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An Experimental Investigation for the Rate of Heat Transfer in Double Pipe Heat Exchanger with Fins on Inside Surface of Internal Tube

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***Abstract**: A set of experimental work is carried out to explore the rate of heat transfer and behavior of heat transfer analysis in turbulent flow pattern in counter flow arrangement. The experimental work conducted for both smooth and two enhanced tube using water as a working fluid in a same test section. The test will be accompanied for keeping mass flow of cold fluid in annulus side is constant by varying mass flow of hot fluid from 320 to 1080LPH using glove valve.The smooth tube having diameter of 25mm and two enhanced tubes of same diameter with V corrugations were made on inner surface with different pitch p=1.5mm and p=2mm and constant depth of 0.5mm.The increment of rate of heat transfer for both smooth and two enhanced given by 2.64kW, 3.34kW and 3.93kW respectively. The performance ratio for tube with p=1.5mm is rises with increasing Reynolds number; it ranges from 0.514 to 0.862 for different Reynolds number. Likewise, performance ratio for tube with p=2mm is given by 0.544 to 0.901.The friction factor for smooth tube is just about nearly equal. For corrugated tube of p=1.5mm, the friction factor ranging from 0.1967 to 0.0564 and p=2mm is ranges from 0.2541 to 0.0690.*

***Keywords**: Heat transfer rate. Enhanced tube, Reynold's number, Friction factor, Performance ratio.*

I. Introduction

Heat exchangers are the devices that assist the exchange of thermal energy among two fluids that are at different temperature without permitting them from mixing with each other. A heat exchanger is normally used to transmission of heat energy in among two fluids, which are separated by solid surface, and a liquid medium, or between solid matters and a fluid. A fluid may single mixture or compound mixtures and involved in different applications like evaporation or condensation in a different fluid stream (single pass or multicomponent pass) and cooling or heating application in heating system. In other applications involved to distill, pasteurize, concentrate, sterilize, fractionate, processing of fluid and crystalize. There are different types of heat exchanger developed for variety of technological uses and sophistication in size are chemical processing industries, pharmaceutical industries, steam power plants, food processing industries, radiators in automobile vehicle, air conditioning devices, radiator in space vehicles, and so on.

The design method for heat exchanger equipment is pretty complex matter. In designing factors like rate of heat transfer, friction factor, pressure drop estimation, analysis, sizing and performance assessment plays very significant role. Materials, energy, cost saving concerns and economic aspects are led produce more operational in designing of equipment. Developments of the designing of heat transmission equipment is to enable reduce its weight and size, to increase the heat transmission rate.The major principle of increasing the thermal efficiency of heat exchanger is given by artificial alteration of the thermal boundary layer of heat transmission wall surface The artificial amalgamation of wall surface promotes creation of high degree turbulence, thus due to creation of thinning viscous layer on the thermo-hydraulic boundary surface the proper mixing of hot & cold fluid under steady state. So, enhancement of rate of heat transfer by using surface corrugation is slightly inappropriate method due to increase of pressure drop and pumping cost.

II. Experimental Setup

The experimental apparatus comprises of double pipe heat exchanger, globe valve, rotameter, data acquisition system, thermocouple, U-tube manometer, heating coil for heating hot water, centrifugal pump is to deliver hot water to tube side, overhead tank for supply of cold water under action of gravity and hot water bath with submersion of heating coil is demonstrate in the above figure 1

Figure 1: Experimental setup

The experiment section is the one of part of experimental set up arrange in a counter flow pattern in heat exchanger with double pipe suitable connections with piping system. The setup consists of two circular tubes; inner tube (smooth or corrugated tube) possess material with high thermal conductivity and lower price as compared to other high thermal conductivity metals, so aluminum tube with length of 1.5m and outer diameter is 25mm used for hot water flow in tube side and outer tube for cold water flow in annulus side made of PVC pipe with effective length is 1.4mm and outer diameter is 40mm. The reason of using PVC material in annulus side is it possesses lesser thermal conductivity, so it act as insulator for this reason no insulation is needed for cold water side and it reduces heat loss to surrounding

The experiment is conducted for total three tubes; one smooth tube & other two tubes are corrugated tube of different pitch and all tubes were tested in same test section itself. The smooth tube is the reference tube compare with values obtained by both enhanced tubes. The working fluid selected as normal water because of available easily with less abundance in our environment. The normal water from the overhead tank is directly fed to the annulus part of test section and centrifugal pump is used to deliver hot water directly from hot water bath submersion with heating coil.

