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CFD Analysis of Conical Coil Tube Heat Exchanger with Varying Cone Angle

Sumit Singh[1], Ravindra Randa[2]

[1]DDIPG Student, Mech. Dept., RGPV, Bhopal (M.P.), India, [2]Assistant Professor, Mech. Dept., RGPV, Bhopal (M.P.), India

Abstract: Now-a-days heat exchangers have become indispensable in various fields of engineering such as in air conditioning, power stations, chemical plants, automobiles, etc. It necessitates the compact size along with the high performance of the heat exchangers. This work aims to study on passive technique of heat transfer enhancement by some variation in geometry. In this work the heat transfer and pressure drop analysis has been done for conical coil with varying cone angle. The cone angles taken for the study are 0° i.e. simple helical coil, 30°, 60°, and 90°. The comparative study of these coils have been done under three different flow rates i.e. 60 lph, 130 lph, and 200 lph for the hot fluid inside the tube. The analysis is done by CFD simulation in ANSYS Fluent 14.5 considering a double pipe heat exchanger where cold fluid is flowing on the outer portion of heat exchanger with a flow rate of 200 lph in each case. It has been observed that heat transfer coefficient is maximum in case of simple helical coil while pressure drop is minimum in case of 30° conical coil.
Keywords: Conical coil, CFD analysis, Cone angle, Secondary flow, Simple helical coil, Double pipe heat exchanger.

I. INTRODUCTION

The two main aspects of effective design of heat exchangers are the consideration of simultaneous improvement of the heat transfer performance and reduction in the pressure drop. For the improvement in heat transfer performance of heat exchangers, heat transfer coefficient has an important role to play. Generally modifications which are made to enhance heat transfer coefficient leads to increase in pressure drop in heat exchangers. Hence the selection of most appropriate configuration is an important consideration in design of a heat exchanger. Secondary flow pattern formed in curved tubes plays an important role in mixing of fluid which leads to an increase in heat transfer coefficient along with an increase in pressure drop also.

Jeschke noted a substantial difference in his analysis of heat transfer coefficient of coiled tubes and straight tubes [1]. The factor base parameter on curvature ratio (δ) was proposed as: $Nu_c = Nu_s[1+3.5(r/R)]$. further Seban and McLaughlin showed that the factor i.e. [1 + 3.5 (r / R)] was not accurate and there could be more improvement possible in heat transfer coefficient [2]. Shah and Joshi noted that the enhancement in heat transfer depends on the intensity of secondary flow developed in coiled tube [3]. Eustice and many other researchers observed the fluid motion in curved pipes [4]. The main effects observed in the helical coil configuration were because of the flow pattern formed in the curved tube. The effect of curvature on flow in coiled tube was first noted by Grindley and Gibson [5]. Koutsky and Adler performed the experimental as well as numerical analysis on flow field and presented the effect of prandtl number and reynold number on the flow pattern and nusselt number [6]. Mori and Nakayama presented the fully developed flow in a curved tube for large dean number with a uniform heat flux [7]. McConalogue and Srivastava studied the secondary flow characteristics for developed laminar flow [8]. They observed that the maximum velocity shifted towards the outer wall. Due to this there is increase in velocity towards the outer wall.

The pumping power required for the heat exchanger is directly related to the pressure drop in heat exchanger. It is an important aspect from the selection point of view of the heat exchanger for various applications. In various studies pressure drop is represented mainly by friction factor. Ito was the first researcher who analyzed friction factor [9]. His experiments were performed in smooth pipes in turbulent flow conditions. For his analysis he selected the curved pipes having ratio of tube diameter to curvature diameter as 1/16.4 to 1/648. The analysis was done for the range of De of 0.034 to 300. The most important observation in his analysis was that of the equivalence in the value of friction factor in the curved tube as well in straight tube for values of De < 0.034. Nunge and Lin studied on friction factor for straight tubes and highly curved tubes with different curvature ratios [10]. He observed that at high dean number there is decrease in friction factor (f) as the curvature increases. But the results were further contradicted by the results of Austin and Seader [11]. The work on helical coil has also been performed by Jaykumar, with variation in the coil pitch from 0 to 60 mm [12]. He got same Nu value at top and bottom of the coil in case of zero mm pitch. He observed that the difference between the value of local Nu at the top and bottom of the coil increases as the coil pitch increases. This concludes that there are three forces which comes into effect, which are centrifugal force, torsional force and rotational force. Shinde Digvijay D. analyzes between a

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conical and simple helical coil and found that the inner nusselt number, convective heat transfer coefficient & overall heat transfer coefficient are higher in case of conical coil than that of simple helical coil [13].

