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"AN EXPERIMENTAL AND CFD INVESTIGATION OF A FABRICATED SHELL AND HELICAL TUBE HEAT EXCHANGER"

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ABSTRACT

The performance has been investigated experimentally as well as on the basis of computational analysis. Experimental work has been performed on the fabricated shell and helical tube heat exchanger in the departmental laboratory with exhaust gas from a 15 HP horizontal diesel engine as the shell side fluid and water from the overhead tank as tube side fluid. The results obtained from the computational and experimental analysis have been compared. The effectiveness of helical tube heat exchanger that was obtained from CFD and Experimental analysis are being compared with the effectiveness of the heat exchanger for counter flow for a capacity ratio of 0.16. It can be observed from CFD and Experimental results are in good agreement within some experimental error limits.

Key Words Diesel engine, effectiveness, Experimental, exhaust gas, Heat Exchanger.

I. INTRODUCTION

The core of a heat recovery system is the heat exchanger. Decreasing size and increasing heat load is the typical feature of the modern day Heat Exchanger. Helical coils have more heat transfer surface due to their compact configuration. Compact heat exchanger can be used for heat recovery purpose because of its several advantages over conventional type. Compact heat exchangers are characterized by having a high area density, which means a high ratio of heat transfer surface to heat exchanger volume. Helical tube heat exchanger comes under the category of compact heat exchanger due to its compact configuration. The waste gas from various sources at different temperature ranges are usually dumped into the environment which could otherwise be used for some useful purposes. Compact heat exchangers are primarily used in gas-flow systems where the overall heat transfer coefficients are low and it is desirable to achieve a large surface area in a small volume. Compact heat exchangers offer a high surface area to volume ratio typically greater than 700 m2/m3 for gas-gas applications, and greater than 400 m2/m3 for liquid-gas applications. They are often used in applications where space is usually a premium such as in aircraft and automotive applications. They rely heavily on the use of extended surfaces to increase the overall surface area while keeping size to a minimum. As a result, pressure drops can be high. Typical applications include gas-to-gas and gas-to-liquid heat exchangers. They are widely used as oil coolers, automotive radiators, intercoolers, cryogenics, and electronics cooling applications.

Rating and sizing are two important problems in the thermal analysis of heat exchanger. The rating problem is concerned with the determination of heat transfer rate, the fluid outlet temperature and pressure drop whereas the sizing problem is concerned with the determination of the matrix of dimension to meet the specified heat transfer and pressure drop requirement.

The performance of the exchanger has been tested not only as simulated experimental exchanger but also as a waste

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heat recovery device for a 60 HP gas turbine. In one of the paper carried out by J.S. Jayakumar et al. on "Experimental and CFD estimation of Heat Transfer in helically coiled Heat Exchanger", they made an attempt to find out the boundary condition for proper modeling considering different boundary conditions. They found that constant temperature or constant heat flux boundary conditions does not yield proper modeling. Hence, the heat exchanger was analyzed considering conjugate heat transfer. The CFD analysis was made using FLUENT . The experimental and CFD results were compared and based on the experimental results, a correlation was developed to calculate the inner tube heat transfer coefficient of the helical coil. J.S. Jayakumar et al. Made an investigation on "CFD analysis of single phase flows through helical coil". Here, they made an attempt to see the outcome by varying the coil pitch, pipe diameter and pitch circle diameter using the CFD package FLUENT. It was observed that when the coil pitch is zero, local Nusselt number at the top and bottom points on the periphery of a cross section are almost the same. For this case, only centrifugal but no torsonal force is acting on the fluid. As we increase the pitch, torsonal or rotational forces comes into effect. When the pipe diameter is small, the secondary flows are weaker and hence mixing is lesser. This produces nearly the same heat transfer in the upper half cross section in a given plane. When the pitch coil diameter is more, the effect of coil curvature on flow decreases and hence centrifugal force plays a lesser role in flow characteristic.

In another paper on "Development of Heat transfer coefficient correlation for concentric helical coil heat exchanger", by Rahul Kharat, Nitin Bhardwaj and R.S. Jha, improved heat transfer coefficient correlation was developed for the flue gas side of heat exchanger from experimental and CFD data. Also the effect of different functional dependent variable such as gap between the concentric coil, tube diameter and coil diameter which affects the heat transfer were analyzed.