To measure the flow rate of working fluid two calibrated flowmeters, i.e. Rotameter were used, which is placed at both inlet section of cold & hot water side and it measure discharge valve ranging from 80-1080LPH for both inlet side. Rotameter operated manually and flow controls with help of glove valve placed inlet side of both hot and cold fluid. The cold fluid directly accessible from the overhead tank easily with assist under action of gravity and available with range of 29-31^{0}c with nominal range of atmospheric temperature for water obtainable. The hot water produced with help of placing heating coil into the bath and with the help of centrifugal pump hot water with temperature range around 65-70^{0}c is pumped to the tube part of the investigation section with assistance of replaceable piping system.

The calibrated K-type thermocouple used to temperature measurement with ranges from 0-220^{0}c, were placed in five positions namely; two at hot water inlet and outlet side, another two at cold water inlet and outlet side and one placed at hot water bath for hot water measurement. The pressure tapping of upstream and downstream for pressure drop measurement at hot water side the U-tube manometer were used measure pressure drop along tube side. The U-tube manometer is connected to both inlet & outlet condition of hot water with mercury used as a manometric fluid.

III. Results and Discussion

The experimentation conducted for both plain tube and two inner corrugated tubes. This section describes the compare results attained for together smooth and corrugated tube in turbulent region. The attained results relate with reference of taking smooth tube. The smooth tube having length 1.5m long and outer diameter is 25mm is used for the analysis. The two corrugated tubes must inner corrugation made on inner surface aluminum tube with constant depth of e=0.5mm and different pitch length is p=1.5mm and p=2mm.The results such as average heat transfer rate, convective heat transfer co-efficient, overall heat transfer co-efficient, friction factor, pressure drop and thermal performance factor is compared for corrugated tube with reference of variation Reynolds number on hot water side

(i) Heat Transfer Rate

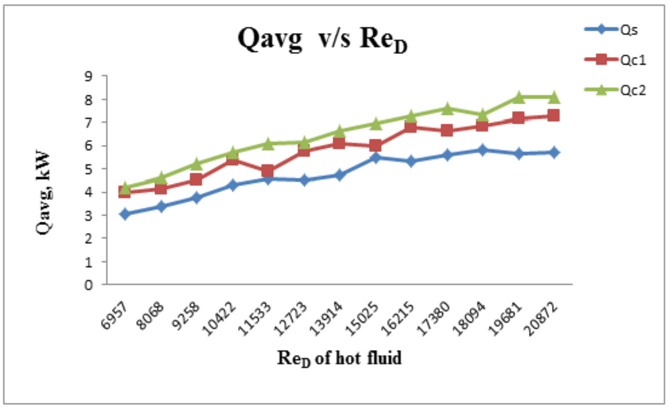

Figure 2: Average heat transfer rate v/s Reynolds number of hot fluid

The above achieved outcome debates the variation of average heat transfer rate along the change in Reynolds number in hot water side is as shown in the figure 2. The above graph demonstrations three set of data for Qs for average rate of heat transfer for smooth tube, Qc_1 for typical heat transfer data enhanced tube having p=1.5mm and Qc_2 typical heat transfer data for enhanced tube of pitch=2mm. For smooth tube, the average increment in heat transfer rate from 3.06kW to 5.7kW for Reynolds number 6957 to 20872 along the hot fluid side. The tube of pitch 1.5mm is also comprised for increase in typical heat transfer rate from 3.96kW to 7.3kW for the given range of Reynolds number. As corrugation made on inner wall surface, this leads to formation of thinning layer surface and also thermo hydraulic layer along the inner wall surface is decreases hence, very rapid variation shows in increasing heat transfer co-efficient helps to increase rate of heat transfer. Likewise, corrugated tube of pitch 2mm increases the heat transfer rate 4.17kW to 8.1kW for given range of Reynold number It is conclude that, as Reynold number rises average heat transfer coefficient also increases. The simultaneous increment of heat transfer rate for both smooth and two enhanced two given by 2.64kW, 3.34kW, 3.93kW respectively.

