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CFD Analysis and Optimization of Heat Transfer in Double Pipe Heat Exchanger with Helical-Tap Inserts at Annulus of Inner Pipe

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Abstract: *In heat exchanger for increasing the rate of heat transfer with helical-tape inserts have useful. Generally, helical-tape inserts causes swirl flow introduces at outside of inner tube which continuously disrupts the thermal boundary layer of fluid on the tube. This analysis has done on a single unit of pipe of heat exchanger those are used in a heat exchanger and this investigation is useful to increase the thermal characteristics of a heat exchanger. To analyze the characteristics of helical-tap inserts at annulus of inner pipe, a 3-D analytical model has been developed. From the analysis it is cleared that there are good relation between Nusselt no. and friction factor for enhancing the heat transfer. SST k-ω turbulent model is to be selected for simulation because it gives better turbulent model. From analysis, helical-tape inserts increase the heat transfer rate with expectation of pressure drop. In this work an analysis has been done on heat transfer phenomenon of helical-tap inserts at annulus of the inner pipe.*
Keywords*: Double pipe heat exchanger, Heat transfer augmentation techniques, Helical-tape insert, Pitch length, Numerical investigation, Computational fluid dynamics, Turbulence modeling, Friction factor, Nusselt number.*

I. Introduction

The procedure of designing the heat exchanger is difficult, because it required correct analysis of heat transfer rate, flow rate, drop in pressure and other factor calculation that give the result for long term established and economically preference of the equipment. The challenge is to designing a heat exchanger for better performance and makes it to compact to gain a more heat transfer. Normally a heat exchanger used in power plants, chemical plants, AC equipments, Freezer and other plants which provide or remove heat from different types of fluid. This involves huge investments annually for both operation and capital costs. It is required to reduce the overall dimensions and characteristics of heat exchanger. The need to analyze and optimize the heat exchanger to conserved and developed it.

Heat exchanger with helical-tap inserts at annulus of inner pipe, from the velocity vector, it is observed that the flow of water in plain tube is straight line but in case of helical-tap flow is swirl, swirl flow in tube causes the surface area of effected heat transfer in increases, thereby rate of heat transfer increases but pressure drop also increases because of flow abstraction. This pressure drops varies along with Reynolds no. on helical-tap. To predict the performance heat exchanger is relating to the governing parameters such as surface area for transferring heat overall heat transfer coefficient and temperature difference. Assuming there is no transfer of heat to the surrounding and negligible KE and PE.

$\mathbf{m_h}$ - mass of hot fluid entering (kg), $\mathbf{m_c}$ - mass of cold fluid entering (kg), $\mathbf{C_h}$ - Sp. heat of hot fluid entering (J/kg K), $\mathbf{C_c}$ – Sp. heat of cold fluid entering (J/kg K), $\mathbf{t_{h1}}$ - temperature of hot fluid entering (K), $\mathbf{t_{c1}}$ - temperature of cold fluid entering (K), $\mathbf{t_{h2}}$ - temperature of hot fluid exit (K), $\mathbf{t_{c2}}$ - temperature of cold fluid exit (K), d_h- hydraulic diameter.

Heat rejected by hot fluid–

$$Q_h = m_h C_h (t_{h1} - t_{h2}) \quad (1.1)$$

Heat absorbed by cold fluid

$$Q_c = m_c C_c (t_{c1} - t_{c2}) \quad (1.2)$$

Heat exchange by two fluids

$$Q = UA\theta_m \quad (1.3)$$

Where, U- Overall heat transfer coefficient, A- Effective heat transfer area, θ_m - appropriate means temperature difference across heat exchanger. As there is variation in temperature difference of hot and cold fluids point to point, so that by the concept of mean temperature difference the term Θ_m has introduced which is appropriate mean temperature difference across heat exchanger or known as log mean temperature difference.

For parallel flow log mean temperature difference is given by

$$\theta m = \frac{\theta 2 - \theta 1}{In(\theta 2/\theta 1)} \quad (1.4)$$

For counter flow log mean temperature difference is given by

$$\theta m = \frac{\theta 1 - \theta 2}{In(\theta 1/\theta 2)} \quad (1.5)$$

Where, $\theta_1 = t_{h1} - t_{c1}$, $\theta_2 = t_{h2} - t_{c2}$

Of the many enhancement techniques that can be employed, swirl flow generation by means of full-length helical-tape inserts is found to be extremely effective [1], [2]. Significant heat transfer improvement can be obtained, particularly in laminar flows. Other examples of techniques that promote swirl flows include curved ducts, tangential fluid injection, and twisted or convoluted ducts. Their thermal-hydraulic characteristics, heat transfer improvement potential, and typical applications have been outlined.