II. PROBLEM FORMULATION

This work is concerned with carrying out three-dimensional simulations on conical coil heat exchangers geometry with different cone angles. Water is taken as hot as well as cold fluid for the analysis. The study is being carried out for cone angles 0° (simple helical coil), 30°, 60°, and 90°. Therefore total four coils were simulated to analyze the effect of cone angle on heat transfer characteristics and pressure drop across the concentric pipe heat exchanger. In all the four coils inside tube contain hot fluid flow while outside tube have cold fluid flow. The material between hot and cold side fluid is copper with a thickness of 1.12 mm. the mean diameter for each coil is kept same i.e. 210 mm. The flow rates taken for analysis are 60 lph, 130 lph, and 200 lph.

A. The geometric specifications which are common to all the coils are as below;

1) Copper tube outer diameter = 9.53 mm
2) Copper tube inside diameter = 8.41 mm
3) Outermost diameter of coil for cold fluid flow = 12.71 mm
4) Straight tube length = 5610 mm
5) Mean Coil diameter = 210 mm
6) Coil height = 170 mm
7) No. of turns = 8.5
8) Pitch of the coils = 20 mm

B. The operating parameters common to all four coils for analysis are

1) Flow rate of water in cold coil = 200 lph
2) Inlet temperature of cold coil = 298.15 K
3) Inlet temperature of hot coil = 333.2 K

C. The flow is assumed to be steady, incompressible and turbulent three-dimensional. The tube wall material is homogeneous and isotropic. No-slip boundary condition is given to the walls in contact with the fluid in the model. The working fluid in all cases including hot and cold fluid is water. The thermo-physical properties of water is below;

1) Density of water = 998.2 kg/m^3
2) Viscosity of water = 0.001003 kg/m-s
3) Specific heat of water = 4182 J/kg-K
4) Thermal conductivity of water = 0.6 W/m-K

III. METHODOLOGY

The work has been done by the help of CFD by using ANSYS Fluent 14.5. At first the geometry were made for each case according to the dimensions and then meshing was done.

The geometry of different coil after meshing is shown in the figures in next page starting from meshed geometry of simple helical coil, then conical with cone angle of 30°, then of 60° and finally of 90°.

Table with description of top coil diameter and bottom coil diameter for different geometries is given below:

Parameters	0°	30°	60°	90°
Top coil dia.	210 mm	164.4 mm	111.8 mm	40 mm
Bottom coil dia.	210 mm	255.6 mm	308.2 mm	380 mm

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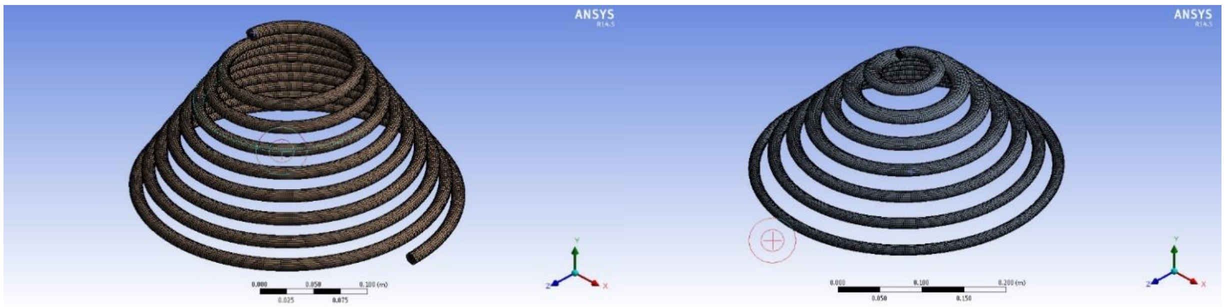


A. Boundary Conditions

No slip boundary condition is assigned on all the walls of the double pipe heat exchanger. At the inlet, hot fluid enters with a different uniform velocity in each case i.e. for flow rate of 60 lph velocity is 0.3 m/s, for 130 lph velocity is 0.65 m/s, and for 200 lph velocity is 1 m/s. For cold fluid velocity in each case is 1 m/s. Hot fluid inlet temperature is 333.2 K while inlet temperature of cold fluid is 298.15 K.

