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Experimental Analysis of Heat Transfer Enhancement in Horizontal Circular Double Tube Heat Exchanger Using Snail

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ABSTRACT

In present study, in order to increase of heat transfer in a horizontal circular tube heat exchanger by means of Snail Entrance with air as the working fluid. In the experimental set up, cold air in ambient condition was passed through the inner pipe while hot water was flowing through outer tube. The Reynolds no of air varies from 10000 to 60000. The results are compared with the plain tube where no Snail Entrance was used. The work includes the determination of friction factor and heat transfer coefficient for snail entrance in both counter and parallel flow. Snail entrance was observed that the heat transfer coefficient could vary from 90 to 110% the smooth tube value but the corresponding friction factor increases by 1 to 1.3 times the smooth tube values. It was also observed that with an increase in Reynolds number (Re), the heat transfer coefficient increases where as the friction factor decreases.

KEYWORDS: Heat transfer Enhancement, Heat Transfer, Snail Entrance.

Introduction: -

In recent years considerable emphasis has been placed on the development of heat transfer enhancement techniques. Aydin Durmus et al, investigated heat transfer in concentric double pipe heat exchanger by passive method, snail type, swirl generator which is mounted at inlet and inside the inner pipe were used to augument heat transfer rate , because the swirl flow enhances the heat transfer mainly due to reduced boundary layer and increased resultant velocity . An augmentation of up to 120% in Nusselt number was obtained in the swirl flow for counter flow. [1] Ebru Kavak Akpinar, the study explored the effect of different helical wires on the heat transfer, friction factor and dimensionless exergy loss in a double concentric pipe heat exchanger. The key findings from the study may be summarized as follows: Heat transfer rates increased with decreasing pitch and with increasing helical number of the helical wires used in the experiments. The heat transfer rates in this heat exchanger increased up to 2.64 times with the help of the helical wires.[4] S.N. Sarada et al investigated on enhancement of turbulent flow heat transfer with mesh inserts in a horizontal tube under forced convection with air flowing inside are carried out with CFD analysis. [11] Sarac and Bali conducted experiments to investigate heat transfer and pressure drop characteristics of a decaying swirl flow by the insertion of vortex generators in a horizontal pipe at Reynolds numbers ranging from 5000 to 30000. It was observed that the Nusselt number increase ranging from 18% to 163% compared to smooth pipe. [6] Thianponget al, investigated Compound heat transfer enhancement of a dimpled tube with the heat transfer and friction factor are increase with decreasing both of pitch ratio (PR) and twist ratio (y/w). Depending on the pitch ratio and twist ratio, the heat transfer rate and friction factor in the dimpled tube with twisted tape, are respectively 1.66 to 3.03 and 5 to 6.31 times of those in the plain tube twisted tape swirl generator. [12]

THEORETICAL ANALYSIS

While the heat transferred to the cold fluid (i.e. air) is

$$Q_c = m_c Cp_c (T_{co} - T_{ci}) = hcA_i \Delta T_{mi}$$

The heat given by the hot fluid (i.e. water) at any Reynolds number is

$$Q_h = m_h C_{ph} (T_{hi} - T_{ho}) = h_h A_o \Delta T_{mo}$$

As usual, this heat may be expressed in terms of a heat transfer coefficient and tube logarithmic mean temperature difference ΔTm:

$$Q_t = hA\Delta T_m$$

By equalizing the energy loss of the hot fluid and the energy received by the cold fluid, convective heat transfer coefficients were deduced and Nusselt numbers were acquired as follows [1]

$$Nu = hD_{H/k}$$

For the hot and cold fluids, the Reynolds numbers are

$$Re = VD_{H/}\nu$$

Experimental Work

The apparatus consists of a blower unit fitted with a pipe, which is connected to the test section located in horizontal orientation. The experimental study on passive heat transfer augmentation using rectangular inserts was carried on in a circular double tube heat exchanger having the specifications are Inner pipe ID = 25mm, Inner pipe OD=28mm, Outer pipe ID =50mm, Outer pipe OD =56mm, Material of construction= GI (galvanized iron), Heat transfer length= 2.50m, and Pressure tapping to pressure tapping length = 2.42m. The experimental set-up used in this investigation is shown schematically in Fig. 1 In the experiment; precautions were taken to prevent leakages in the system.