Helical-tap – we know that the heat transfer is increase considerably for flow is well mixed and stirred. Because of this principle of development for equipment to enhancement technique of heat exchanger helical-tap inserts are used at annulus of inner pipe.

Heat transfer increase because of:

a. Tap reduce the hydraulic diameter, cause affect and enhance the coefficient of heat transfer.
b. The helical-tap causes a tangential velocity component hence speed of flow of fluid is increase near the wall surface.
c. There may be heat transfer from the tape.

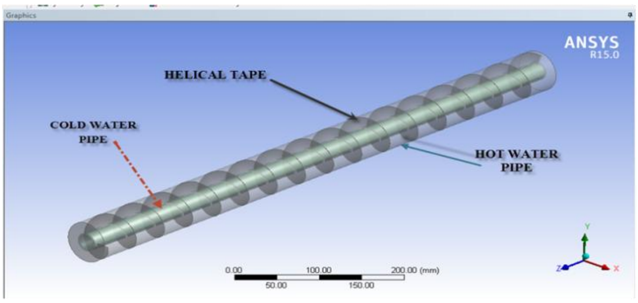

Figure 1: CFD model of helical tape at annulus of inner pipe

Figure 1.1 shown of helical-tap, is described by the helical twisting nature of tap providing the fluid longer flow region or greater time for transferring heat, the helical force for bulk flow are forcing for generation a secondary circulation because of well mixed swirl flow increase the convective heat transfer[5].

The swirl flow which is in fully developed nature, performance of helical-tap and functional relation are given as below

$$f = \emptyset(Re, y, \delta/d) \quad (1.6)$$

$$Nu = \emptyset(Re, y, \delta/d\ , Pr) \quad (1.7)$$

Based on a fundamental balance between inertia, viscous, and tape-geometry helical curvature induced forces, it is proposed that tape-induced swirl flows can be scaled by a swirl parameter defined as [1]

$$Sw = Re/\sqrt{y} \quad (1.8)$$

Where, the swirl Reynolds number is based on the swirl velocity, and

$$Re_s = \rho V_s d/\mu \quad (1.9)$$

II. Numerical Investigation Of Heat Exchanger With Helical Tape Inserts At Annulus Of Inner Pipe

For analysis of our work the input data and boundary condition will be taken in [1] which is experimentally investigate the heat transfer phenomenon.

Table I Input data for double pipe heat exchanger

Length of tube	2.2 m
Inner diameter of inner pipe, d_i	0.022 m
Outer diameter of inner pipe, d_o	0.026 m
Inner diameter of outer pipe, D_i	0.054 m
Outer diameter of outer pipe, d_o	0.058 m
Material of pipe	Copper
Inner pipe fluid	Cold water(300K)
Annulus fluid	Hot water(353k)

Table II Properties of water

Density, ρ	998.2 kg/m^3
Specific Heat Capacity, C_p	4182 J/kg K
Thermal Conductivity, k	0.6 W/m K
Viscosity, μ	1.003×10^{-3} kg/m s

Table III Boundary condition for inner fluid

Inlet condition	Velocity Inlet (0.376 m/s)
Outlet condition	Pressure Outlet
Inlet Temperature	300 K

Table IV Boundary condition for annulus fluid

Inlet condition	Velocity Inlet (Varies from 0.127 to 0.557 m/s)
Outlet condition	Pressure Outlet
Inlet Temperature	353

III. Data Reduction Equations

The area weighted average temperature and static pressure were noted at the inlet and outlet surfaces of the pipe. After setting all the above mentioned parameters simulation has been done and after simulation Friction Coefficients for all the mentioned or considered Reynolds Number have been calculated as per the following relation and verified with the findings of Patnala shankra Rao [1].

