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Research Article

Experimental Prediction of Helical Coil Heat Exchanger

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Abstract

Enhancements of heat transfer in helically coil have researched by many researchers. So many literatures are available on characteristics of heat transfer coefficient on helically coils. Basically this paper focus on experiment prediction of fluid heat transfer enhancement of varying pitch of helical coil. The technique of experiment investigation of a helical coils tubes heat exchanger, the effect of the annulus tubes at steady value of mass flow rate in Reynolds and Dean Number and as well established the surface of coefficient of heat transfer. While taking various boundary conditions on inside tube helical coils for finding characteristics of heat transfer. Hence, considering different mass flow inside and outside of helically coil heat exchanger. The fabrication of experimental setup is estimate the heat transfer enhancement in inside helical coil tubes.

Keywords: *Helical Coil Tubes Heat Exchanger, Coil Configuration, Flow Parameters (Parallel and Counter), Dean Number.*

1. Introduction

Heat exchanger is a device which is used to transfer heat between two fluids which may be in direct contact or may flow separately in two tubes or channels. We find numerous applications of heat exchangers in day today life. For example condensers and evaporators used in refrigerators and air conditioners. In thermal power plant heat exchangers are used in boilers, condensers, air coolers and chilling towers etc. Similarly the heat exchangers used in automobile industries are in the form of radiators and oil coolers in engines. Heat exchangers are also used in large scale in chemical and process industries for transferring the heat between two fluids which are at a single or two states.

Working towards the goal of saving energies and to make compact the design for mechanical and chemical devices and plants, the enhancement of heat transfer is one of the key factors in design of heat exchangers.

Helical coils are very alluring for many processes in heat exchanger and reactors because of it accommodate higher heat transfer rate in small space. The fluid motion in curved pipe was observed by Eustice in 1911.Since then numerous studies have been reported on the flow fields that arise in curved pipes (Dean, White, Hawthorne, Horlock, Barua, Austin and Seader) including double pipe helical coils, which is a subset of curved. Jayakumar *et al*was investigated in helical coil tubes at numerous process parameters. Mohamed A. Abd Raboh *et al*. carried out on experimental work for condensation overall heat transfer inside double pipe helical coil. Pablo Coronel *et al* have been made known on the helically coiled heat exchanger is higher than that in straight tubular heat exchanger. Rahul Kharat *et al* determine the heat transfer coefficient correlation for concentric helical coil tubes heat exchanger. Ashok Reddy *et. al*.studies on the effect of dean number over heat transfer coefficient in an agitated vessel. Timoty *et al* worked on experimental studies of double pipe heat exchanger

2. Problem Formulation

In the literature review, found that so many worked has been done to enhancement of the heat transfer coefficient in heat exchanger. But there is no work has been done to optimize the heat transfer rate with respect to Geometrical specification. In this paper work, I optimize the given helical coil heat exchanger keeping in mind that it should produce maximum heat transfer rate with variation of specification Because some times in the process of improving the heat transfer coefficient.

2.1 Problem Specification

In my study I consider the double tube helical coil heat exchanger or tube in tube helical coil heat exchanger with two different pitches. For simplification in experimental analysis I consider.

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2.2 Helical coil geometry and parameters

The geometry of the helical tube is as follows, Experimental Setup is shown in Fig.1; The bottom radius of curvature is denoted (R), the pipe diameter (a), the helical pitch as (P), the straight height (h) and For a straight helical coil the height (h) will be equal to the height of the coil (I) will change in accordance to that angle, while keeping (h) constant.

Figure1: Front view of helical coil heat exchanger showing different helical coil

Table 1 Coil Dimension Parameter

S. No.	Dimensional Parameters	Dimension
1	Inner Coil Diameter	16 mm
2	Outer Coil Diameter	25.4 mm
3	Pipe Thickness	1 mm
4	Tube Pitch	35 mm and 40 mm
5	Pitch Circle Diameter	200 mm

After manufacturing helical coil tubes and doing the experiments on their setup the problem was analyzed, for different boundary conditions as specified later. For analysis of the problem was to be taken in laminar flow fluid flow condition was considered.

2.3 Formulation

In this experimental work heat transfer coefficients and heat transfer rates were based on the measured on temperature data.