(ii) Convective Heat Transfer Coefficient

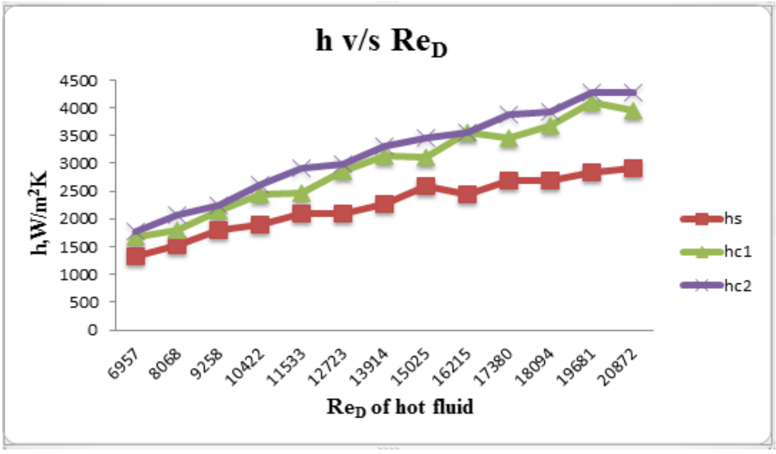

Figure 3: Convective heat transfer coefficient v/s Reynolds number of hot fluid

The above figure 3 indications of deviation of convective heat transfer coefficient with Reynolds number of hot fluid along the tube side. Whenever, corrugation made on inner surface of inner tube of heat exchanger, it leads to enhance heat transmission rate as well as convective heat transfer coefficient. For smooth tube, heat transfer coefficient intensifications from 29% to 64% for Reynolds number from 6957 to 20872 and with increasing Reynolds number. It is noted that enhanced tube of pitch 1.5mm, the increment from 37% to 88% and maximum augmentation achieve at higher value of Reynolds number because of inner surface corrugation on tube side. For the tube with pitch 2mm, the increment in heat transfer coefficient is achieved from 39% to 95%. The maximum heat transfer coefficient enhancement is occurred at tube of pitch=2mm, this is because of increasing the inner surface roughness average increment takes place. Moreover, for given surface roughness convective heat transfer coefficient is rises with cumulative Reynolds number on hot water side. As roughness along the heat transmission side increases, this prompts higher transport properties for fluid functioning on tube side and thinning on thermos-hydraulic boundary layer.

