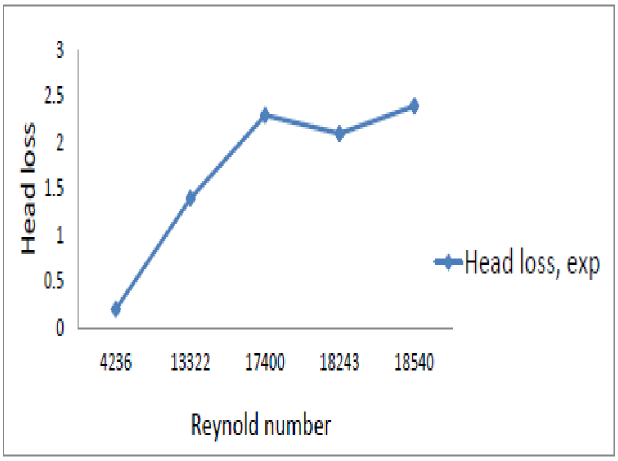

Fig. 5 Variation of Head Loss w.r.t Reynolds number for experimental work

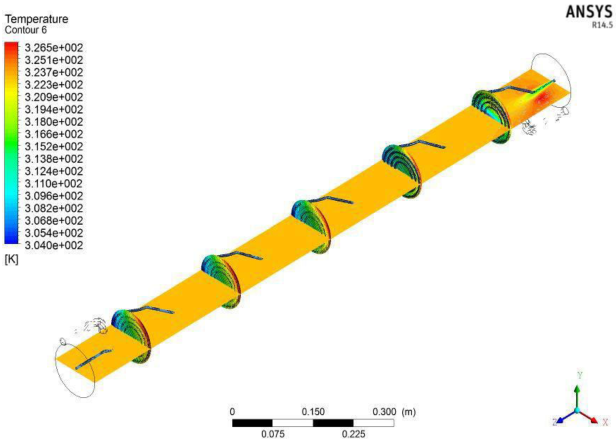

Fig. 6 Contours of static temperature at Reynolds number13322

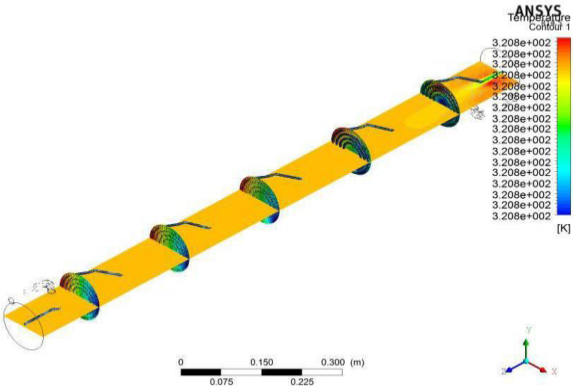

Fig. 7 Contours of static temperature at Reynolds number18540

VI. CONCLUSION AND FUTURE SCOPE

The heat transfer, friction factor and effectiveness of heat exchanger has been analyzed, and its dependency on Reynolds number is reported in this study by comparing experimental and simulated result, the results shows the similarity, and hence validating each other, in future the results of this cascaded structure can be compared with different designs of inner tube in a concentric tube heat exchanger, applying different thermal boundary condition and analyzing the variation by varying the distance of cascaded structure.

VII. REFERENCE

[1] B. Zheng, C.X. Lin, M.A. Ebadian, “Combined laminar forced convection and thermal radiation in helical pipe”, Int. J. Heat Mass Transfer 43 (2000) 1067–1078.

[2] C.C.M. Rindt, J.J.M. Sillekens, A.A. Van Steenhoven, “The influence of the wall temperature on the development of heat transfer and secondary flow in a coiled heat exchanger”, Int. Comm. Heat Mass Transfer 26 (1999) 187–198.

[3] C.J. Bolinder, B. Sunden, “Numerical prediction of laminar flow and forced convective heat transfer in a helical square duct with finite pitch”, Int. J. Heat Mass Transfer 39 (1996) 3101–3115.

