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Review Of Experimental And Numerical Analysis Of Heat Exchanger In The Light Of Waste Heat Recovery Applications

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ABSTRACT
The present work has been carried out with a review of the performance of a helical coil heat exchanger in the light of waste heat recovery applications. In maximum cases analysis was done for number of cases and the effect on temperature rise and the pressure drop in the helical tube and shell was observed by researchers. 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 NTU value of the helical coil has been reasonably low thereby justifying the name compact. The effectiveness of the helical tube heat exchanger is quite comparable with other conventional heat exchanger design
Keywords: Helical Coil, heat exchanger, empirical correlation, experimental Analysis

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I. INTRODUCTION

The recovery of waste heat from exhaust gases has become a necessity due to the dwindling energy resources and production cost. The need to use energy more efficiently has become a necessity since the large increase in oil prices. Energy conservation is primarily concerned with the task of extracting maximum production from specific energy consumption. A major result of the energy conservation drive is the development of process recovery aimed at reducing the amount of waste heat discharged to the environment thus increasing the overall efficiency of various processes and systems. Heat recovery conserves energy, reduces the overall operating costs and thereby reduces peak loads.

1.1 Review of Experimental work on Helical Tube Heat Exchanger

Several studies have indicated that helically coil tubes are superior to straight tubes when employed in heat transfer applications. In one of the work carried out by **D.G. Prabhanjan et al.** [1] on "Comparison of heat transfer rates between a straight tube heat exchanger and helically coiled heat exchanger", it was observed that the heat transfer rate was affected by the geometry of the heat exchanger and the temperature of the water bath surrounding the heat exchanger. Also the flow rate did not affect the heat transfer coefficient, most likely from the fact the flow was turbulent and increasing the flow rate does not greatly change the wall effects. Temperature rise of the fluid was found to be effected by the coil geometry and by the flow rate. In another work carried out by **M.R. Salimpour et al.** [2] on "Heat transfer coefficient of Shell and coiled tube heat exchanger", Heat exchanger with three different coil pitches were tested for counter flow configuration. From the result of the study, it was found that the shell side heat transfer coefficient of the coil with larger pitches is higher than those for smaller pitches. Finally based on the result of the study, two correlations were developed to predict the inner and outer heat transfer coefficient of the coiled tube heat exchanger.

M.R. Salimpour [3] also made an investigation to study the heat transfer coefficient of temperature dependent property engine oil flow inside shell and coiled tube heat exchanger experimentally. From the result of the study, it was observed that increasing the coil tube pitch decreases the inner nusselt number. Also, increase of coil tube pitch leads to higher value of shell side Nusselt number because in smaller coil pitches, the coolant water is confined in the space between successive coil rounds and a semi dead zone is formed, as in this region the flow is decelerated, heat transfer coefficient will be descended.

Another similar work was carried out by **W. Witchayanuwat and S. Kheawhom** [4] on "Heat transfer coefficient for particulate air flow in shell and coiled tube heat exchanger". From the result of the study it was found that variation in the pitches of coiled tube slightly affects the shell side heat transfer coefficient. Two empirical correlations were also developed to predict the inside and outside heat transfer coefficient of the coiled tube heat exchanger for the particulate airflow water system. In another work by **H. Shokouhmand et al. [5] on** "Experimental and investigation of Shell and Coil tube Heat exchanger using Wilson Plot", **an experiment was performed for both the Parallel flow and counter flow configuration.** Overall heat transfer coefficients of the heat exchangers were calculated using Wilson plots. It was observed that the shell-side Nusselt numbers of counter-flow configuration were slightly more than the ones of parallel-flow configuration. Finally, it was observed that the overall heat transfer coefficients of counter-flow configuration are 0–40% more than those of parallel-flow configuration.