(iii) Pressure Drop Results

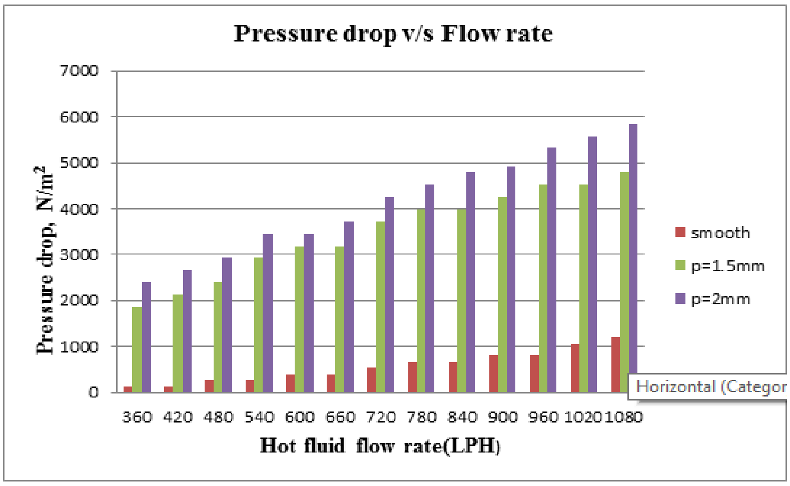

Figure 4: pressure drop v/s discharge rate of hot fluid

The above figure 4 illustrats deviation pressure drop in tube side v/s hot fluid volemetric flow rate. The pressure drop along smooth and two corrugated tubes is studied under different hot water mass flow circumstances along tube side. Pressure drop measure at both inlet and outlet condition of hot fluid side using mercury manometer. The variation of pressure drop for smooth tube is ranges from 133 to 1196 N/m^2 for hot fluid flow rate between 320 to 1080 LPH. Wherease, pressure drop along corrugated tube of p=1.5mm is ranging from 1861 to 4786 N/m^2 and corrugated tube of pitch p=2mm is ranges from 2392 to 5849 N/m^2 for different flow condition of hot fluid. As compared with the smooth, pressure drop along both corrugated tubes is significantly increases because of inner surface roughness made oninner wall surface. With cumulative tube surface roughness convinces more secondary viscosity effect and hence, leads to formation of more shear force in turbulent flow condition. The friction factor is upturn drastically with incresing pressure drop.

(iv) Friction Factor Results

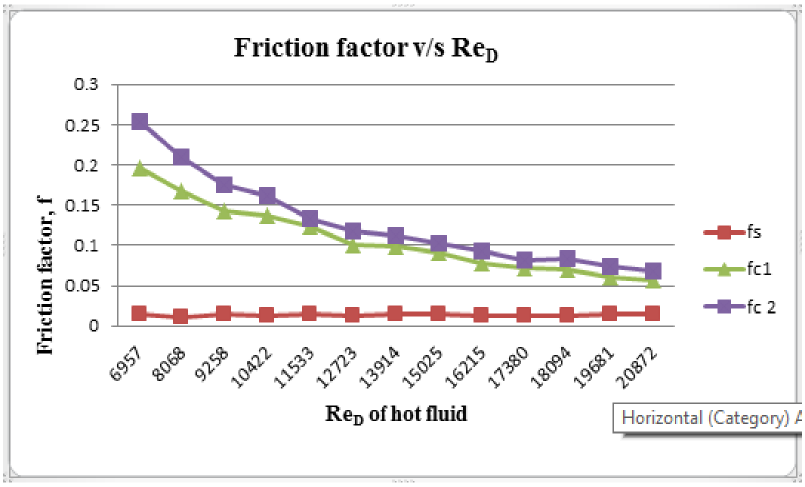

Figure 5: Friction Factor v/s Reynold Number for Hot Fluid

The above graph deliberates the difference of friction factor for both smooth and enhanced tubes with Reynold number of hot fluid can be seen in figure 5. Pressure drop across tube side measured with help of mercury manometer and friction factor is deliberate for tube side. The amendment in pressure drop across tube side is help to analyze, one of main hydrodynamic features such as friction factor (f). Friction factor supports to measures variation in kinetic energy in a system. It is notify that, with increasing Reynold number helps to reduction of friction factor, because of it is directly proportionate to alteration drop in pressure across tube side and inversely proportionate to square root of flow velocity. The friction factor for smooth tube is just about nearly equal, this is due to difference of pressure drop in smooth tube is very small related to that of corrugated tube. For corrugated tube of p=1.5mm, the friction factor ranging from 0.1967 to 0.0564 and friction factor for tube of p=2mm is ranges from 0.2541to 0.0690. It is clearly indicated that from close inspection, Reynold number upturn considerably friction factor drops. Moreover, hydraulic boundary layer thickness decreased due to cumulative Reynolds number hence indorses higher turbulent intensity.

(v) Thermal Performance Ratio

The thermal performance ratio given by ratio of augmentation of heat transfer rate for both corrugated tube to the modification in friction factor associated to that of smooth tube. As enhancement ratio is intensification with aggregate inner surface corrugation over inner wall surface with penalty on pressure drop of hot fluid. Usually, overall enhancement ratio is used to designate the feature of enhancement technique for together corrugated tubes.