B. Turbulence Model

In this work the turbulence model selected for analysis is k-ε model. This model uses two kinds of transport equations for the analysis. First determines turbulent kinetic energy (k) while second determines the rate of dissipation of the turbulent kinetic energy (ε).

C. Solver Type

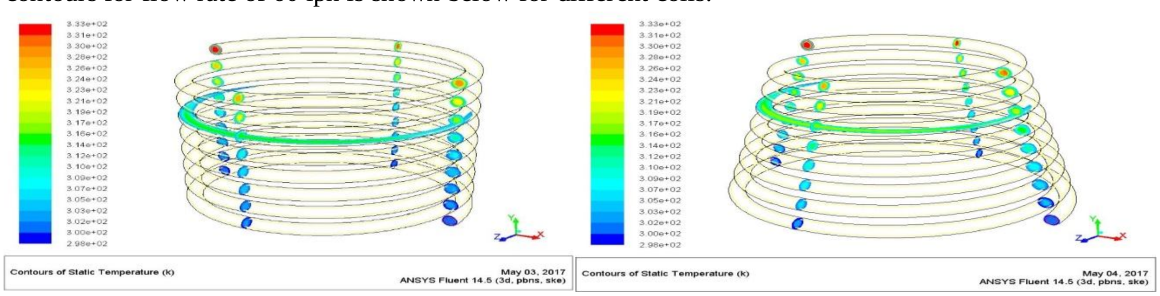
Pressure based solver has been used in the analysis.

D. Solution Method

The solution method used to solve the problem is SIMPLE. While Standard pressure option has been used along with the second order in momentum option.

IV. RESULTS AND DISCUSSION

To measure the thermal performance in a heat exchanger, heat transfer coefficient (h) & nusselt number (Nu) are the parameters which need to be analyzed. While pressure drop across the heat exchanger tells us about the power required for pumping and in the heat exchanger, friction factor (f) is the measure of the pressure drop. In the present work the heat transfer and flow characteristics of conical coil with different cone angles can be visualized from the different contour diagrams of each coil. Some of the temperature contours for flow rate of 60 lph is shown below for different coils.



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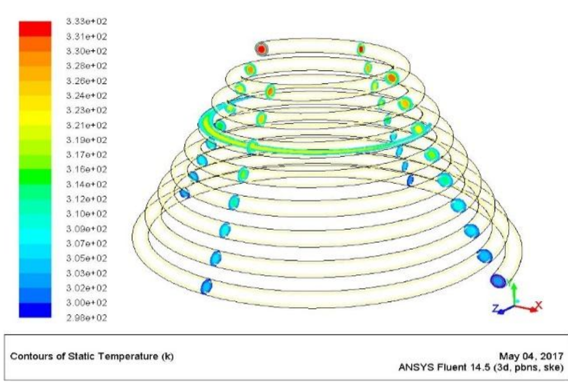
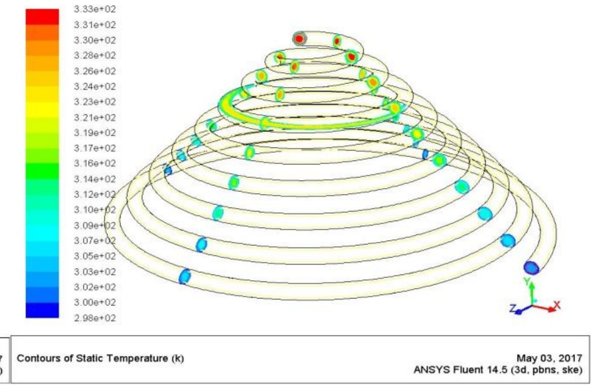
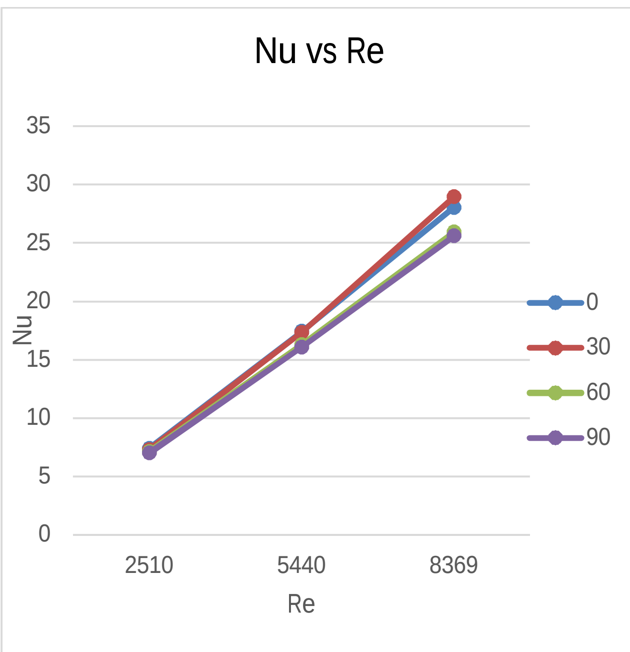
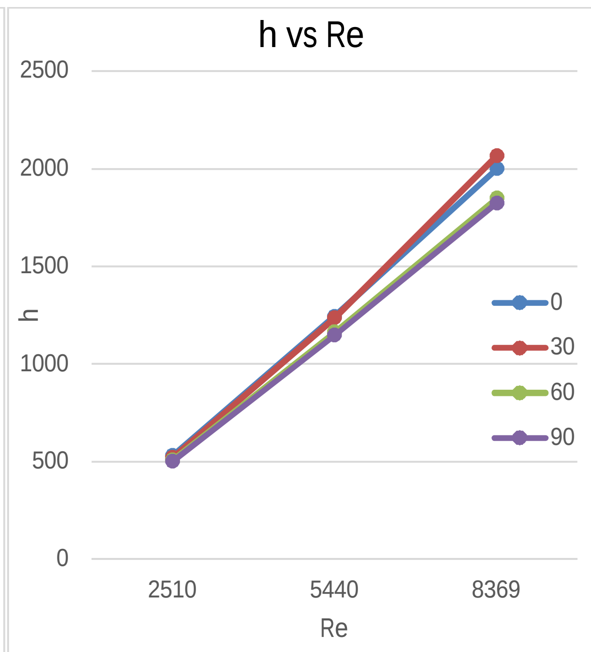