Paisarn Naphon et al. [6] made a detailed survey on Single phase and double phase flow and Heat transfer characteristic in helically coiled tubes, spirally coiled tubes and other coiled tubes. In one of his paper "Thermal performance and pressure drop of the Helical coil Heat Exchanger with or without helically crimped fins", [7] he studied the thermal performance and pressure drop of Heat Exchanger in which the heat exchanger consists of thirteen turns concentric helical coil tubes with coil tubes consisting of two different coil diameters. He concluded that outlet cold water temperature increases with increasing hot water mass flow rate. Inlet hot and cold water mass flow rates and inlet hot water temperature also have a significant effect on the heat exchanger effectiveness. Paisarn Naphon et al. [8] also made a study on effect of curvature ratio on the heat transfer and flow development in the horizontal spirally coiled tubes. It was observed that because of centrifugal force, the heat transfer and pressure drop are more in spirally coil tube compared to that of straight tube.

Andrea cioncolini et al. [9] made a study on laminar to turbulent flow transition in helically coiled tubes. The influence of curvature on the laminar to turbulent flow transition in helically coiled pipes was analyzed from direct inspection of the experimental friction factor profiles obtained for twelve coils. The coils studied had ratios of coil diameter to tube diameter ranging from 6.9 to 369 while the coil pitches were small enough to neglect the effect of torsion on the flow.

Unlike the study made by **Andrea cioncolini et al.** [9] on laminar to turbulent flow transition in helically coiled tubes, **R.A. Seban et al.** [10] have done an investigation on laminar flow of oil and turbulent flow of water in coiled tubes having ratio of coil to tube diameter of 17 and 104. The friction factor for laminar and turbulent flow corresponding with the results of Ito and are predictable by his equations when for non-isothermal flow the properties are evaluated at the mean film temperature. **B.V.S.S.S. Prasad et al.** [11] also conducted experiments on helical tube heat exchanger and developed a correlation for pressure drop and heat transfer coefficient for the tube and shell side. In the tube side, the laminar friction factor and Nusselt numbers are represented as functions of $Re\sqrt{(d/D)}$, whereas in turbulent flow the results are correlated with $Re(d/D)^2$. The pressure drop and heat transfer values for the shell side are found to follow the classical Blasius and Dittus-Boelter type relations, while a strong dependence on the coil to tube diameter ratio is detected. The performance of the exchanger has been tested not only as simulated experimental exchanger but also as a waste heat recovery device for a 60 HP gas turbine.

1.2 Review of Computational work on Helical Tube Heat Exchanger

Not Much work has been carried out in Computational Fluid Dynamics in respect of Heat Exchanger. In one of the paper carried out by **J.S. Jayakumar et al.** [12] 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 ANSYS. 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. [13] also 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 ANSYS. 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**, [14] 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 Geometry which is the preprocessor.
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 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.

III. HEAT EXCHANGER

A heat exchanger is a device that is used to transfer thermal energy between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions. Many types of Heat Exchanger have been developed for use at such varied levels of technological sophistication and sizes as Steam power plants, Chemical processing plants, building heating and air conditioning, household refrigerators, car radiators and so on. The proper design, operation and maintenance of heat exchangers will make the process energy efficient and minimize energy losses. Heat exchanger performance can deteriorate with time, off design operations and other interferences such as fouling, scaling etc. It is necessary to assess periodically the heat exchanger performance in order to maintain them at a high efficiency level.

3.1. Classifications of Heat Exchanger

Heat Exchangers are available in so many sizes, types, configuration and Flow arrangement and they are classified according to as:

I. **Classification by transfer process:** Heat exchanger can be classified as direct contact and indirect contact. In the Direct contact type, heat transfer takes place between two immiscible fluids, such as a gas and liquid, coming into direct contact. Cooling towers, jet condensers for water vapor, and other vapors utilizing water spray are typical example of direct contact exchangers. In the Indirect contact type of Heat exchangers, such as automobile radiators, the hot and cold fluid are separated by an impervious surface and they are referred to as surface heat exchanger. There is no mixing of the two fluids.
II. **According to compactness:** The definition of compactness is quite an arbitrary matter. The ratio of the heat transfer surface area on one side of the heat exchanger to the volume can be used as a measure of the compactness of heat exchangers. A heat exchanger having surface area density on any one side greater than about 700 m^2/m^3 quite arbitrarily is referred to as compact heat exchanger regardless of its structural design. For example, automobile radiators having an area density of the order of 1100 m^2/m^3 and the

glass ceramic heat exchanger for some vehicular gas turbine engines having an area density on the order of 6600 m^2/m^3 are compact heat exchanger. On the other extreme of the compactness scale, plane tubular and shell and tube type exchangers, having an area density in the range of 70 and 500 m^2/m^3 are not considered compact.