The correlation for the Calculation of 'Friction Factor (*f*)' Colburn's Equation [1] is:

$f = 0.046({R_L}^{-0.2})$…………………………………… (4.1)

Reynolds No. R_L is:

$$R_L = \frac{\rho V d}{\mu} \quad (4.2)$$

Heat transfer rate is given by:

$$Q_1 = m \times C_P \times (T_{co} - T_{ci}) \quad (4.3)$$

$$Q_2 = m \times C_P \times (T_{hi} - T_{ho}) \quad (4.4)$$

$$Q_{Avg} = \frac{Q_1 + Q_2}{2} \quad (4.5)$$

Area:

$$A = \pi \times d_h \times L \quad (4.6)$$

LMTD is given by:

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln\left(\frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}}\right)} \quad (4.7)$$

Convective heat transfer coefficient is:

$$h = \frac{Q}{A \times LMTD} \quad (4.8)$$

Nusselt No. is given by:

$$N_u = \frac{h \times d_h}{k} \quad (4.9)$$

IV. Result And Discussion

The analysis of heat exchanger is done by the using different pitch length of 50, 100, 150, 200 and 250 mm for 40 design points. This analysis is for hot water at velocity range 0.127 to 0.577 m/s and for cold water at constant velocity 0.367 m/s. The inlet temperature of cold water and hot water is respectively 300 K and 353 K [1].

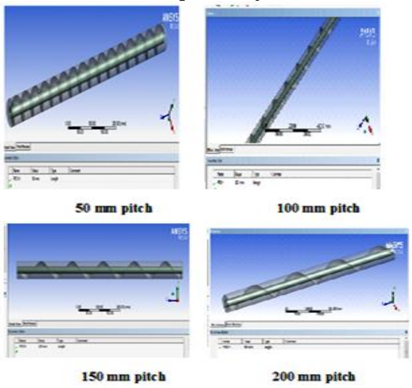

Figure 2: CFD Model for heat exchanger with helical tap of different pitch length inserts.