The overall heat transfer coefficient, U_O,

$$U_o = \frac{q}{A_o T_{LMTD}} \quad (1)$$

Hot Water Heat transfer Rate

$$Q_h = \left(m_{h,f}\ x\ C_p\ x\ (T_{h,i} - T_{h,o})\right) \quad (2)$$

Cold Water Heat transfer Rate,

$$Q_c = \left(m_{c,f}\ x\ C_p\ x\ (T_{c,o} - T_{c,i})\right) \quad (3)$$

Mean Heat Transfer Rate , $Q_{mean} = \frac{Q_h + Q_c}{2}$ (4)

The physical properties of water are taken at average temperature; $T_{mean} = \frac{T_i + T_o}{2}$ (5)

LMTD is the log mean temperature difference, based on the inlet temperature difference, (ΔT_1), and the outlet temperature difference, (ΔT_2), using the following equation (White, 1984):

For Parallel flows Condition:

$$T_{LMTD} = \frac{(\Delta T_1)-(\Delta T_2)}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{(T_{h,i}-T_{c,i})-(T_{h,o}-T_{c,o})}{ln\left(\frac{T_{h,i}-T_{c,i}}{T_{h,o}-T_{c,o}}\right)} \quad (6)$$

For Counter-flow Condition:

$$T_{LMTD} = \frac{(\Delta T_1)-(\Delta T_2)}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} = \frac{(T_{h,i}-T_{c,o})-(T_{h,o}-T_{c,i})}{ln\left(\frac{T_{h,i}-T_{c,o}}{T_{h,o}-T_{c,i}}\right)} \quad (7)$$

Reynolds number (Re); $Re = \frac{\rho v d}{\mu}$ (8)

For Dean Number; $De = Re\sqrt{\frac{d_i}{D_h}}$ (9)

Where di is inner tubes diameter, Dh is Helix diameter.

3. Experimental Procedure

Flow rates in the annulus and in the inner tube were varied. The following four levels were used: 0.0028, 0.0056, 0.0084 and 0.0112 kg/sec. All possible combinations of these flow rates in both the inner and annulus tube were tested. Testing is to be done for both coils configuration, and under both flow condition like parallel flow and counter-flow configurations.

Furthermore, every combination of flow rate is to be done under three replicates, coil size and configuration. This resulted in a total of 150 trials. Every ten seconds, temperature data was recorded, after the system had stabilized. Temperature measurements from the 60 seconds of the stable system were used, with temperature reading fluctuations within +/- 0.1˚C. Though the type-K (error of 2.2˚C) thermocouples had limits, when placed in a common water solution the readings at steady state were all. All the thermocouples were constructed from the same roll of thermocouple wire, and hence the repeatability of temperature readings was high.

Figure: 2 Experimental Setup

4. Results and Discussion

In Figure 3, figure 4, figure 5 and figure 6 are presented Overall heat transfer coefficients (OHTC) for parallel and counter flow for the two different pitch helical coil heat exchanger results. The overall heat transfer coefficient (OHTC) is drawing the graph against the inner Dean number(N_{De}) for all flow rates of the annulus. A fluid-to-fluid helically coils heat exchanger with overall heat transfer coefficient(OHTC) increases in inner as well as annulus flow rates. Figure shows that in the annulus and inner flow rate, increasing the overall heat transfer coefficient of the helically coils in high pitch coil. This figure is define the value of overall heat transfer rate with increasing the Inner dean number, the experimental results we can see in table no.-02, table no.-04. Table no-02 and 04 is denoted the parallel flow condition of 35 mm and 40 mm pitch and table no.-03 and table no.-05 are denoted the counter flow condition of the 30 mm and 40 mm pitch of the helical coil heat exchanger.