Similarly the temperature and pressure contours for different cases have been observed in the work.

Based on the study performed in the software the findings have been listed and the graphs have been plotted to get a conclusion based on the study done. Graphs have been plotted with respect to Reynolds number of the different flow rates. Here the flow rates 60 lph, 130 lph, and 200 lph corresponds to the Reynolds number 2510, 5440, and 8369 respectively. Following are the graphs plotted:

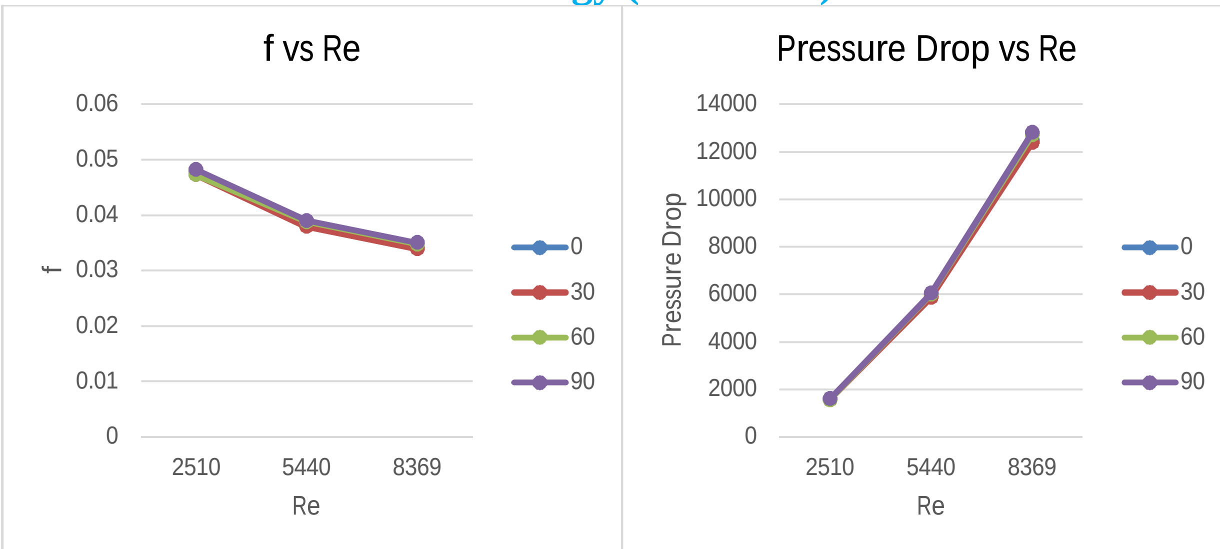




From the graph between Nu and Re, it is clear that as the Reynolds number is increasing the vertical distance between nusselt number of 30° coil and 60° coil is mainly increasing. Although up to Reynolds number 5440 simple helical coil has highest nusselt number value but after that it seems that the pattern is changing. And finally at Reynolds number 8369, 30° coil has the highest nusselt number value. The variation between nusselt numbers of coils at different angle, for the coil of 30° is increasing for greater Reynolds number while at other angles it is almost parallel lines with decreasing value of nusselt number as the cone angle of the conical coil heat exchanger is increasing.