[4] D.G. Prabhanjan, G.S.V. Raghavan, T.J. Rennie, “Comparison of heat transfer rates between a straight tube heat exchanger and a helically coiled heat exchanger”, Int. Comm. Heat Mass Transfer 29 (2002) 185–191

[5] H. Peerhossaini, T. Lemenand, “A thermal model for prediction of the Nusselt number in a pipe with chaotic flow”, Appl. Therm. Eng. 22 (2002) 1717–1730.

[6] Incropera, David P. Dewitt, “Fundamentals of Heat and Mass Transfer”, WILEY 2008.

[7] J.C. Ho, N.E. Wijeysundera, S. Rajasekar, T.T. Chandratilleke, Performance of a compact spiral coil heat exchanger”, Heat Recovery Syst. & CHP 15 (1995) 457–468

[8] J.C. Ho, N.E. Wijeysundera, “Study of a compact spiral-coil cooling and dehumidifying heat exchanger unit”, Appl. Therm. Eng. 16 (1996) 777–790.

[9] J.C. Ho, N.E. Wijeysundera, “An unmixed-air flow model of a spiral cooling dehumidifying heat transfer”, Appl. Therm. Eng. 19 (1999) 865–883.

[10] J.J.M. Sillekens, C.C.M. Rindt, A.A. Van Steenhoven, “Developing mixed convection in a coiled heat exchanger”, Int. J. Heat Mass Transfer 41 (1998) 61–72.

[11] Launder, B.E.; Spalding, D.B. (March 1974). "The numerical computation turbulent of flows". Computer Methods in Applied Mechanics and Engineering. 3 (2): 269–289.

[12] N. Acharya, M. Sen, H.C. Chang, “Analysis of heat transfer enhancement in coiled-tube heat exchangers”, Int. J. Heat Mass Transfer 44 (2001) 3189–3199. Analysis of Heat Transfer by Cascading Spiral Inner Tube in a Heat Exchanger 44

[13] P. Naphon, S. Wongwises, “An experimental study on the in-tube heat convective heat transfer coefficients in a spiral-coil heat exchanger”, Int. Comm. Heat Mass Transfer 29 (2002) 797–809.

[14] P. Naphon, S. Wongwises, “Investigation of the performance of a spiral-coil finned tube heat exchanger under humidifying conditions”, J. Eng. Phys. Thermophys. 76 (2003) 83–92.

[15] P. Naphon, S. Wongwises, “Experimental and theoretical investigation of the heat transfer characteristics and performance of a spiral-coil heat exchanger under dry-surface conditions, 2nd International Conference on Heat Transfer, Fluid Mechanics, and Thermodynamics, 24–26 June, 2003, Victoria Falls, Zambia.

[16] Pardhi, Baredar, “Performance Improvement of Double Pipe Heat Exchanger by Using Turbulator” IJESAT, Vol. 2, Issue 4, ISSN: 2250-3676

[17] R.C. Lin, M.A. Ebadian, “Developing turbulent convective heat transfer in helical pipes”, Int. J. Heat Mass Transfer 40 (1997) 3861–3873.

[18] S. Garimella, D.E. Richards, R.N. Christensen, “Experimental investigation of heat transfer in coiled annular ducts”, J. Heat Transfer 110 (1988) 329–336.

Mayank Bhola, M.Tech from B.T.K.I.T formerly known as K.E.C, Dwarahat in 2015. Numerical, experimental and CFD investigation for various Heat exchangers for enhancement of heat transfer characteristics, and decrement of friction factor for low operating cost.

Awadhesh Kumar Roy, B. Tech from Amrapali Group of Institutions, Department of Mechanical Engineering, Haldwani in 2017. Experimental and CFD simulation for heat transfer enhancement of concentric tube heat exchangers.

Nidhi Bora, B. Tech from Amrapali Group of Institutions, Department of Mechanical Engineering, Haldwani in 2017. Experimental and CFD simulation for heat transfer enhancement of concentric tube heat exchangers.

Shivasheesh Kaushik, M.Tech from B.T.K.I.T formerly known as K.E.C, Dwarahat in 2015. Experimental, numerical and CFD simulation for heat transfer enhancement of various types of heat reservoirs, heat transfer and thermal equipments like heat exchangers, fins, packed bed heat regenerator.