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Experimental Investigation of Double Pipe Helical Coil Heat Exchanger with and without Micro-fins

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ABSTRACT

The various techniques for achieving improved heat transfer are usually referred to as “heat transfer augmentation” or “heat transfer enhancement” and the heat exchanger provided with heat transfer enhancement techniques as “Augmented Heat Exchanger”. The objective is to reduce as many of the factors as possible: Capital Cost, Power Cost, Maintenance Cost, Space and Weight, Consistent with safety and reliability. Present work describes the principal techniques of industrial importance for the augmentation of single phase heat transfer on the inside of tubes namely “Micro fin’’. So micro fins should be used in heat exchanger when high heat transfer rate is required and pressure drop is of no significance. This study investigates the heat transfer characteristics, flow friction characteristics, of a horizontal tube-in-tube heat exchanger for single phase heat transfer with water as the working fluid. The material used for the construction of heat exchanger is copper, owing to its high thermal conductivity. Experiments were carried out for plain tube and with micro fins insert at constant wall heat flux and different mass flow rates. The Reynolds number varied from 500 to 6500. Both heat transfer coefficient and pressure drop are calculated and the results are compared with those of plain tube. The obtained results show that the Nusselt number in the tube with the micro fin increases. It was observed that, based on constant flow rate, the heat transfer rates for finned heat exchanger were found to be more than the plain tube.

Keywords: Nusselt number, Prandlt number, heat exchanger, micro fins.

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I. INTRODUCTION

Helical coil heat exchangers are one of the most common equipment found in many industrial applications ranging from chemical and food industries, power production, electronics, environmental engineering, air-conditioning, waste heat recovery. The development of the flow in the helically coiled tubes is due to the centrifugal forces. The curvature of the tube produces a secondary flow field with a circulatory motion, which causes the fluid particles to move toward the core region of the tube. The secondary flow increases heat transfer rates and it reduces the temperature gradient across the cross-section of the tube. There is additional convective heat transfer mechanism, perpendicular to the main flow, which does not exist in conventional heat exchangers.

Techniques for Heat Transfer Enhancement

Many different methods have been considered to increase the rate of heat transfer in forced convection while reducing the size of the heat exchanger and effecting energy savings. Surface methods include any techniques, which directly involve the heat exchanger surface. They are used on the side of the surface that comes into contact with a fluid of low heat transfer coefficient in order to reduce the thickness of the boundary layer and to introduce better fluid mixing. The primary mechanisms for thinning the boundary layer are increased stream velocity and turbulent mixing. Secondary recirculation flows can further enhance convective transfer.

The development of high performance thermal systems has stimulated interest in methods to improve heat transfer. The need to increase the thermal performance of heat exchangers, thereby effecting energy, material & cost savings have led to development & use of many techniques termed as Heat transfer Enhancement. These techniques are also referred as Heat transfer Enhancement or Intensification. Enhancement techniques increase convective heat transfer by reducing the thermal resistance in a heat exchanger.

Applications

The use of Double pipe heat exchangers continues to increase due to enhancement of heat transfer so its applications include-

- Liquid Heating/Cooling.
- Cryogenic Cooling/ Waste Heat Recovery Technique
- Chemical and Food Industries
- Environmental Engineering
- Air-Conditioning

II. LITERATURE REVIEW

It has been widely reported in literature that heat transfer rates in helical coils are higher as compared to that in straight tubes. Due to the compact structure and high heat transfer coefficient, helical coil heat exchangers are widely used in industrial applications.

Zimparov et. al. [1] Heat transfer and isothermal friction pressure drop results have been obtained experimentally for two three-start spirally corrugated tubes combined with five twisted tape inserts with different relative pitches in the range of Reynolds number. The characteristic parameters of the tubes are: height to diameter ratio, and relative pitch. Significantly, higher friction factor and inside heat-transfer coefficients than those of the smooth tube under the same operating conditions have been observed. Extended performance evaluation criteria (PEC) equations for enhanced heat transfer surfaces have been used to assess the multiplicative effect. Thermodynamic optimum can be defined by minimizing the entropy generation number compared with the relative increase of heat transfer rate or relative reduction of heat transfer area.