NOMENCLATURE	
A	Heat transfer area(m^2)
Cp	Specific heat capacity (KJ/kg K)
D_H	Equivalent hydraulic diameter (m)
f	Friction factor
H	Height of air channel(m)
h	Convective heat transfer coefficient(W/m2°C)
k	Thermal conductivity (W/m °C)
m	Mass flow rate of air (kg/s)
Nu	Nusselt number
Pr	Prandlt number
Q	Heat transfer rate (KW)
Re	Reynolds number
T	Temperature (C)
ΔT_m	Logarithmic mean temperature difference (C)
V	Average axial velocity (m/s)
μ	Dynamic viscosity
v	Kinematic viscosity
Subscripts	
c	cold fluid
h	hot fluid
i	inlet
o	outlet
t	total

The inlet and outlet temperatures of the water and the air and of certain points along the outer surface of the pipes were measured with a multi-channel temperature measurement unit in conjunction with copper–constantan thermocouples. Pressure taps for measuring pressure losses were provided at the inlet and outlet ends of the pipes, and they were connected to the two U manometers, one of which was filled with water and used to measure the air-side pressure drop. The other was used for the water-side pressure drop, and its manometer liquid was mercury. In order to determine air flow rates, pressure taps were also mounted at Pitot tube, and they were connected to another U manometer filled with water. The set-up also incorporated a throttling valve and a Rotameter to control the water-flow rate, and necessary accessories (e.g. valves) to change the flow mode. In each experiment run, the data for temperatures, flow rates, and fluid pressure drops were recorded after steady-state was established. Reynolds numbers ranged from 2500 to 35000 and 10000 to 60000 for hot water and cold air, respectively. During the experiments, the hot-water temperature and cold-air temperature varied from 30 to 60 and 25 to 50°C, respectively. All fluid properties were determined at the overall bulk mean temperature.

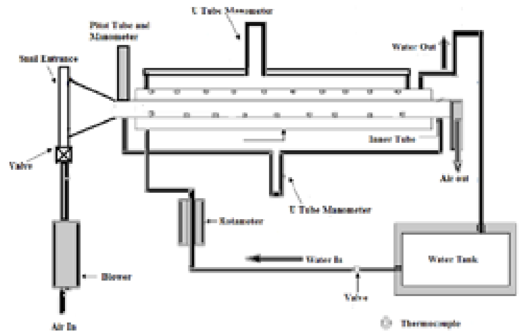

Fig 2 Experimental Set up

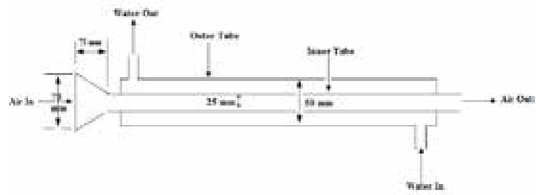

Fig 2 Test Section

Experimental Uncertainty: - Experiments were conducted initially for plain tube without snail entrance difference indicated in U-tube water manometer (with mass flow rate of air 0.0079 to 0.0178 m^3/sec). The Nusselt number obtained from experimental work is compared with the value obtained using Dittus-Boelter equation (theoretical).The experimental uncertainty is found as 10% for Nusselt number.

RESULT & DISCUSSION: - The most important aspects of this work were the extent of augmentation of heat transfer and increase in friction factor associated with the introduction of snail entrance into the air flow. The turbulent flow and heat transfer in the inner tube mounted with snail entrance was measured with air as working fluid. Experimentation is performed with two cases parallel flow and counter flow.

Validation of Plain Tube: - Preliminary experiments have been carried out on a plain tube in both parallel and counter flow to check the facility performance and to verify the measuring uncertainties. The experiments were carried out for a smooth tube to verify the validity. The values of friction factor and Nusselt number obtained from the experiments were compared with the values obtain from correlation of the Dittus - Boelter Equation for Nusselt number Nu = $0.023Re^{0.8}$ $Pr^{0.4}$ …… [1] And Karman–Nikuradse equation for friction factor f = $0.046Re^{0.2}$ ……. [2] Fig 4 shows the comparison of the dittus-Boelter equation and plain tube values of Nusselt number in counter and parallel flow it was found that the plain tube data agree reasonably well with the values predicted by correlation.