III. Classification by construction type:

a) **Tubular heat exchanger**: This type of heat exchanger can accommodate a wide range of operating pressure and temperature. Shell and tube heat exchanger is a common example of tabular heat exchanger which consists of round tubes mounted on a cylindrical shell with their axes parallel to that of the shell.

b) **Plate heat exchanger**: As the name implies, Plate heat exchangers are usually constructed of thin plates. The plates may be smooth or may have some form of corrugation. The compactness factor for plate heat exchangers ranges from 120 to 230 m^2/m^3.

c) **Plate fin heat exchanger**: This type of heat exchanger is generally used for gas to gas applications but they are used for low pressure applications not exceeding about 10 atm. The maximum operating temperatures are limited to 800 °C. Plate fin heat exchangers also have been used for cryogenic applications.

d) **Tube fin heat exchanger**: This type of heat exchanger is used for wide range of tube fluid operating pressure not exceeding about 30 atm. and operating temperature from low cryogenic applications to about 87 °C.

e) **Regenerative heat exchanger**: Regenerative heat exchanger can be either static or dynamic. The static type has no moving parts and consists of a porous mass through which hot and cold fluid passes alternatively. In dynamic type regenerators, a matrix is arranged in the form of a drum which rotates about an axis so that a given portion of the matrix passes periodically through the hot stream and then through the cold stream.

IV. Classifications by flow arrangement:

a) **Parallel flow:** The hot and cold fluids enter at the same end of the heat exchanger, flow through in the same direction, and leave together at the other end.

b) **Counter flow:** The hot and cold fluids enter in the opposite ends of the heat exchanger and flow through in opposite directions.

c) **Cross flow:** The two fluids usually flow at right angles to each other. The flow may be mixed or unmixed, depending on the design.