Prabhanjan et. al. **[2]** The purpose of this study was to determine the relative advantage of using a helically coiled heat exchanger verses a straight tube heat exchanger for heating liquids. The particular difference in this study compared to other similar studies was the boundary conditions for the helical coil. Most studies focus on constant wall temperature or constant heat flux, whereas in this study it was a fluid to fluid heat exchanger.

Akpinar et. al [3] The effect of heat transfer rates of swirl generators with holes for the entrance of fluid were investigated by placing them at the entrance section of inner pipe of heat exchanger. Various swirl generators having different arrangements of whole where used. Hot and cold water were passed through the inner pipe annulus respectively. Experiments were carried out for both parallel and counter flow models of the fluid at the Reynolds numbers between 8500-17500. Heat transfer, friction factor and exergy analyses were made for the conditions with and without swirl generators and compared to each other. Some empirical correlations expressing the results were also derived and discussed. It was observed that the Nusselt number could increase up to 130 % at a value of about 2.9 times increase in the friction factor by giving rotation to the air with the help of the swirl elements. The increase the dimensionless exergy loss was about 1.25 times in comparison with that for the inner pipe without swirl generators.

With the values obtained from the experimental data in inner pipe, the changes in the Nusselt numbers with the Reynolds numbers were drawn for various swirl generators containing circular holes at different diameter and number. The experiments were performed for both counter and parallel flow mode, and results were compared to those obtained from the empty tube and Dittus–Boelter correlation $Nu = 0.023Re^{0.8}Pr^{0.4}$ (describes non-swirling flow in the smooth-tube). Indeed, the empty tube has the tangential inlet, which generates the swirl flow. The swirl flow induced by the step-shape distribution of the vorticity has a set of axial velocity profiles under the same Reynolds and swirl numbers. This is the main distinction between flows with and without swirling.

Rennie et. al. [4] studied the double-pipe helical heat exchangers numerically and experimentally neglecting the effect of coiled tube pitch. Though the boundary condition of his work was different from the conventional boundary conditions of constant wall temperature and constant heat flux, however, it is obvious that the geometry of the double-pipe coiled tube heat exchanger is completely different from that of shell and coiled tube heat exchanger of present work. The purpose of the study was to determine the relative advantage of using a helically coiled heat exchanger versus a straight tube heat exchanger for heating liquids.

A double-pipe helical heat exchanger was numerically modeled for laminar fluid flow and heat transfer characteristics under different fluid flow rates and tube sizes. Two different tube diameters were used. The overall heat transfer coefficients were calculated for both parallel flow and counter flow. Validation of the simulations was conducted by comparing the Nusselt numbers in the inner tube with those found in literature; the results fell within the range found in the literature. The greatest thermal resistance was found in the annular region.

Kumar et. al.[5] investigated a tube-in-tube helically coiled heat exchanger for turbulent flow regime numerically. One of the most frequent uses of helically coiled tubes is in shell and coiled tube heat exchangers. Going through the existing literature, it was revealed that there are a few investigations on the heat transfer coefficients of this kind of heat exchangers considering the geometrical effects like coil pitch. In results shown that low Reynolds numbers, heat-transfer is 25% higher as compared to coiled tubes. At high Reynolds numbers, the configuration has less influence on heat transfer.

In the present study a tube-in-tube helically coiled (TTHC) heat exchanger has been numerically modeled for fluid flow and heat transfer characteristics for different fluid flow rates in the inner as well as outer tube. The three-dimensional governing equations for mass, momentum and heat transfer have been solved using a control volume finite difference method (CVFDM). The renormalization group (RNG) k–ε model is used to model the turbulent flow and heat transfer in the TTHC heat exchanger. The fluid considered in the inner tube is compressed air at higher pressure and cooling water in the outer tube at ambient conditions. The inner tube pressure is varied from 10 to 30 bars. The Reynolds numbers for the inner tube ranged from 20,000 to 70,000. The mass flow rate in the outer tube is varied from 200 to 600 kg/h. The outer tube is fitted with semicircular plates to support the inner tube and also to provide high turbulence in the annulus region. The overall heat transfer coefficients are calculated for both parallel and counter flow configurations. The Nusselt number and friction factor values in the inner and outer tubes are compared with the experimental data reported in the literature.