The pattern of graph between h and Re is same as that of the above graph as here nusselt number is directly proportional to heat transfer coefficient. And all other variables on which nusselt number depends upon except heat transfer coefficient are constant as diameter of coil as well as hot and cold fluid taken in each case is similar.

Next two graphs will show the relation of friction factor and pressure drop across the heat exchanger with that of the Reynolds number for different cone angles.



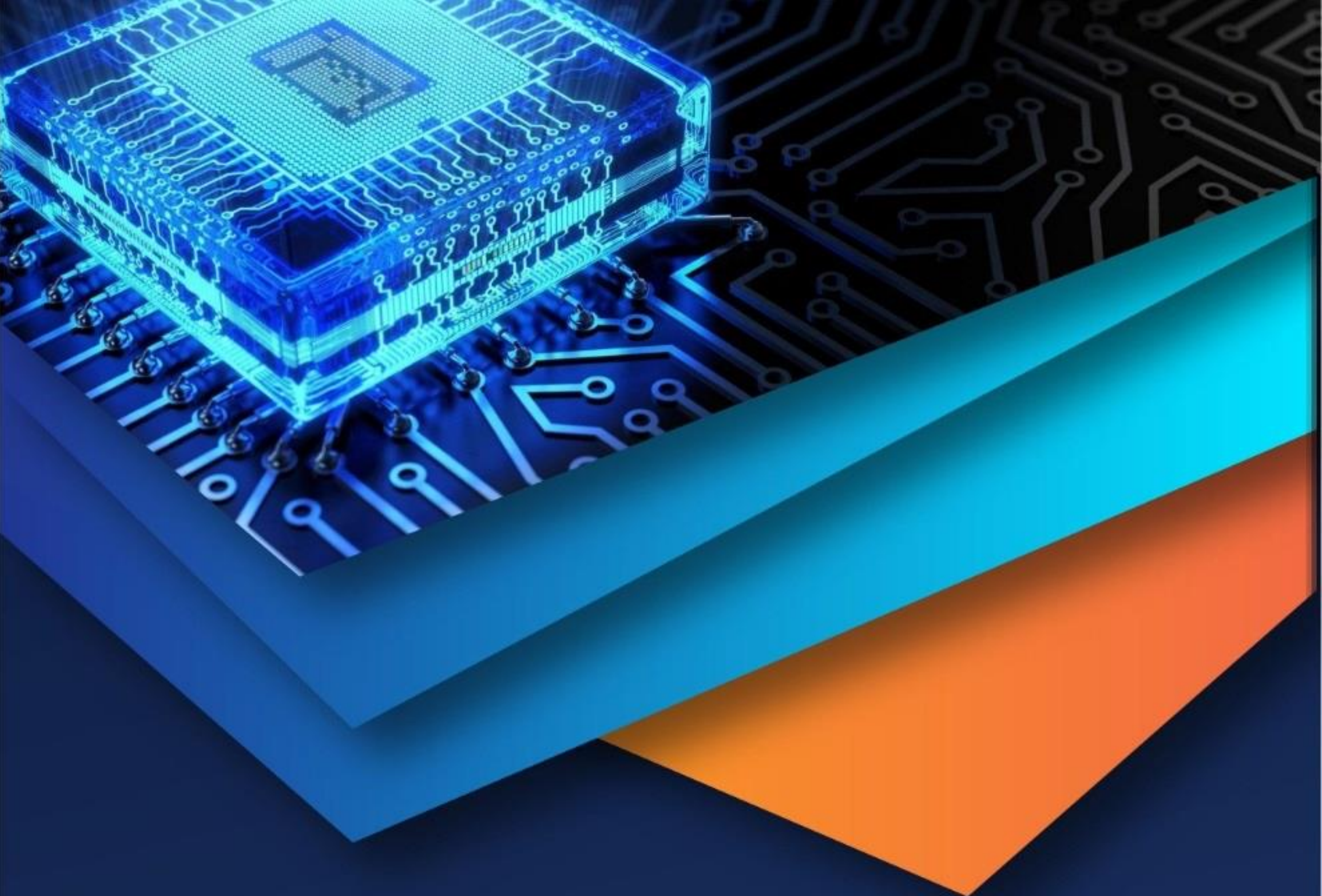
From the above graphs it is clear that as the cone angle is increasing along with it the friction factor and pressure drop is also increasing. Hence the maximum friction factor and pressure drop is for the coil of 90°. Although for the case of simple helical coil its friction factor value seems to lie between the friction factor values of 30° and 60°. So it is clear that in this study the friction factor and pressure drop comes out to be minimum for the case of 30° coil in each case. And also it is clear from the graph that as the Reynolds number is increasing friction factor value decreases while pressure drop increases.