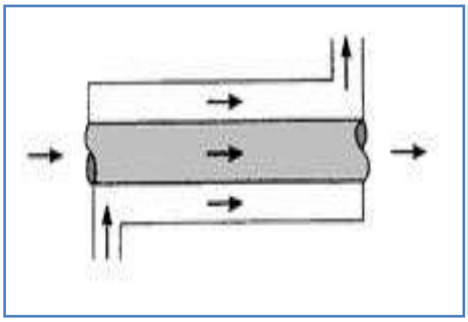


Fig.1 Parallel flow

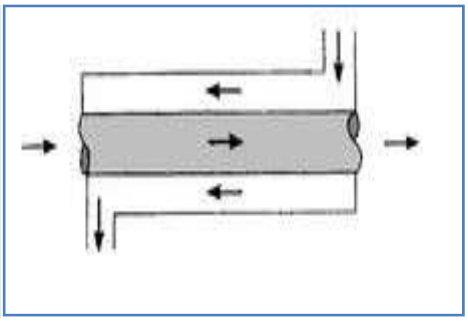


Fig.2 Counter flow

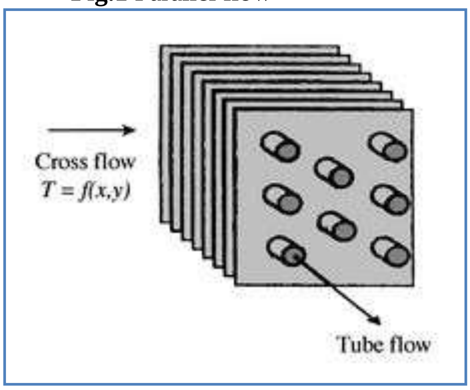


Fig. 3 Cross flow, both fluid unmixed

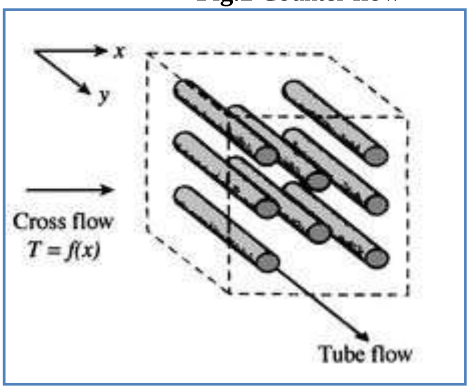


Fig.4 Cross flow, one fluid mixed and other unmixed

IV. Classifications by Heat transfer Mechanism:

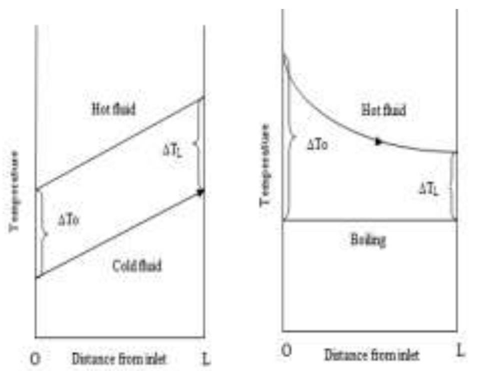
a) Single-phase forced or free convection.

b) Phase change (boiling or condensation)

c) Radiation or combined convection and radiation.

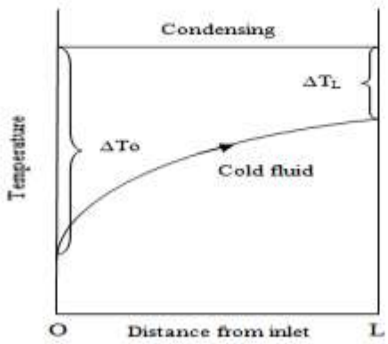
3.2. Temperature distribution in Heat Exchangers

In stationary type heat exchangers, the heat transfer from the hot to the cold fluid causes a change in temperature of one or both fluids flowing through the heat exchanger. Figure 1a characterizes a pure counter flow heat exchanger in which the temperature rise in the cold fluid is equal to the temperature drop in the hot fluid. Thus the temperature difference ΔT between the hot and cold fluids is constant throughout. However, in all other cases (i.e fig.1b to e), the temperature difference ΔT between the hot and cold fluids varies with position along the path of flow. Figure 1b correspond to a situation in which the hot fluid condenses and heat is transferred to the cold fluid, causing its temperature rise along the path of flow. In figure 1c, cold fluid is evaporating and cooling the hot fluid along its path of flow. Figure 1d shows a parallel flow arrangement in which both fluids flow in the same direction, with the cold fluid experiencing a temperature rise and the hot fluid a temperature drop. The outlet temperature of the cold fluid cannot exceed that of the hot fluid. Therefore, the temperature effectiveness of parallel flow exchangers is limited because of this limitation, generally they not considered for heat recovery. Figure 1e shows a counter flow arrangement in which fluid flow in opposite directions. The exit temperature of the cold fluid can be higher than that of hot fluid. Theoretically, the exit temperature of one fluid may approach the inlet temperature of the other. Therefore, the thermal efficiency of the counter flow heat exchanger can be twice that of parallel flow heat exchanger. Figure 2 shows the temperature distribution of a multipass and cross-flow arrangement in which heat transfer exhibit a more complicated pattern.

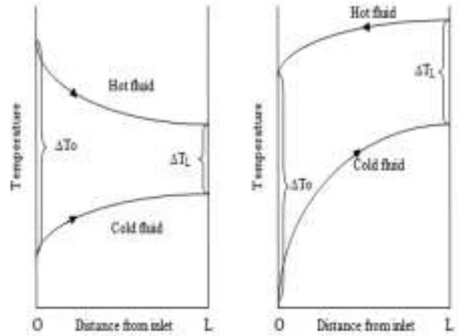


(a) Uniform temperature difference or uniform heat flux

(c) Uniform surface temperature (as in boiling)



(b) Uniform surface temperature (as in a condenser)



(d) Parallel flow heat exchanger

(e) Counter flow heat exchanger

Fig.5 Axial temperature distribution in typical single-pass heat transfer matrices