Eiamsa et. al. [6] in its present work experimentally investigates the heat transfer and friction characteristics in double pipe heat exchanger by inserting louvered strips. The louvered strip was inserted into the tube to generate turbulent flow which helped to increase the heat transfer rate of the tube. The turbulent flow devices consist of the louvered strips with forward or backward arrangements, and the louvered strip with various inclined angles (θ=15°, 25° and 30°), inserted in the inner tube of the heat exchanger. In the experiment, hot water was flowed through the inner tube whereas cold water was flowed in the annulus. The experimental data obtained were compared with those from plain tubes of published data. Experimental results confirmed that the use of louvered strips leads to a higher heat transfer rate over the plain tube. The increases in average Nusselt number and friction loss for the inclined forward louvered strip were 284% and 413% while those for the backward louvered strip were 263% and 233% over the plain tube, respectively. In addition, the use of the louvered strip with backward arrangement leads to better overall enhancement ratio than that with forward arrangement around 9% to 24%.

It is seen that the Nusselt number ratio value was high at lower Reynolds number and then rapidly decreased for the rise of Reynolds number. The Nusselt number ratios of all cases are higher than unity. This indicates to an advantageous gain of using the louvered strips over the plain tube, especially at low Reynolds number.

Swamee [7] presents the optimal design of the exchanger has been formulated as a geometric programming with a single degree of difficulty. The solution of the problem yields the optimum values of inner pipe diameter, outer pipe diameter and utility flow rate to be used for a double pipe heat exchanger of a given length, when a specified flow rate of process stream is to be treated for a given inlet to outlet temperature. Author has explained the optimal design procedure.

In the current literature the focus is on optimizing the area of the heat exchanger irrespective of the different flow rates of the utility that can be used. Using this pressure drop is not minimized to the fullest extent. This fact can be avoided through the design method discussed in the paper. For the optimal design of the exchanger, it is considered that its cost is optimized by considering three main parameters – the inner and outer diameter of the heat exchanger and the flow rate of the utility. It is assumed that the flow rate, the inlet and the required outlet temperature of the process fluid and the inlet temperature of the utility are known for the specific design of the exchanger.

Mansoor Siddique [8] presents the experimental investigation of double pipe heat exchanger with water as the cooling as well as the heating fluid for six sets of runs. The pressure drop data is collected under isothermal conditions. Data were taken for turbulent flow with $3300 \leq Re \leq 22{,}500$ and $2.9 \leq Pr \leq 4.7$. The main focus of the present study is to experimentally investigate the heat transfer and the pressure drop characteristics of a typical micro-fin tube and to develop accurate, simple and easy to use empirical design correlations for turbulent flow conditions in the range $3300 \leq Re \leq 22{,}500$. Heat transfer and pressure drop characteristics for turbulent flow inside a micro-finned tube were studied. It was observed that the micro-fins have a significant effect on the both the heat transfer rate and the pressure drop in such tubes. It was observed that, the friction factor it decreases with Reynolds number for $Re < 6000$, approximately constant for up to $6000 < Re < 11{,}800$ then resume decreasing for $Re > 11{,}800$. Beyond Rez11,800 the KAU correlation predicted friction factors are higher than the smooth tube Blasius equation predicted friction factors by nearly a constant factor of 2. The reported results presents correlation predicted friction factors values were nearly double that of the Blasius smooth tube correlation predicted friction factors. It was observed that the micro-fins have a significant effect on the both the heat transfer rate and the pressure drop in such tubes.

Shahiti et.al [9] presents the experimental study of heat transfer and pressure drop characteristics of a double-pipe pin fin heat exchanger. Pin fins are used in heat exchangers as very effective elements for heat transfer enhancement. Extensive work is being carried out to select and optimize pin fins for various applications.