Based on the above mentioned comprehensive literature review, it can be concluded that the geometry of a helical tube is the main concern in order to obtain increasing heat load which is the first priority in the modern day heat exchanger. The parameters that affect the heat transfer coefficient are coil to tube diameter ratio, pitch of the coil and coil diameter. So, while doing an analysis, these parameters need to be taken into account with the aim of achieving higher heat transfer coefficient.

II. METHODOLOGY

To achieve the above mentioned objectives, the following methodologies are being adopted.

- **a)** The model used for the computational study is a three dimensional model of a helical tube heat exchanger. All geometries were generated using ANSYS .
- **b)** The flow arrangement that was considered in the problem was a cross counter flow configuration.
- **c)** The tube volume was split from the shell volume in order to generate hollow area corresponding to interior of tubes.
- **d)** Once the geometry is complete, mesh is generated. Due to highly irregular nature of the tube and shell side volume, unstructured grid was generated. The scheme selected for meshing is tetrahedral meshing.
- **e)** Fluid flow and heat transfer characteristic were analyzed using ANSYS FLUENT by applying different conditions at the domain boundary. The inner and outer walls of the tubes were defined as coupled for energy transfer from the hot fluid (exhaust gas) to the cold fluid (water). The analysis was done using k-ε turbulence model with standard wall function.
- **f)** For momentum equation, the walls of the tube were taken as no slip one and the walls of the shell were taken as no-slip adiabatic ones.
- **g)** The analysis were carried out by varying the velocity of cold stream (water) and different output parameters like outlet temperature of both the fluids, heat transfer coefficient of both the tube and shell side were obtained.
- **h)** A correlation was developed using regression analysis in Microsoft excel to estimate the inside tube heat transfer coefficient for turbulent regime.
- **i)** An experimental analysis of helical tube heat exchanger was carried out and the developed correlation was used to estimate the inside tube heat transfer coefficient experimentally. The simulated results were validated by comparing with the present experiments.

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2.1 Overall Heat Transfer Coefficient

In the heat transfer analysis of heat exchangers, various thermal resistances in the path of heat flow from the hot to the cold fluid are combined into an overall heat transfer coefficient (U). Consider that the total thermal resistance (R) to heat flow across a tube, between the inside and outside flow, is composed of the following thermal resistances.

$$R = \left\{ \begin{array}{l} \text{Thermal} \\ \text{resistance} \\ \text{of inside} \\ \text{flow} \end{array} \right\} + \left\{ \begin{array}{l} \text{Thermal} \\ \text{resistance} \\ \text{of inside} \\ \text{material} \end{array} \right\} + \left\{ \begin{array}{l} \text{Thermal} \\ \text{resistance} \\ \text{of outside} \\ \text{flow} \end{array} \right\}$$

And the various terms are given by,

$$R = \frac{1}{A_i h_i} + \frac{1}{k A_m} + \frac{1}{A_o h_o} \qquad (4.1)$$

The thermal resistance (R) can be expressed as an overall heat transfer coefficient based on either the inside or the outside surface of the tube. Overall heat transfer coefficient (U_o) based on outer surface is defined as

$$U_o = \frac{1}{\left(\frac{d_o}{d_i}\right)\left(\frac{1}{h_i}\right) + \left[\frac{1}{2k}\right] d_o \ln\left(\frac{d_o}{d_i}\right) + \left(\frac{1}{h_o}\right)} \qquad (4.2)$$

Similarly, the overall heat transfer coefficient (U_i) based on inner surface is defined as

$$U_i = \frac{1}{\left(\frac{d_i}{d_o}\right)\left(\frac{1}{h_o}\right) + \left[\frac{1}{2k}\right] d_i \ln\left(\frac{d_o}{d_i}\right) + \left(\frac{1}{h_i}\right)} \qquad (4.3)$$

When the wall thickness is small and its thermal conductivity is high, the tube resistance can be neglected and the overall heat transfer coefficient for inner surface reduces to

$$U_i = \frac{1}{\left(\frac{1}{h_i}\right) + \left(\frac{1}{h_o}\right)} \qquad (4.4)$$

2.1 . Number of Transfer Unit (NTU)

Number of Transfer Unit (NTU) is defined as the ratio of overall thermal conductance to the smallest heat capacity rate.

$$NTU = \frac{U_m A}{C_{min}} \qquad (4.5)$$

NTU designates the non dimensional heat transfer size or thermal size of exchanger and therefore it is a design parameter. NTU provides a compound measure of heat exchanger size through the product of heat transfer surface area (A) and the overall heat transfer coefficient (U). Hence, in general, NTU does not necessarily indicate the physical size of heat exchanger. In contrast, the heat transfer surface area designates the physical size of heat exchanger. A large value of NTU does not necessarily mean that a heat exchanger is large in size.