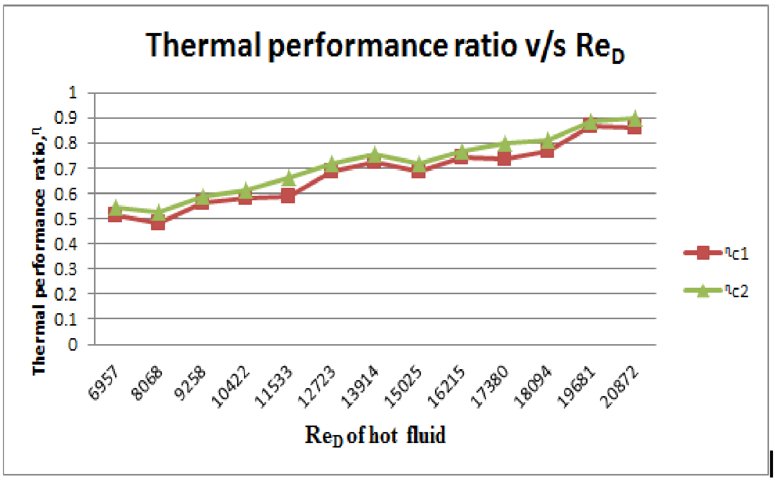

Figure 6: Thermal Performance Ratio v/s Reynolds Number of Hot Fluid

The deviation of thermal performance or overall enhancement ratio w.r.t Reynolds number of hot fluid flowing inside the tube for both corrugated tubes can seen in above figure 6. This technique is used compare the different outlines of enhancement technique and used to distinguish the passive technique. The performance ratio for tube with p=1.5mm is rises with increasing Reynolds number; it ranges from 0.514 to 0.862 for different Reynolds number. Likewise, performance ratio for tube with p=2mm is given by 0.544 to 0.901. From this closely investigate that, the maximum enhancement ratio achieve at highest value Reynolds number. It finally concludes that, for energy saving deliberation maximum value of thermal performance ratio is obtained at extreme value of Reynolds number.

IV. CONCLUSION

The experimental work is accompanied for both smooth and internally threaded aluminium tubes of different pitch and constant pitch using normal hard water as a working fluid. The work is completed for turbulent flow circumstances for different range of Reynolds number varying from 6957 to 20872 by the application of centrifugal pump. The parameters such as heat transfer rate, pressure drop, friction factor, overall heat transfer coefficient, thermal performance factor are acquired for the both corrugated tube is compare by reference of taking smooth tube. The experimental studies leads to drawn following conclusions based on information is obtained for heat rate and pressure drop.

- ➢ The average rate of heat transmission improved with increasing inner surface roughness. For smooth tube, the average increment in heat transfer rate from 3.06kW to 5.7kW and corrugated tubes of pitch p=1.5mm and p=2mm increment of heat transfer rate from 3.96kW to 7.3kW and 4.17kW to 8.1kW respectively for the range of Reynolds number 6957 to 20872 along the hot fluid side. The simultaneous increment of heat transfer rate for both smooth and two enhanced two given by 2.64kW, 3.34kW, 3.93kW respectively.
- ➢ The convective heat transfer coefficient for smooth tube increase from 29% to 64% and tube with pitch p=1.5mm and p=2mm it found to be in range of 37% to 88% and 39% to 95% respectively for given range of Reynolds number in turbulent flow pattern. As roughness along the heat transmission side increases, this prompts higher transport assets for fluid working on tube side and thinning on thermos-hydraulic boundary layer.
- ➢ The pressure drop along smooth and two corrugated tubes is inspected under different hot water mass flow conditions along tube side. The change of pressure drop for smooth tube is ranges from 133 to 1196 N/m^2 for hot fluid flow rate among 320 to 1080 LPH and tube of p=1.5mm is ranging from 1861 to 4786

N/m^2 and tube of p=2mm is ranges from 2392 to 5849 N/m^2 for different flow situation of hot fluid. Thus, With accumulative tube surface roughness induces extra secondary viscosity effect and hence, leads to development of more shear force in turbulent flow circumstance.

- The friction factor is increase drastically with cumulative pressure drop. The friction factor for smooth tube is almost approximately equal. For corrugated tube of p=1.5mm, the friction factor ranging from 0.1967 to 0.0564 and friction factor for tube of p=2mm is ranges from 0.2541to 0.0690. It is clearly designated that from close inspection, Reynold number intensification considerably friction factor declines.
- The thermal performance ratio for tube with p=1.5mm is increases with increasing Reynolds number; it ranges from 0.514 to 0.862 for different Reynolds number and performance ratio for tube with p=2mm is agreed by 0.544 to 0.901. From this closely investigate that, the maximum enhancement ratio achieve at highest value Reynolds number. It finally concludes that, for energy saving consideration extreme value of thermal performance ratio is obtained at extreme value of Reynolds number.

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