The author develops the mathematical model of the entropy generation minimization for different heat exchanger flow lengths and different pin length. The reported conclusions are derived on the basis of the behaviour of the entropy generation Nu number as a function of Re. It could be demonstrated that for all the investigated flow length, an optimal region Re, which ensures a minimal N_S, could be found. Furthermore, based on the empirical correlations used for heat transfer and pressure drop it could be shown that shorter flow lengths are accompanied with lower entropy production rates, and hence these would be thermodynamically preferable. Hence, it can be concluded that no single optimal flow length exist, but this should be as smaller as possible taking into consideration the heat exchanger duty and the available frontal area It is shown that not all definition forms for the entropy generation number leads to the right conclusions. It also concluded that from thermodynamic view of point, larger number of passages with smaller pin height in the given frontal area of heat exchanger is more preferred than less heat exchanger passages with larger pin height.

III. EXPERIMENTAL SET-UP

Fig.A illustrates the experimental set-up used in the present study. The test section is a tube in tube heat exchanger. An experimental apparatus is conducted to study the heat transfer performance with micro fins inserts. It is composed of cold water tank, hot water tank, temperature sensors, flow control valve, temperature display, mono block pump, thermostatic water heater, and annulus.

A double pipe heat exchanger is utilized as the main heat transfer test section. It consists of two concentric tubes in which hot water flows through the inner tube and cold water flows in counter flow through annulus. The outer tube made of mild steel having inside and outside diameters of 22 mm and 25 mm respectively. The inner tube made of a cupper having inside and outside diameters of 10 mm and 13 mm respectively. It has a heat transfer section of a length of 2.5 meter.

The two flow meters are used to maintain shell side and tube side mass flow rates of water. The working range of flow meter is from 100 LPH to 600LPH. The two flow control valve was used to controlled tube side and annulus side mass flow rate. One flow meter used to measure hot water mass flow rate and another flow meter is to measure annulus side cold-water mass flow rates. A PT100 type temperature sensor is directly inserted into inner and outer tube to measure inlet and outlet temperatures of both the fluids.

Temperature data was recorded using data acquisition/switch unit.

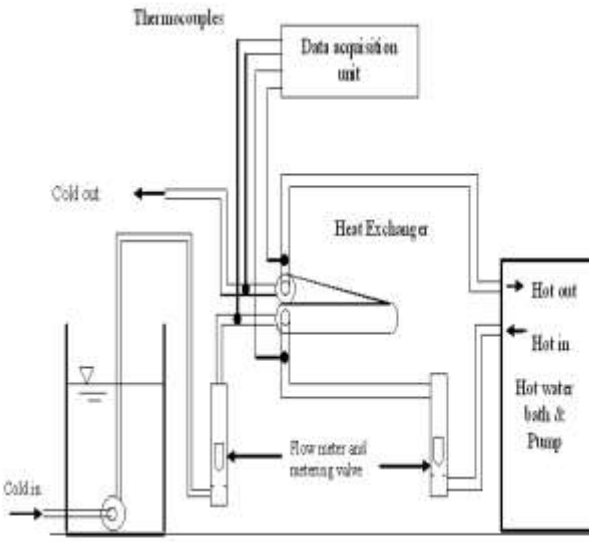

Fig.A

IV. RESULT AND DISCUSSION

The thermal performance of the double pipe heat exchanger is evaluated in heat transfer rate, overall heat transfer coefficients and Nusselt numbers. The tube side flow rate is varied from the 100 LPH to 600 LPH same time annulus side flow rate is maintained constant. The test is conducted for only the counter-flow configuration. Six annulus side flow rates taken i.e. 100, 200, 300, 400, 500 and 600 LPH.

Heat transfer rate with varying flow rates

1 For Constant Hot fluid for plain and finned heat exchanger:

Fig.1 shows that variation of heat transfer rate with varying cold flow rate by keeping constant hot fluid flow through plain heat exchanger.