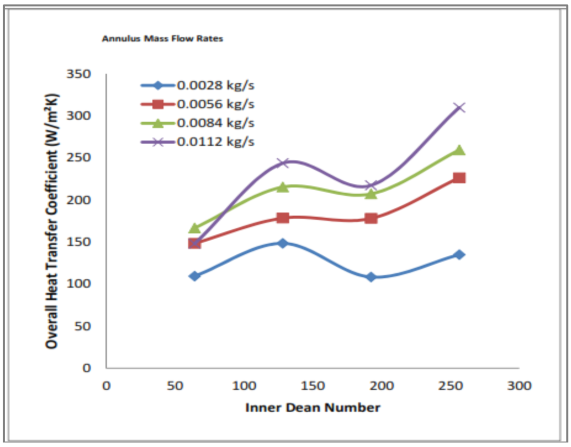

Figure: 3 Overall heat transfer coefficient (OHTC) versus the inner Dean number (N_{De}) for the small pitch (35 mm) of each annulus mass flow rate in parallel flow.

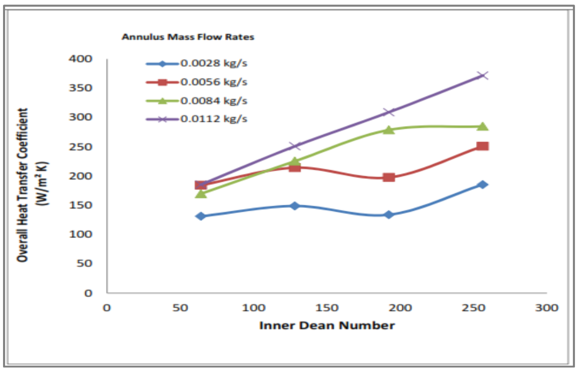

Figure: 4 Overall heat transfer coefficient (OHTC) versus the inner Dean number (N_{De}) for the large pitch (40 mm) of each annulus mass flow rate in parallel flow.

The results come from the experiments to shows that the value of counter-flow configuration has similar to the parallel flow configuration, as is expected, if changing the flow configuration should have negligible effect on the overall heat transfer coefficients. Due to the increased LMTD, heat transfer rates, are much higher in the counter-flow configuration.

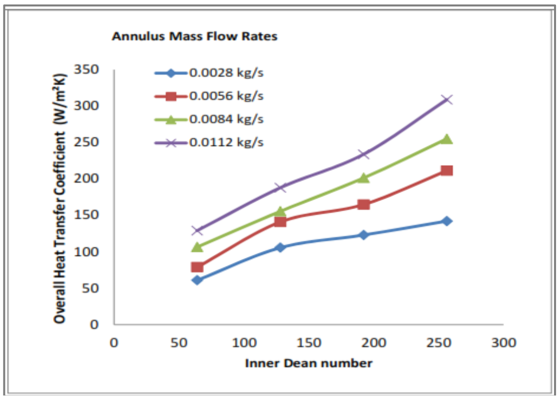

Figure: 5 Overall heat transfer coefficient (OHTC) versus the inner Dean number(N_{De}) for the large pitch (35 mm) of each annulus mass flow rate in Counter flow

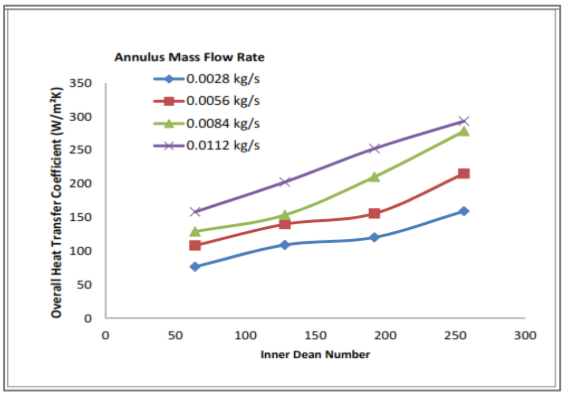

Figure: 6 Overall heat transfer coefficient versus the inner Dean number for the large pitch (40 mm) of each annulus mass flow rate in Counter flow

From figure-7 and figure-8, has to be plotted graph between counter flow verses the parallel flow overall heat transfer coefficients, where the values plotted against each other are from the same experimental parameters. There is a reasonable agreement between the two values.