V. CONCLUSION

From the present work it has been observed that the pressure drop and friction factor comes out to be minimum for the case of conical coil heat exchanger with cone angle 30°. While for the case of heat transfer coefficient and nusselt number it decreases as the cone angle increases hence maximum value for both parameters comes out for the case of simple helical coil. From these points it may be concluded that if we see from the perspective of heat transfer characteristics it seems that the simple helical coil heat exchanger would be a better option as compared to any of the conical coil heat exchanger discussed. But if we consider the pressure drop characteristics which gives us the idea about the power consumption of the heat exchanger it seems that the 30° conical coil heat exchanger would be the best option as compared to all other heat exchangers discussed in the present work.

REFERENCES

[1] Jeschke, D, “Heat transfer and pressure loss in coiled pipes”. Ergaenzungsheft Z. Ver. Disch. Ing., Volume 68, (1925), pp. 24-28.
[2] Seban, R.A., McLaughlin, E.F, “Heat transfer in tube coils with laminar and turbulent flow”, International Journal of Heat and Mass Transfer, Volume 6, (1963), pp. 387-395.
[3] Shah, R.K., Joshi, S.D, “Convective heat transfer in curved ducts for a natural circulation system”, Journal of Energy, Heat and Mass Transfer, Volume 18, (1987), pp. 39–46.
[4] Eustice, J, “Flow of water in curved pipes”, Proceedings of Royal Society of London, Series A 84, (1910), pp. 107-18.
[5] Grindley, J.H., Gibson, A.H, “The frictional resistance of air through a pipe”, Proceedings of Royal Society of London Series, A, 80, (108) pp. 114-139.
[6] Koutsky, J.A., Adler, R.L., “Minimization of axial dispersion by use of secondary flow in helical tubes”, The Canadian Journal of Chemical Engineering, Volume 42, (1964), pp. 239-245.
[7] Mori, Y., Nakayama, S, “Study on forced convective heat transfer in curved pipes (1st report, laminar region)”. International Journal of Heat and Mass Transfer, Volume 8, (1965), pp. 67-82.
[8] McConalogue, D.J., Srivastava, R.S, “Motion of a fluid in a curved tube”, Proceedings of the Royal Society of London, Series A, Mathematical and Physical Sciences, Volume 307(1488), (1968), pp. 37-53.
[9] Ito, H., “Friction factors for turbulent flow in curved pipes”, Journal of Basic Engineering, Transactions of the ASME, Volume 81, (1959), pp. 123-134.
[10] Nunge, R.J., Lin., T.S, “Laminar flow in strongly curved tubes”, AIChE Journal, Volume 19(6), (1973), pp.1280-1281.
[11] Austin, L.R., Seader, J.D., “Fully developed viscous flow in coiled circular pipes”, AIChE Journal, Volume 19(1), (1973), pp. 85-94.
[12] Jayakumar, J.S., Mahajani, S.M., Mandal, J.C., Iyer, K.N., Vijayan, P.K., “CFD analysis of motion on laminar-flow heat-transfer in helically-coiled tubes”, AICHE Journal, Volume 17, (2008), pp. 1142–1222.
[13] Shinde Digvijay D and Dange H. M, “Heat Transfer Analysis of a Cone Shaped Helical Coil Heat Exchanger”, IJET, Vol. 3, (2013).



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