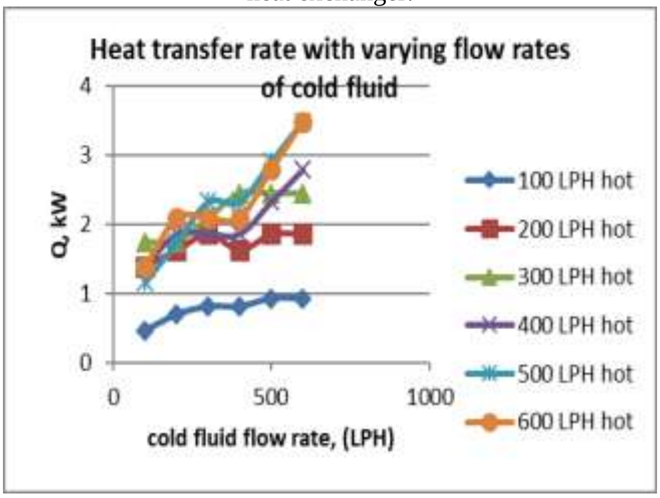

Fig. 1 variation of heat transfer rate with varying cold flow rate

When 100 LPH,200 LPH and 300 LPH hot fluid constantly flows through inner tube and varying flow rate from 100 LPH to 600 LPH of cold fluid through outer tube then small increase in heat transfer rate was observed but for 500 LPH and 600 LPH heat transfer rate more increases.

Fig. 2 shows that variation of heat transfer rate with varying cold flow rate by keeping constant hot fluid flow through finned heat exchanger.

For 100 LPH hot fluid constantly flows through inner tube and varying flow rate from 100 LPH to 600 LPH of cold fluid through outer tube then small increase in heat transfer rate was observed. For 200 LPH, 300 LPH and 400 LPH constantly increase in heat transfer rate. More heat transfer rate occurred for 600 LPH.

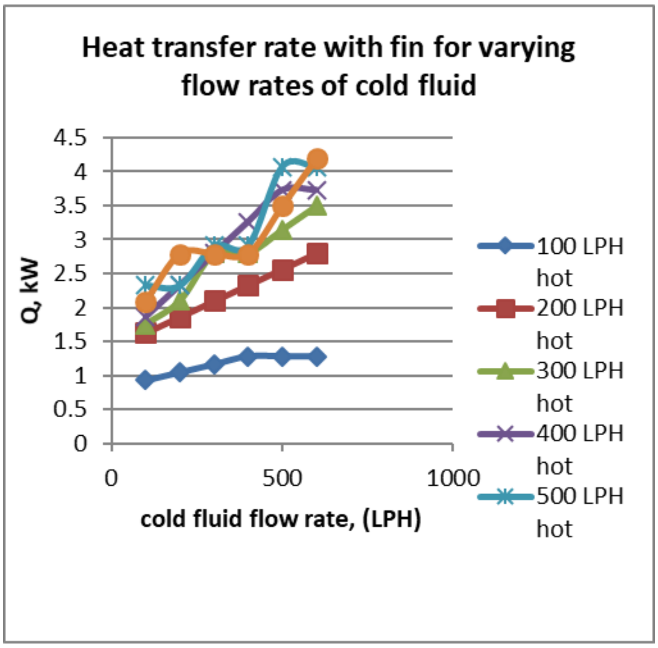

Fig. 2 Heat transfer rate with fin for varying flow rates of cold fluid

From fig. 1 and 2 it was clear that heat transfer rate increases by micro fins insertions for all fluid flow rates.

2. For constant cold fluid for plain and finned heat exchanger:

Fig. 3 shows that variation of heat transfer rate with varying hot flow rate by keeping constant cold fluid flow through plain heat exchanger.