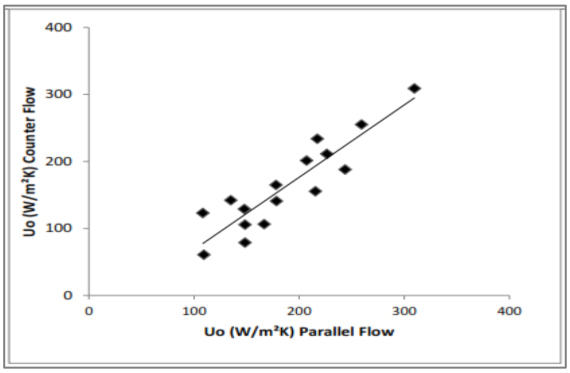

Figure: 7 Graph between counter flow verses the parallel flow

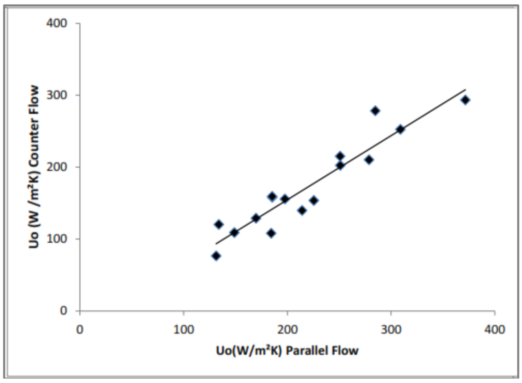

Figure 8: Counter flow overall heat transfer coefficient (OHTC)versus parallel flow overall heat transfer coefficient(OHTC) for all trails.(Pitch = 40 mm)

The inner Nusselt numbers are presented in Figure 9 and figure 10 (with ± 2 standard errors). These values are the inner Nusselt number at each Dean number (parallel flow and counter-flow values).

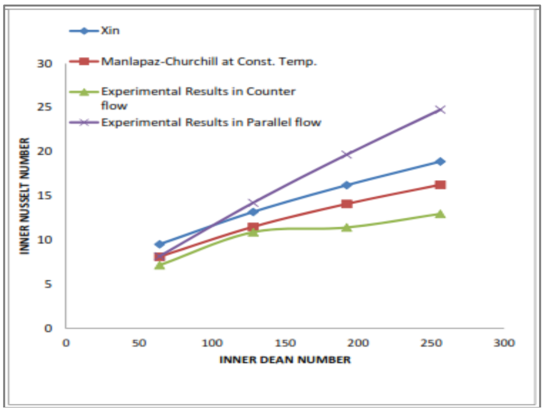

Figure: 9. Nusselt Number(Nu) verses Inner Dean Number(N_{De}) of small coil Pitch (Parallel and Counter Flow)

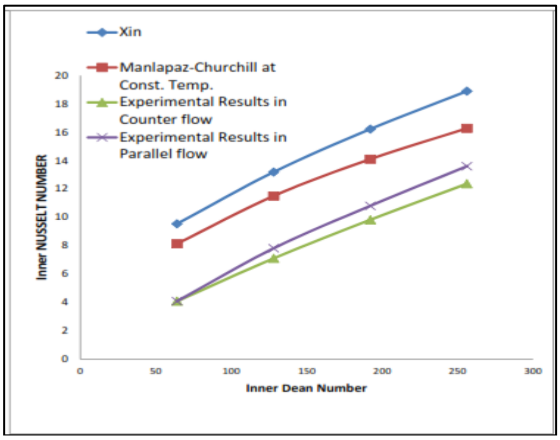

Figure:10 Nusselt Number(Nu) verses Inner Dean Number (N_{De})of Large coil Pitch (Parallel and Counter Flow)

Conclusions

An experimental study of a double pipe helically heat exchanger was performed using two differently sized pitch heat exchangers. The mass flow rates in the inner tube and in the annulus were both varied, as well as both parallel flow and counter-flow configurations were tested.

There were little differences between the overall heat transfer coefficients for the parallel flow and counter-flow configurations.

However, heat transfer rates were much higher in the counter-flow configuration due to the larger average temperature difference between the two fluids.

In table-2 and table -4 have shown the parallel flow condition, very slightly differences in both pitch condition of the helical coil heat exchanger. But in case of counter flow condition in both pitch differences have more as comparing to parallel flow conditions.

The Nusselt number in the inner tube was compared to the Manlapaz- Churchill correlation (1981), Xin and M. R. Salimpour, small coil pitch and large coil pitch values, further work needs to be done to quantify this effect.