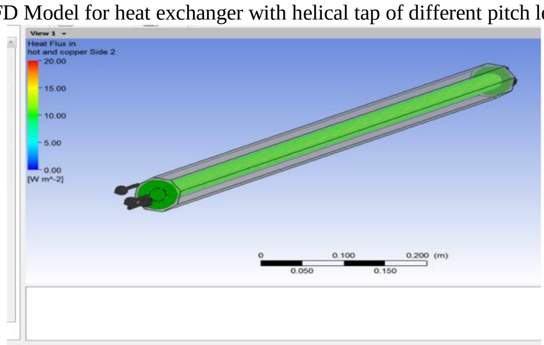

Figure 3: Inlet velocity of hot domin

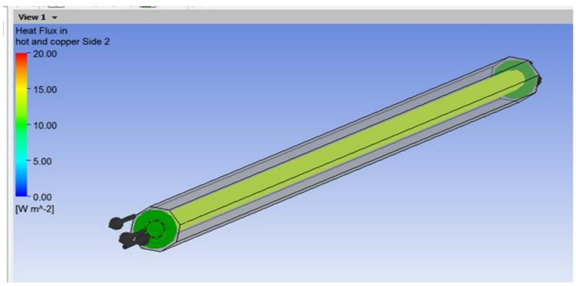

Figure 4: Inlet velocity of cold domin

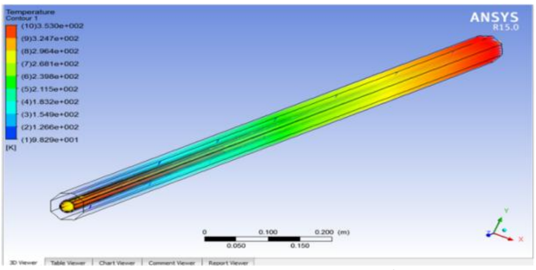

Figure 5: Average outlet temperature of hot water

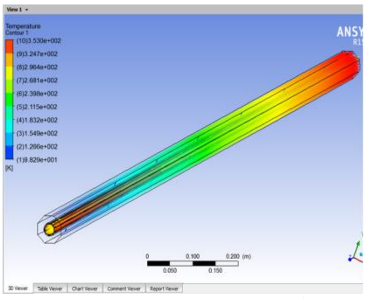

Figure 6: Average outlet temperature of cold water

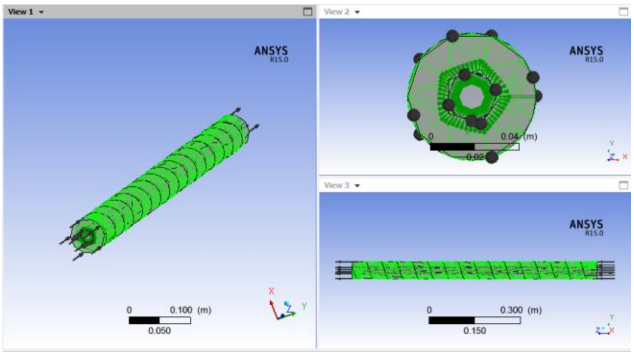

Figure 7: Meshed view of helical tape

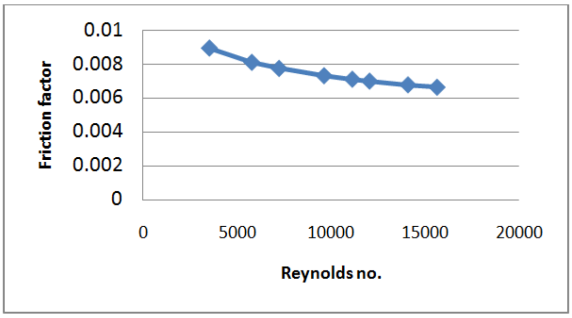

Figure 8: Friction Factor Vs Reynolds No. with helical tape

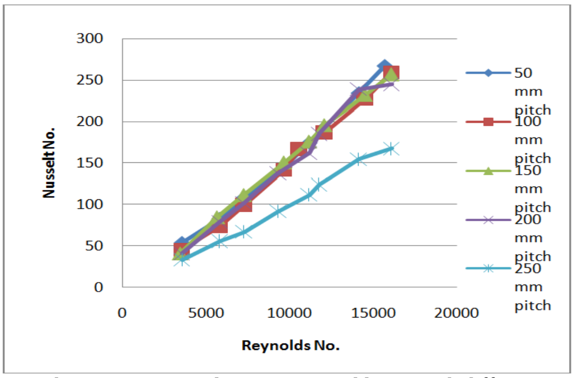

Figure 9: Graph represents Nusselt No. Vs Reynolds No. with different pitch length

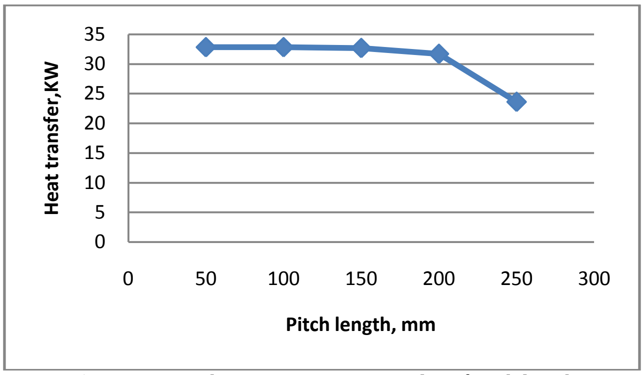

Figure 10: Graph represents optimum value of pitch length

It is cleared from the above result heat transfer and heat transfer rate are increases with decrease the pitch length. The above result is performed for the different pitch length and plot the comparison of Reynolds No. and Nusselt Number and Reynolds No. and Friction factor. It is cleared that the Friction factor decrease with increase in Reynolds Number and Nusselt No. increase with Increase in Reynolds Number. From the above result is also optimizing the helical tape pitch length for different velocity. The maximum heat transfer achieved at minimum pitch length which is 50 mm and maximum velocity of 0.557 m/s. The present research also predicts that by increasing the mass flow rate of fluids, there is miner variation in heat transfer rate.

V. Conclusion

It is clear that insertion of a helical tape in a plain tube increase the thermal performance of the tube and furthermore if the pitch length of helical tape is reduce increase in surface it increases the tube's thermal performance more. Area Weighted Average of the fluid's temperature at the outlet of the tube has been increased due to the insertion of a helical tape in the tube. The reason for the increment of these parameters is that, due to the insertion of a helical tape a swirl flow is created in the pipe which helps the fluid to take more and more heat from the tube wall.

So it may be concluded that, modifications should be done in such a way so that average temperature as well as heat flux both increases. To do this, optimization procedure may be adopted to optimize different parameters to achieve the desired goal.

Scope for future work: Further detailed studies can be carried out in this area either through experiments or with the aid of software. Nusselt number and friction factor values can be obtained for helical tap with the same pitch at different velocity and similarly for helical tap with the same velocity and different pitch in order to study the effect of helical tap pitch length on Nusselt number and friction factor. Some other inserts may be used and similar investigations can be done and the values compared to those of helical tap inserts.