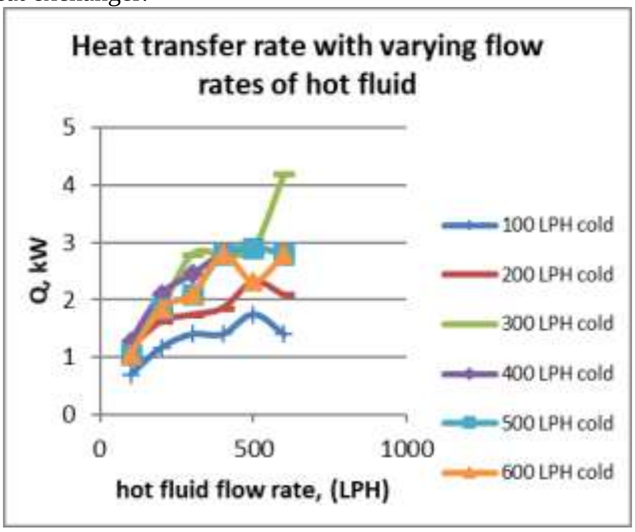

Fig. 3 Heat transfer rate with varying flow rates of hot fluid

When 100 LPH cold fluid constantly flows through outer tube and varying flow rate from 100 LPH to 500 LPH of hot fluid through inner tube then small increase in heat transfer rate was observed but for 600 LPH heat transfer rate decreases. For 200 LPH, 400 LPH and 600 LPH constantly increase in heat transfer rate. For 300 LPH heat transfer rate was more compare to 100, 200, 400, 500 and 600 LPH.

Fig. 4 shows that variation of heat transfer rate with varying hot flow rate by keeping constant cold fluid flow through plain heat exchanger.

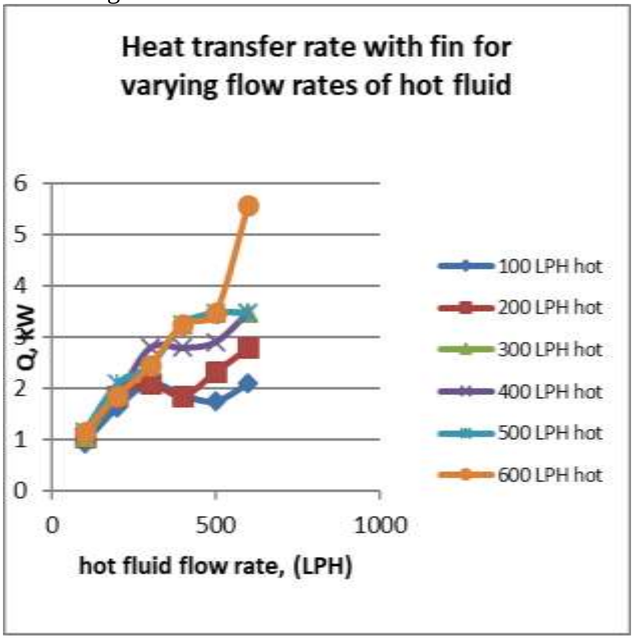

Fig. 4 Heat transfer rate with fin for varying flow rates of cold fluid

When 100 LPH, 200 LPH, 300 LPH, 400 LPH and 500 LPH cold fluid constantly flows through outer tube and varying flow rate from 100 LPH to 600 LPH of hot fluid through inner tube then small increase in heat transfer rate was observed but for 600 LPH cold fluid constantly flows through outer tube and varying flow rate from 100 LPH to 600 LPH of hot fluid through inner tube then sudden increase in heat transfer was observed for 600 LPH.

From fig. 3 and 4 it was clear that heat transfer rate increases by micro fins insertions for all fluid flow rates.

V. CONCLUSION

The conclusions from experimentation work are drawn as follows,

1. In a heat exchanger, while the inserts can be used to enhance the heat transfer rate, they also bring in an increase in the pressure drop. When the pressure drop increases, the pumping power cost also increases, thereby increasing the operating cost. So depending on the requirement, one of the above mentioned inserts can be used for heat transfer augmentation
2. The heat transfer rates for finned heat exchanger were 40% higher than the plain heat exchanger.
3. Flow rate did not affect the heat transfer coefficient, most likely from the fact that the flow was turbulent and increasing the flow rate does not greatly change the wall effects.
4. In a heat exchanger, while the inserts can be used to enhance the heat transfer rate, they also bring in an increase in the pressure drop. When the pressure drop increases, the pumping power cost also increases, thereby increasing the operating cost. So depending on the requirement, one of the above mentioned inserts can be used for heat transfer augmentation.
5. On the basis of performance evaluation criteria based on constant flow rate , we can say that the , the micro fin heat exchanger gives the highest maximum value of heat transfer rate 20% more than Plain tube.

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