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Appendix

Table 2 Experimental results in Parallel flow of Pitch = 35 mm

mh (kg/s)	mc(kg/s)	LMTD	Qav.(W)	U (W)
0.0028	0.0028	11.166649	245.9016	109.3442566
	0.0056	11.166649	333.7236	148.3957768
	0.0084	11.166649	374.7072	166.6198195
	0.0112	13.363524	398.1264	147.9304028
0.0056	0.0028	11.166649	333.7236	148.3957768
	0.0056	13.363524	480.0936	178.3866622
	0.0084	13.363524	579.6252	215.3692629
	0.0112	13.363524	655.7376	243.6500752
0.0084	0.0028	15.022979	327.8688	108.3681135
	0.0056	15.022979	538.6416	178.0333293
	0.0084	16.421704	685.0116	207.127217
	0.0112	17.659944	772.8336	217.2972072
0.0112	0.0028	13.363524	362.9976	134.8777202
	0.0056	13.363524	608.8992	226.2464984
	0.0084	15.022979	784.5432	259.3094145
	0.0112	15.022979	936.768	309.6231814

Table 3 Experimental results Counter flow of Pitch = 35 mm

mh (kg/s)	mc(kg/s)	LMTD	Qav.(KW)	U (KW)
0.0028	0.0028	20.9841174	257.6112	60.9739206
	0.0056	18.7550355	345.4332	78.86187033
	0.0084	18.1952605	433.2552	106.5396575
	0.0112	17.4422544	491.8032	128.985105
0.0056	0.0028	18.7159013	398.1264	105.6526312
	0.0056	19.39269	550.3512	140.9522292
	0.0084	19.9983152	626.4636	155.5867183
	0.0112	18.5681758	702.576	187.9291533
0.0084	0.0028	18.1952605	450.8196	123.0593446
	0.0056	19.9331546	661.5924	164.8483328
	0.0084	19.9331546	807.9624	201.3192029
	0.0112	19.1547945	901.6392	233.7897012
0.0112	0.0028	16.9798632	485.9484	142.14319
	0.0056	18.7159013	796.2528	211.3052624
	0.0084	18.929616	971.8968	255.0048469
	0.0112	18.4593881	1147.5408	308.7599208

Table 4 Experimental results in Parallel flow of Pitch = 40 mm

Mh (kg/s)	Mc(kg/s)	Lmtd	Qav.(w)	U (w)
0.0028	0.0028	11.167	245.902	131.2471
	0.0056	11.167	345.433	184.371
	0.0084	13.364	380.562	169.7289
	0.0112	13.364	415.691	185.3962
0.0056	0.0028	13.364	333.724	148.8392
	0.0056	13.364	480.094	214.1195
	0.0084	15.023	567.916	225.3093
	0.0112	15.023	632.318	250.8598
0.0084	0.0028	16.422	368.852	133.8708
	0.0056	16.422	544.496	197.6187
	0.0084	15.023	702.576	278.7331
	0.0112	15.023	778.688	308.9292
0.0112	0.0028	13.364	415.691	185.3962
	0.0056	15.023	632.318	250.8598
	0.0084	16.422	784.543	284.741
	0.0112	15.023	936.768	371.6442

Table 5 Experimental results Counter flow of Pitch = 40 mm

Mh (kg/s)	Mc(kg/s)	Lmtd	Qav.(w)	U (w)
0.0028	0.0028	20.49593431	263.466	76.61400937
	0.0056	19.99831516	362.9976	108.1836621
	0.0084	18.13148967	392.2716	128.9450613
	0.0112	16.76773081	444.9648	158.1621997
0.0056	0.0028	20.16639053	368.8524	109.012368
	0.0056	21.46507088	503.5128	139.8071947
	0.0084	22.04447843	567.9156	153.5448673
	0.0112	19.64442002	667.4472	202.5018696
0.0084	0.0028	20.5976476	415.6908	120.2829643
	0.0056	22.40710059	585.48	155.7319542
	0.0084	21.4027489	755.2692	210.3214422
	0.0112	20.59699077	872.3652	252.4328621
0.0112	0.0028	17.31234049	462.5292	159.2335962
	0.0056	19.79355214	714.2856	215.0797329
	0.0084	19.93315462	930.9132	278.3456805
	0.0112	20.93635363	1030.4448	293.3425209