2.3 Effectiveness (ε_1)

Effectiveness is the measure of thermal performance of heat exchanger. It is defined as the ratio of actual heat rate to the maximum possible heat transfer thermodynamically permitted.

$$\varepsilon_1 = \frac{q}{q_{max}} \qquad (4.6)$$

Under ideal condition, using the value of actual heat transfer rate (q) from the energy conservation equation, the effectiveness (ε_1) valid for all flow arrangement of the two fluids is given by

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$$\varepsilon_1 = \frac{T_{h,i}-T_{h,o}}{T_{h,i}-T_{c,i}} \qquad (4.7)$$

$$\varepsilon_1 = \frac{T_{c,o}-T_{c,i}}{T_{h,i}-T_{c,i}} \qquad (4.8)$$

Table 1 Heat exchanger Effectiveness (ε_1) formulas [19, 20]

Flow arrangement	**ε_1 formula**
Parallel flow	$\varepsilon_1 = \frac{1-\exp[-N(1+C)]}{1+C}$
Counter flow	$\varepsilon_1 = \frac{1-\exp[-N(1-C)]}{1-C\exp[-N(1-C)]}$
Cross-flow: Both fluids unmixed	$\varepsilon_1 = 1-\exp\left[\frac{\exp(-NCn)-1}{Cn}\right]$ Where n= $N^{-0.22}$
Cross-flow: One fluid mixed, other unmixed	1. C_{min} mixed, C_{max} unmixed: $\varepsilon_1 = 1-\exp[-\frac{1}{C}(1-e^{-NC})]$
	2. C_{min} unmixed, C_{max} mixed: $\varepsilon_1 = \frac{1}{C}\{1-\exp[-C(1-e^{-N})]\}$
All heat exchanger with C=0	$\varepsilon_1 = 1-e^{-N}$

III. EXPERIMENTAL SET-UP

The schematic diagram of experimental set-up is shown in fig.1. This set-up is a well instrumented heat exchanging medium in which the cold water flowing through the coiled tube is heated by the exhaust gas from diesel engine flowing in the shell side. The heat exchangers include a copper tube and an insulated shell. Before start-up, water was circulated through the coil in order to check if there is any leakage. The engine was started and the hot gases were allowed to flow through the shell. Water was also supplied to the engine to avoid overheating. The dimensions of the heat exchangers are given in table 2. To measure the flow rate of the cold stream, Rotameter is installed while for the hot stream, Pitometer is placed in the passage of engine exhaust. The inlet and outlet temperatures of the cold stream are measured using thermometer whereas for the hot stream, the temperatures were measured using thermocouples.

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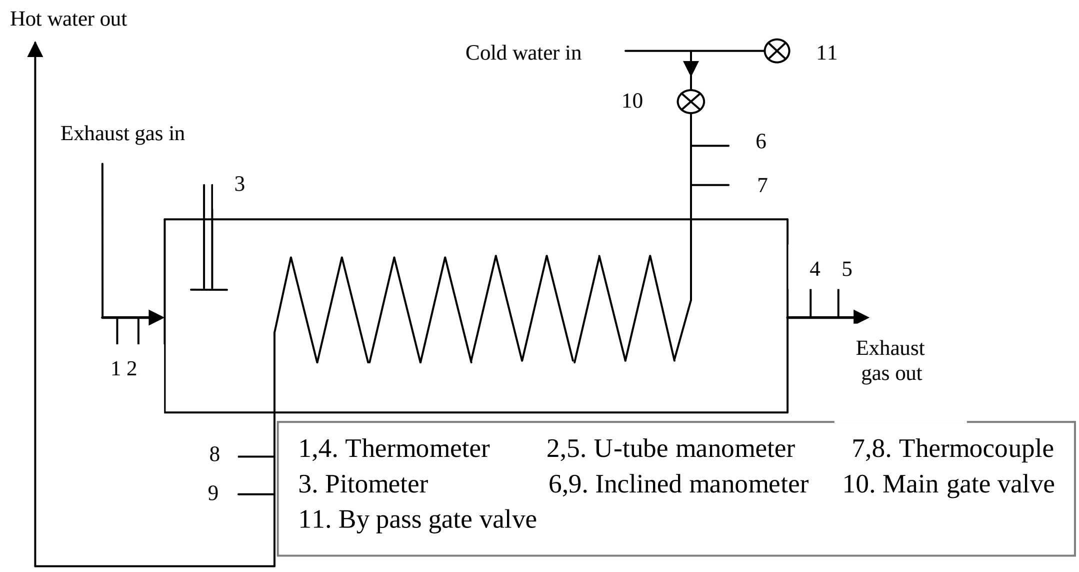