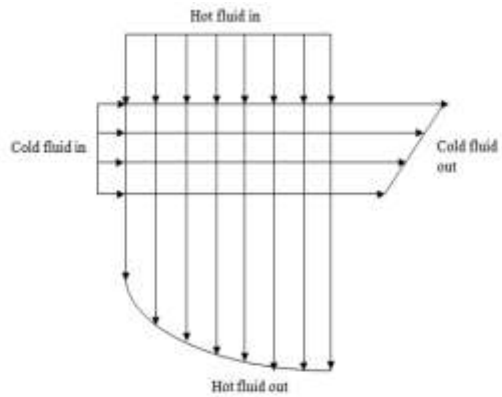


Fig. 6 Temperature distribution in a cross-flow heat exchanger

3.3. 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.

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

And the various terms are given by,

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

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

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

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

$$U_i = \frac{1}{\left(\frac{d_i}{d_o}\right)\left(\frac{1}{h_o}\right) + \left[\frac{1}{2k}\right] d_i \ln\left(\frac{d_o}{d_i}\right) + \left(\frac{1}{h_i}\right)} \qquad (3)$$

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

$$U_i = \frac{1}{\left(\frac{1}{h_i}\right) + \left(\frac{1}{h_o}\right)} \qquad (4)$$

3.4. 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.

$$NTU = \frac{U_m A}{C_{min}} \qquad (5)$$

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.

3.5. 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.

$$\varepsilon_1 = \frac{q}{q_{max}} \qquad (6)$$

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

$$\varepsilon_1 = \frac{T_{h,i} - T_{h,o}}{T_{h,i} - T_{c,i}} \qquad (7)$$

$$\varepsilon_1 = \frac{T_{c,o} - T_{c,i}}{T_{h,i} - T_{c,i}} \qquad (8)$$

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	$\varepsilon_1 = 1 - e^{-N}$

IV. HELICAL COIL TUBE HEAT EXCHANGER

Helically coiled tubes are superior to straight tubes when employed to heat transfer applications due to their compact structure and high heat transfer coefficient. Helical Coil Tube Heat Exchangers are used in a wide variety of application like Heat Recovery system, air conditioning and refrigeration, Chemical reactor, food and dairy process etc. There application is mainly in controlling the temperature of the reactors for exothermic reactions, in cryogenics and also in other heat transfer applications.

4.1Fluid flow in curved tubes

When a fluid flows through a straight tube, the fluid velocity is maximum at the tube center, zero at the tube wall & symmetrically distributed about the axis. However, when the fluid flows through a curved tube, the primary velocity profile is distorted by the addition of secondary flow pattern. The secondary flow is generated by centrifugal action and acts in a plane perpendicular to the primary flow. Since the velocity is maximum at the center, the fluid at the center is subjected to the maximum centrifugal action, which pushes the fluid towards the outer wall. The fluid at the outer wall moves in ward along the tube wall to replace the fluid ejected outwards. This results in the formation of two vortices symmetrical about a horizontal plane through the tube center.

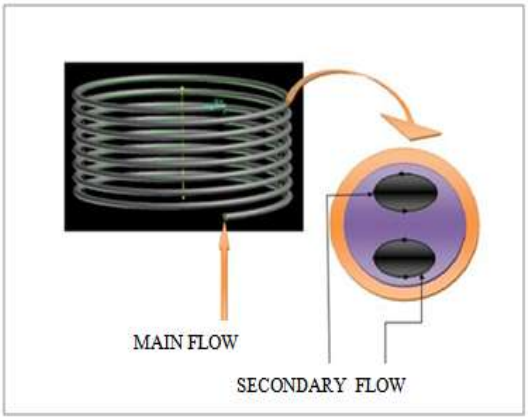


Fig.7 Schematic of Helically coiled tubes and Secondary flow in enlarged cross-sectional view

4.2 Geometry of Shell and coiled tube heat exchanger

The geometry of the helical tube and shell is shown in fig.4.4. In the figure, d is the diameter of the coiled tube, R_c is the curvature radius of the coil, D is the diameter of the shell and b is the coil pitch.