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Appendix

Table V Numerical result at 50mm pitch

S No.	Velocity of annulus fluid (m/s)	Average heat transfer KW	Convective heat transfer coefficient h (KW/m^2K)	Reynolds No. R_L	Friction factor *F*	Nusselt No. N_u
1	0.127	7.04181	1.15676	3538.98	0.00897	53.98
2	0.208	11.10660	1.78283	5796.12	0.00813	83.19
3	0.26	13.98420	2.32348	7245.16	0.00777	108.42
4	0.346	18.89485	3.11277	9641.63	0.00734	145.26
5	0.4	22.24153	3.75048	11146.40	0.00713	175.02
6	0.433	24.34318	4.14922	12065.97	0.00702	193.63
7	0.52	29.22962	5.02292	14110.47	0.00680	234.40
8	0.577	32.82157	5.72580	15657.20	0.00666	267.20

Table VI Numerical result at 100mm pitch

S No.	Velocity of annulus fluid (m/s)	Average heat transfer KW	Convective heat transfer coefficient h (KW/m^{2}K)	Reynolds No. R_L	Friction factor f	Nusselt No. N_u
1	0.127	6.33410	0.97627	3538.98	0.00897	45.559
2	0.208	10.3044	1.58114	5796.12	0.00813	73.786
3	0.26	13.5127	2.13015	7245.16	0.00777	99.407
4	0.346	18.9805	3.04332	9641.63	0.00734	142.021
5	0.4	21.8896	3.57807	9952.14	0.00729	166.976
6	0.433	24.1505	4.00747	12065.97	0.00702	187.015
7	0.52	29.1781	4.87827	14490.32	0.00676	227.65
8	0.577	32.8194	5.55982	16078.68	0.00663	259.46

Table VII Numerical result at 150mm pitch

S No.	Velocity of annulus fluid (m/s)	Average heat transfer KW	Convective heat transfer coefficient h (KW/m^{2}K)	Reynolds No. R_L	Friction factor F	Nusselt No. N_u
1	0.127	5.94475	0.8868	3538.98	0.00902	41.384
2	0.208	11.2788	1.8226	5796.12	0.00817	85.054
3	0.26	14.47137	2.3996	7245.16	0.00777	111.98
4	0.346	19.472	3.2434	9641.63	0.00734	151.35
5	0.4	22.6103	3.7846	11146.4	0.00713	176.61
6	0.433	24.9284	4.2096	12065.97	0.00702	196.44
7	0.52	29.4910	4.9762	14490.32	0.00676	232.22
8	0.577	32.6799	5.5098	16078.68	0.00662	257.12

Table VIII Numerical result at 200mm pitch

S No.	Velocity of annulus fluid (m/s)	Average heat transfer KW	Convective heat transfer coefficient h (KW/m^{2}K)	Reynolds No. R_L	Friction factor f	Nusselt No. N_u
1	0.127	6.5382	1.0267	3538.98	0.00897	40.91
2	0.208	10.7652	1.6922	5796.12	0.00813	78.96
3	0.26	13.7418	2.1890	7245.16	0.00777	102.15
4	0.346	18.4353	2.9554	9294.83	0.00739	137.91
5	0.4	21.5496	3.4813	11146.40	0.00713	162.46
6	0.433	24.0553	3.9819	11749.68	0.00705	185.82
7	0.52	29.9605	5.1237	14110.47	0.00680	239.10
8	0.577	31.6990	5.2577	16078.68	0.00662	245.35

Table IX Numerical result at 250mm pitch

S No.	Velocity of annulus fluid (m/s)	Average heat transfer KW	Convective heat transfer coefficient h (KW/m^{2}K)	Reynolds No. R_L	Friction factor f	Nusselt No. N_u
1	0.127	4.7826	0.7153	3538.98	0.00897	33.38
2	0.208	7.5287	1.1951	5796.12	0.00813	55.77
3	0.26	9.6671	1.4319	7245.16	0.00777	66.82
4	0.346	13.1439	1.9699	9294.83	0.00734	91.92
5	0.4	15.6422	2.3993	11146.40	0.00713	111.96
6	0.433	17.2848	2.6673	11749.68	0.00702	124.47
7	0.52	21.2171	3.3257	14110.47	0.00680	155.19
8	0.577	23.5976	3.5927	16078.68	0.00666	167.65