Fig.1 Flow diagram of Helical Tube Heat Exchanger

Table 2 Dimensions of helically coiled heat exchanger

1.	Inner diameter of tube	6.25 mm
2.	Outer diameter of tube	6.35 mm
3.	Coil diameter	165 mm
4.	Length of tube	11.362 m
5.	Surface area of tube	0.227 m^2
6.	Length of shell	1.2 m
7.	Diameter of shell	0.3 m
8.	Free length of coil	0.97 m
9.	Number of turns of coil	22
10.	Pitch	4.57 cm

3.1 Experimental procedure

a. Initially, water was circulated through the helical tube in order to check if there is any leakage. When it is ensured that water is circulating through the helical tube, the engine was started.
b. The water is allowed to flow through the helical tube through the main control valve.
c. The water flow rate was slowly increased and it was allowed to attain a maximum flow rate. Water is also supplied to the engine so that it does not get over heated during the process.
d. The analysis has been carried out by varying the tube side fluid velocity from 0.5 m/s to 1.75 m/s, while the shell side velocity was kept constant at 0.3 m/s. During the operation, different parameters like inlet and outlet temperature of both the fluids, mass flow rate and pressure drop readings were noted at different conditions for further calculations.
e. The experiments were performed covering the turbulent flow ranges for both the tube and shell side.
f. All measurements were taken at steady conditions, by ensuring constant temperature records.
g. The steady state condition arises when the temperature and flow rate at all the measuring points were no longer fluctuating.

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Fig.2 Diesel engine

3.2 Experimental data of Shell and Helical tube heat exchanger

Table 3 Tube side experimental data

Sl.no.	Flow meter (LPM)	Velocity (m/s)	Inlet temperature (K)	Outlet temperature (K)	Mean temperature (K)	Pressure drop (cm of Hg)
1.	0.95	0.50	299	329	314.0	6.0
2.	1.45	0.75	299	326	312.5	6.2
3.	1.80	1.00	299	324	311.5	6.4
4.	2.80	1.50	299	320	309.5	6.8
5.	3.30	1.75	299	317	308.0	7.0

Table 4 Shell side experimental data

Sl.no.	Pitometer reading (mm of H_2O)	Velocity (m/s)	Inlet temperature (K)	Outlet temperature (K)	Mean temperature (K)	Pressure drop (mm of H_2O)
1.	8	0.38	393	353	373.0	1.5
2.	8	0.38	393	351	372.0	1.5
3.	8	0.38	393	348	370.5	1.5
4.	8	0.38	393	345	369.0	1.5
5.	8	0.38	393	343	368.0	1.5

Table 5 Tube side and Shell side Reynold Number

Sl.no.	Tube side				Shell side		
	Reynold no.	Dean no.	Nusselt no.	Prandtl no.	Reynold no.	Nusselt no.	Prandtl no.
1.	5123	997	7.4	4.274	3903	7.25	0.7111
2.	7041	1370	9.8	4.407	3921	7.28	0.7113
3.	9793	1906	12.4	4.541	3948	7.31	0.7116
4.	13055	2541	15.4	4.809	3975	7.34	0.7119
5.	14695	2860	17.0	5.010	3993	7.36	0.7122

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Table 6 Overall Heat Transfer Coefficient

Sl.no.	Inner tube heat transfer coefficient (W/m^2K)	Outer tube heat transfer coefficient (W/m^2K)	Overall heat transfer coefficient (W/m^2K)
1.	1017	35.36	34.17
2.	1315	35.38	34.45
3.	1642	35.41	34.66
4.	2012	35.44	34.83
5.	2204	35.46	35.00