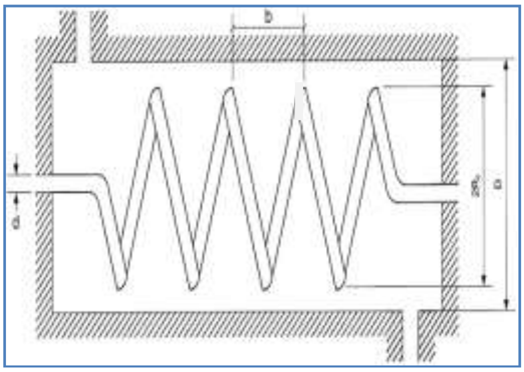


Fig.8 A typical shell and coil tube heat exchanger

The important dimensionless parameters of coiled tube are Reynold number (Re), Nusselt number (Nu), and Dean number (De). As Reynold number is used in order to characterize the flow in case of a straight pipe, Dean number comes into account in case of a helical pipe.

Reynold number (Re) $=\frac{\rho V d}{\mu}$, Nusselt number (Nu) $= \frac{hd}{k}$

Dean number (De) = $Re(\frac{d}{2R_c})^{0.5}$

Shell side Reynold number and Nusselt number are defined as,

$R_e = \frac{\rho V d_h}{\mu}$ and $Nu = \frac{h d_h}{k}$, Where d_h is the hydraulic diameter of shell and it is expressed as, $d_h = \frac{D^2 - 2\pi R_c d_o^2 \gamma^{-1}}{D - 2\pi R_c d_o \gamma^{-1}}$, γ is the dimensional pitch $= \frac{b}{2\pi R_c}$

The critical Reynold number for the transition from laminar to turbulent flow in helical coils is a function of the coil parameters. The critical Reynolds number may be determined using the correlation developed by Schmidt.

$$Re_{cr} = 2300[1 + 8.6(^{d}/_{2R_c})^{0.45}]$$

(9)

For flow inside a circular tube, the turbulent flow is usually observed for

$$R_e = \frac{\rho v D}{\mu} > 2300$$

(10)

However, this critical value is strongly dependent on the surface roughness, the inlet condition and the fluctuation in the flow. In general, the transition may occur in the range 2000>R_e>4000.

4.3 Applications of Helical tube heat exchanger

a) Helical coils are used for transferring heat in chemical reactors and agitated vessels because heat transfer coefficients are higher in helical coils. This is especially important when chemical reactions have high heats of reaction are carried out and the heat generated (or consumed) has to be transferred rapidly to maintain the temperature of the reaction. Also, because helical coils have a compact configuration, more heat transfer surface can be provided per unit of space than by the use of straight tubes.
b) Because of the compact configuration of helical coils, they can be readily used in heat transfer application with space limitations, for example, in steam generations in marine and industrial applications.
c) Helical coiled tubes have been and are used extensively in cryogenic industry for the liquefaction of gases.

4.4 Advantages and Disadvantages of Helical tube heat exchanger

Advantages

a) Coils give better heat transfer performance, since they have lower wall resistance & higher process side coefficient.

b) A coil can provide a large surface area in a relatively small reactor volume.

c) Coils are more versatile for scale – up.

Disadvantages

a) For highly reactive material or highly corrosive material coils cannot be used, instead jackets are used.

b) Cleaning of vessels with coils becomes much difficult than with jackets.

c) Coils play a major role in selection of agitation system. Densely packed coils can create unmixed regions by interfering with fluid flow.

V. WASTE HEAT RECOVERY

Waste heat is heat, which is generated in a process by way of fuel combustion or chemical reaction, and then “dumped” into the environment even though it could still be reused for some useful and economic purpose. The essential quality of heat is not the amount but rather its “value”. Large quantity of hot flue gases is generated from Boilers, Kilns, Ovens and Furnaces. If some of this waste heat could be recovered, a considerable amount of primary fuel could be saved. The energy lost in waste gases cannot be fully recovered. However, much of the heat could be recovered and loss can be minimized [22].

The energy conservation and its efficient utilization are very much essential and it can be achieved by the following ways.