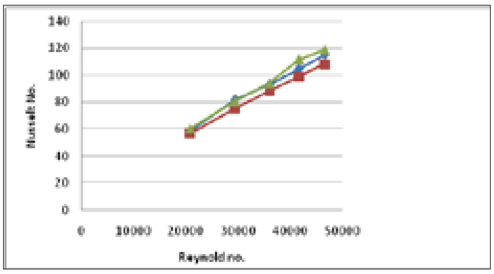

Fig 4 Comparison of experimental and theoretical values for plain tube

Snail Entrance Result: - Experiments were performed for parallel flow and counter flow results were compared to those obtained from plain tube, In the case of parallel flow and counter flow for snail entrance the average increase in Nusselt number was 110% in comparison with that for the smooth tube. The swirling flow gives higher values of Nusselt number than those for plain tube. Snail entrance increase in heat transfer rate ranges from 90 % to 110 % over the values obtained for plain tube in the Reynolds number range of 10000 – 60000. The increase in heat transfer with snail entrance is due to the higher swirl intensity imparted to the flow at the pipe inlet. The swirling motion of the fluid (air) results in a pressure gradient being created in the radial direction, thus affecting the boundary layer development.

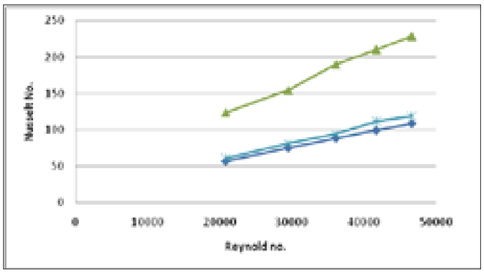

Fig. 5 Counter Flow Comparison for Snail Entrance, plain tube and theoretical values

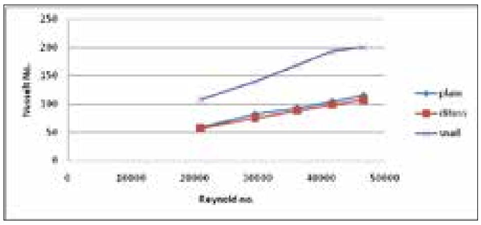

Fig.6 Parallel flow Comparison for Snail Entrance, plain tube and theoretical values

The increased rate of heat transfer in such flows is a consequence of the reduced boundary layer thickness and increased resultant velocity. From this figure 5 and 6 flow modes and increased with Reynolds number. With the values obtained from parallel flow and counter flow experimental data in inner pipe, the changes in Nusselt numbers with Reynolds numbers were drawn at air side, as shown in Figure. In the figure 5 and 6, the Nusselt number was related as a function of Reynolds number using the mass average velocity in the preliminary calculations. The results obtained for plain tube and predicted values are also plotted for comparison in fig 5 and 6. It is seen that the effect of applying snail entrance on the heat transfer rate is significant for all Reynolds numbers used due to the induction of high reverse flow (turbulence) and thin boundary layer. This technique results in an improvement of heat transfer rate over that of the plain tube.

Friction factor: - Figure 7 shows the plots of Snail Entrance values of the friction factor as the function of Reynolds number for smooth plate and Snail Entrance. It is clear that values of friction factor drop proportionally as the Reynolds number increases due to the suppression of viscous sub layer with increase in Reynolds number. The variation of pressure drop in terms of friction factor across the test section as a function of Reynolds number for snail entrance is presented in fig. 7. It can be seen that the friction factor obtained from snail entrance are in similar trend and decrease with increasing Reynolds number. The increase in friction factor with turbulent flow in general, is much higher than that with axial flow or plain tube flow. This is because of the dissipation of the dynamic pressure of the fluid due to higher surface area and the action caused by the reverse flow or turbulence. Moreover the pressure drop has a high possibility of occurring by the interaction of the pressure forces with inertial forces in the boundary layers it is seen from the figure that there is much reduction in the friction factor for using snail entrance.

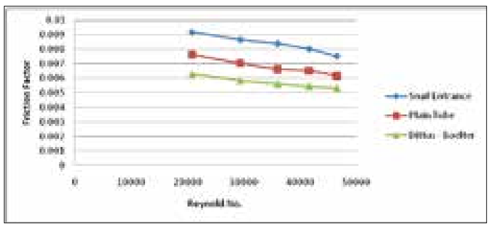

Fig 6 Comparison for Snail Entrance, plain tube and theoretical values

Conclusions

Experimental investigations on enhancement of turbulent flow heat transfer with Snail Entrance in a horizontal tube under forced convection with air flowing inside are carried out. The variations of temperatures, heat transfer coefficients, Nusselt number in the horizontal circular tube fitted with Snail Entrance have been studied. The maximum increase in Nusselt number of approximately 2.1 times was obtained through experimental investigation due to high resistance offered to air flow for Snail Entrance. Pressure drop using Snail Entrance was found to be maximum 1.7 times compared to that of plain tube.

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