Table 7 Heat capacity

Sl.no.	Cp of cold flow (J/kgK)	Cp of hot flow (J/kgK)	Heat capacity of cold flow (Cc) (W/K)	Heat capacity of hot flow (Ch) (W/K)	Heat capacity (C)
1.	4178.25	1009.0	63.71	20.23	0.318
2.	4178.00	1008.9	95.73	20.28	0.212
3.	4178.00	1008.8	127.67	20.36	0.159
4.	4178.00	1008.6	191.61	20.44	0.107
5.	4178.00	1008.5	223.63	20.49	0.088

Table 8 Effectiveness

Sl.no.	Overall heat transfer coefficient (W/m^2K)	NTU	Effectiveness (%)
1.	34.17	0.383	30.3
2.	34.45	0.385	30.9
3.	34.66	0.386	31.2
4.	34.83	0.387	31.6
5.	35.00	0.388	31.7

IV. COMPARISON OF CFD AND EXPERIMENTAL RESULTS

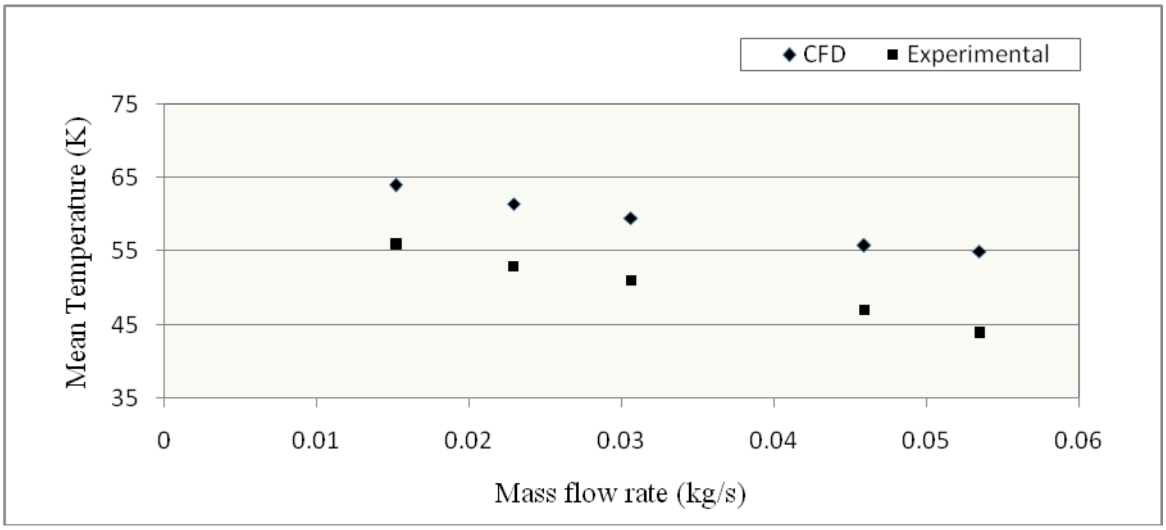

Fig.3 Mean temperature V/S Mass flow rate

From fig.3, it is observed that the average temperature difference between the CFD and Experimental value is 8.5. This is because there must be some losses taking place in the helical tube and shell. The tube might be corroded due

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to which the heat transfer is not taking place properly. In case of shell also there must be a heat loss even though insulation is provided since 100% insulation is not possible practically.