- By increasing the overall thermal efficiency of existing process.
- Using the heat recovery system to extract heat that would otherwise go as waste.
- By tapping alternative energy sources namely, tidal, solar, geothermal, hydrothermal and wind power.

5.1. Benefits of Waste heat recovery

I. **Direct benefit**: Recovery of waste heat has direct effect on the efficiency of the process. This is reflected by reduction in the utility consumption and costs and process cost.

II. **Indirect benefit**:

- Reduction in pollution.
- Reduction in equipment size.
- Reduction in auxiliary energy consumption.

The exhaust gas can be recovered by using suitable heat exchanger. Compact heat exchanger can be used for heat recovery purpose because of its several advantageous 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.

Table 2 Area density of generic compact heat exchanger

Liquid-liquid compact heat exchanger	>300 m^2/m^3
Gas-liquid compact heat exchanger	>700 m^2/m^3
Laminar flow heat exchanger	>3000 m^2/m^3

5.2. Temperature of waste gases from process equipment in low, medium and high temperature ranges [22]

Table 3 Typical waste heat temperature at low temperature range from various sources

Sl. No.	Name of device	Temperature (°C)
1.	Air compressor	27-50
2.	Pumps	27-88
3.	Internal combustion engine	66-120
4.	Air conditioning and Refrigeration condensers	32-43
5.	Drying, baking and curing ovens	93-230

Table 4 Typical waste heat temperature at medium temperature range from various sources

Sl. No.	Name of device	Temperature (°C)
1.	Steam boiler exhausts	230-480
2.	Gas turbine exhausts	370-540
3.	Heat treatment furnaces	425-650
4.	Reciprocating engine exhausts	315-600
5.	Catalytic crackers	425-650

Table 5 Typical waste heat temperature at High temperature range from various sources

Sl. No.	Name of device	Temperature (°C)
1.	Nickel refining furnace	1370-1650
2.	Aluminum refining furnace	650-760
3.	Steel heating furnace	925-1050
4.	Open hearth furnace	650-700
5.	Glass melting furnace	1000-1550

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.

Heat transfer is defined as the transmission of energy from one region to another as a result of temperature gradient that takes place by three modes namely Conduction, Convection and Radiation. Heat transmission, in majority of real situation, occurs as a result of these modes of heat transfer. The three modes are similar in that a temperature differential must exist and the heat exchange is in the direction of decreasing temperature. In the present work, the exhaust gas from diesel engine which comes under the category of low temperature range (66-120°C) has been used as the shell side fluid for heat transfer analysis. The exhaust gas transfer heat to the cold fluid (water) that is flowing through the helical tube.

Computational Fluid Dynamics is becoming a wide spread tool, used by a vast number of engineers. CFD provides an option, which is cheaper, obtains a complete set of results and is suitable for almost all complexity of problems. CFD is also well suited for trouble shooting and also it has a faster turnaround time than experiments.

The present work begins with the CFD analysis of helical tube heat exchanger in order to see the effect of temperature rise and pressure drop along the length of the helical tube and the shell. The exhaust gas from diesel engine has been used as the shell side fluid for heat transfer analysis. The exhaust gas transfer heat to the cold fluid (water) that is flowing through the helical tube.

CFD provides the flexibility to change design parameters without the expense of hardware changes. It therefore costs less than laboratory or field experiments, allowing engineers to try more alternative designs than would be feasible otherwise. It also reduces design cycle time and cost by optimizing through computer predictions and provides higher level of confidence in prototype or field installed performance. Moreover it investigates and understands the “why” for existing problem or new equipment. The main objective of the present study is to analyze the shell and helical tube heat exchanger both computationally and experimentally and to validate the CFD results by comparing with the present experiment.

VI. CONCLUSION

A literature survey has been carried out to study the Shell and helical tube heat exchanger both computationally and experimentally. In maximum cases analysis was done for number of cases and the effect on temperature rise and the pressure drop in the helical tube and shell was observed by researchers. 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 NTU value of the helical coil has been reasonably low thereby justifying the name compact. The effectiveness of the helical tube heat exchanger is quite comparable with other conventional heat exchanger design.

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