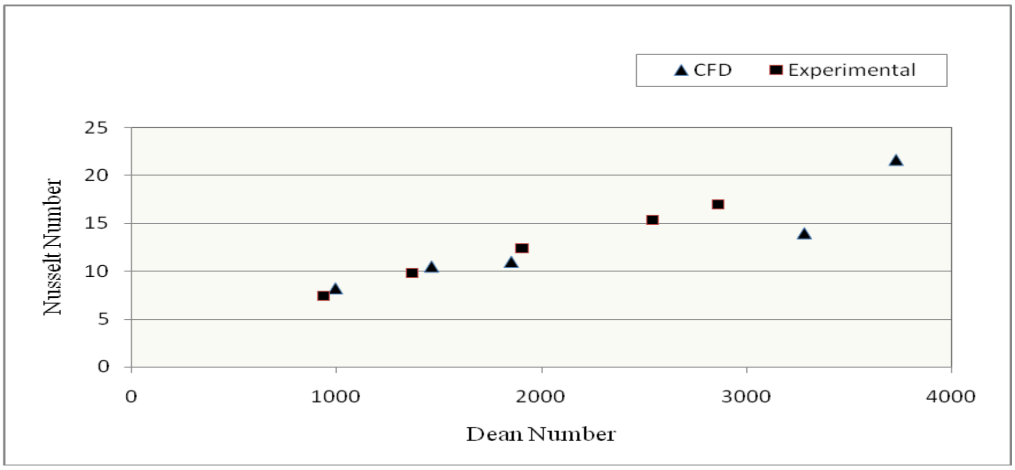

Fig.4 Comparison of Nusselt number

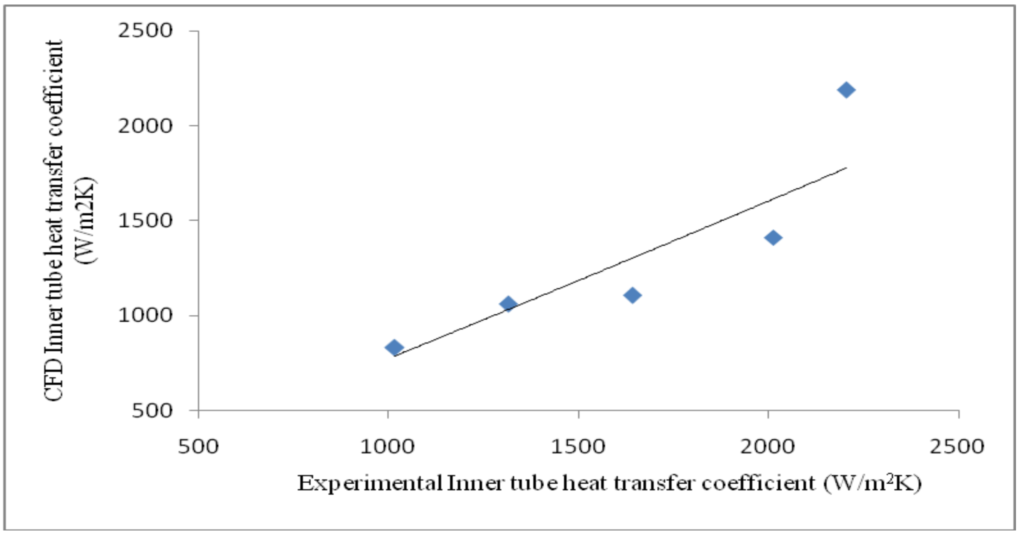

Fig.5 Comparison of CFD and Experimental Inside tube Heat Transfer Coefficient

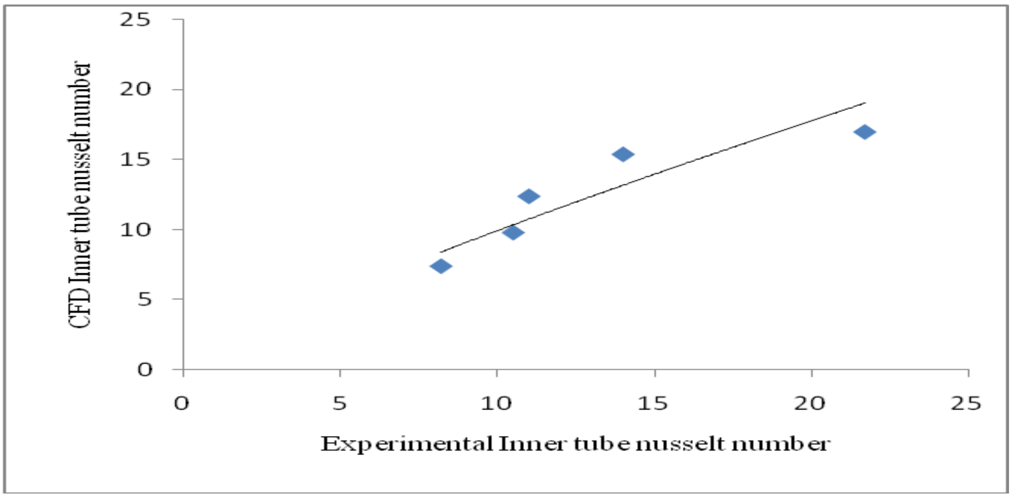

Fig.6 Comparison of CFD and Experimental Inner Tube Nusselt Number

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The comparison between the Experimental Nusselt number by the proposed correlation and the CFD Nusselt number is presented in fig.6. It is evident that the proposed correlation is in good agreement with the Experimental data within some experimental error limits.

$$\text{Percentage error} = \frac{(\text{CFD Nusselt number}) - (\text{Experimental Nusselt number})}{(\text{CFD Nusselt number})}$$

$$= \frac{13.08 - 12.40}{13.08}$$

= 0.052

= 5.2%

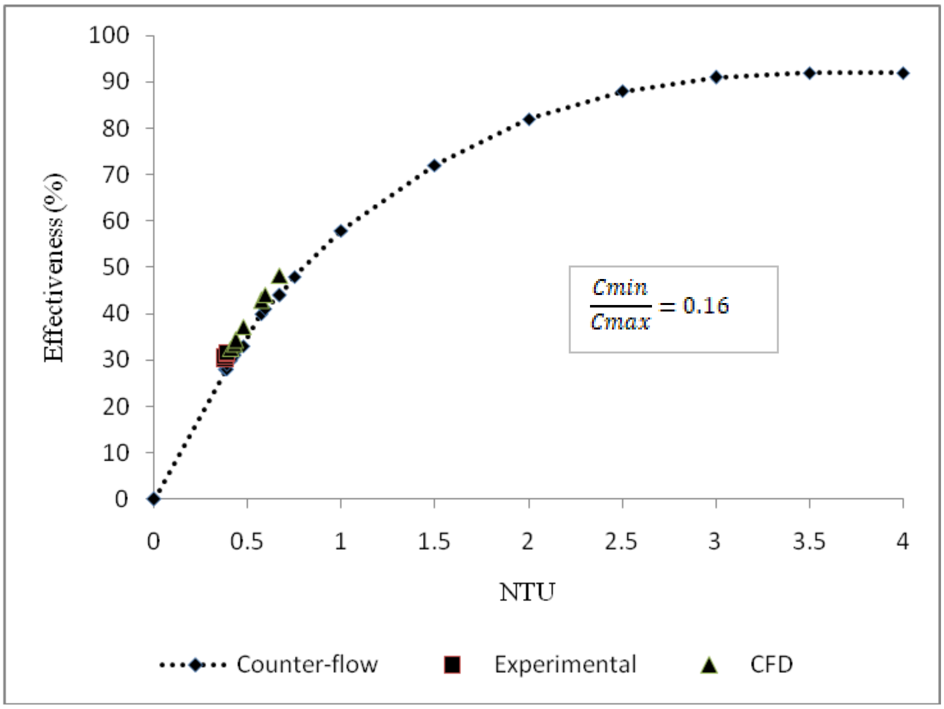

Fig.7 Effectiveness V/S NTU

The effectiveness of helical tube heat exchanger that was obtained from CFD and Experimental analysis are being compared with the effectiveness of the heat exchanger for counter flow for a capacity ratio of 0.16. It can be observed from the graph 7 that the CFD and Experimental results are in good agreement within some experimental error limits. In heat exchanger, rating and sizing are two important problems that are encountered in the thermal analysis of heat exchanger. The sizing problem is concerned with determination with the matrix of dimension to meet the specified heat transfer and pressure drop requirement. NTU is the measure of the actual heat transfer area, or the physical size of the heat exchanger. The higher the NTU, the larger is the physical size. Table 8 gives the evaluated values of the NTU and Effectiveness of cross counter flow heat exchanger obtained from CFD and Experimental analysis. Since helical tube heat exchanger comes under the category of compact heat exchanger, the values that were obtained for NTU are low. The lower value of NTU gives more compact shape of heat exchanger.

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V. Conclusion

An investigation was carried out to study the Shell and helical tube heat exchanger both computationally and experimentally. It was revealed that the empirical correlation is quite in agreement with the experimental results within experimental error limits. Based on the results obtained from the CFD and Experimental analysis, the following conclusions have been drawn out.

a) CFD and Experimental analysis are being compared with the effectiveness of the heat exchanger for counter flow for a capacity ratio of 0.16. It can be observed from CFD and Experimental results are in good agreement within some experimental error limits.
b) The pressure drop in the coil is very high thereby making the necessity of large pumping power. It is recommended to use shorter length of coil.
c) The NTU value of the helical coil has been reasonably low thereby justifying the name compact.

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