ENHANCEMENT OF HEAT TRANSFER USING WIRE COIL INSERT IN TUBES

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ABSTRACT

This work presents an extensive experimental study on five wire coils of different pitch inserted in a smooth tube in laminar and transition regimes. Isothermal pressure drop tests and heat transfer experiments under uniform heat flux conditions have been carried out the air flow friction and heat transfer characteristics in a round tube fitted with coiled wire turbulators for the turbulent regime, Re = 2000 –10,000 and Pr = 0.7. The use of coiled circular wire causes a high pressure drop increase, which depends mainly on spring pitches and wire thickness the heat transfer in case of the conical coil is highest as compare to the plain pipe and the pipe containing the coil of different pitches. The enhancement efficiency increases with the decreasing pitches and found highest in the conical sets. The friction factor is highest around at a Reynolds number 2200- 3000

Keywords: Coil-wire insert; Heat transfer enhancement; Pressure drop, Friction factor, Heat transfer

INTRODUCTION

The heat transfer duty or thermal performance of heat exchangers can be improved by heat transfer enhancement techniques. Coil-wire insert has been used as one of the passive heat transfer enhancement techniques and are the most widely used tubes in several heat transfer applications, for example, heat recovery processes, air conditioning and refrigeration systems, chemical reactors, food and dairy processes. P. Bharadwaj[1] Passive heat transfer enhancement techniques (for example,wall roughness, swirl flow inducement) are preferred over active(for example, surface vibration, electro-static fields) ones to obtain more compact heat exchangers
and to reduce energy costs Naga Sarada[2] Among many techniques investigated for augmentation of heat transfer rates inside circular tubes, a wide range of inserts have been utilized, particularly when turbulent flow is considered. The inserts studied included twisted tape inserts, brush inserts, mesh inserts, strip inserts etc. the enhancement of heat transfer by using several coil-wire inserts based on exergy analysis. five different coil-wire inserts were tested in turbulent flow regions. Bergles presented the general correlations for the friction factor and heat-transfer coefficient for the single-phase turbulent flow in internally augmented tubes.

Enhancement techniques can be classified as passive methods, which require no direct application of external power, or as active schemes, which require external power. The effectiveness of both types depends strongly on the mode of heat transfer, which might range from single-phase free convection to dispersed-flow film boiling. Heat transfer enhancement can improve the heat exchange effectiveness of internal & external flow. Typically, they increase fluid mixing, by increasing flow vortices, unsteadiness, or turbulence or by limiting the growth of fluid boundary layers close to the heat transfer surfaces. The heat transfer in case of the conical coil is highest as compare to the plain pipe and the pipe containing the coil of different pitches. The enhancement efficiency increases with the decreasing pitches and found highest in the conical sets. The friction factor decreases with the increase of the Reynolds number. The conical set -1 is the curve on the top with respect to the other curves and it shows that with the increase of pitches the friction factor curve decreases. The friction factor is highest around at a Reynolds number 2200-3000.

LITERATURE REVIEW

P.Bharadwaj et.al He found that the pressure drop and heat transfer characteristics of flow of water in a 75-start spirally grooved tube with twisted tape insert was presented. Laminar to fully turbulent ranges of Reynolds numbers was considered[1]. Naga Sarada et.al In this Sixteen types of mesh inserts with screen diameters of 22mm, 18mm, 14mm and 10mm for varying distance between the screens of 50mm, 100mm, 150mm and 200mm in the porosity range of 99.73 to 99.98 are considered for experimentation. The Reynolds number is varied from 7000 to 14000. Correlations for Nusselt number and friction factor are developed for the mesh inserts from the obtained results. It is observed that the enhancement of heat transfer by using mesh inserts when compared to plain tube at the same mass flow rate is more by a factor of 2 times where as the pressure drop is only about a factor of 1.45 times[2]. Anil yadav The experimental results revealed that the increase in heat transfer rate of the twisted-tape inserts is found to be strongly influenced by tape-induced swirl or vortex motion. The heat transfer coefficient is found to increase by 40% with half-length twisted tape inserts when compared with plain heat exchanger[3]. Pongjet Promvonge et.al In the experiments, the swirling flow was introduced by using twisted tape placed inside the inner test tube of the heat exchanger with different twist ratios. The experimental results revealed that the increase in heat transfer rate of the twisted-tape inserts is found to be strongly influenced by tape-induced swirl or vortex motion[4]. The purpose of this study is to investigate the heat transfer and pressure drop characteristics in a wire coiled tube with different pitches. In the experiment, the turbulence flow near the tube wall is produced by using wavy-surfaced wall while the swirling flow is generated by inserting the helical-tape along the core region.
NOMENCLATURE

D  Diameter of test Pipe.
Do  Outer Diameter.
F  Friction factor.
Nu  Nusselt Number.
H  Heat transfer coefficient for air(W/m$^2$ K).
Pr  Prandtl number.
Δp  Pressure Drop(N/m$^2$).
Re  Reynold number.
T1,T2,T3  Tube wall Temperature(°C).
Tin,Tout  Air Temperature at inlet & outlet resp(°C).
Q  Volume flow rate.
q  Heat flux.

EXPERIMENTAL SET-UP AND PROCEDURE

The experiments were performed in an open loop experimental facility as shown in Fig. 1. The loop consisted of a 7.5 kW blower, an orifice meter to measure the flow rate, the calming tube (2500 mm) and the heat transfer test tube with a coiled wire insert. The copper test tube has a length of $L = 1250$ mm, with 47.5 mm inner diameter (D), 50.5 mm outer diameter (Do) and 1.5 mm thickness ($t$) as depicted in Fig. 2. The coiled wire was made of a small steel wire with circular and square cross sections. The cross section of wire turbulators of 2 and 3 mm wire thicknesses was given in Fig. 2. The coiled wires of different cross sections and two arrays (15 and 20 mm spring pitches) were inserted into the tube by wall attached position. The tube was heated by continually winding flexible electrical wire to provide a uniform heat flux boundary condition. The electrical output power was controlled by a varying dimmer stat to obtain a constant heat flux along the entire length of the test section and by keeping the current less than 3 A. Convective heat loss to the surroundings, and necessary precautions were taken to prevent leakages from the system. The inner and outer temperatures of bulk air were measured at certain points with a multi-channel temperature measurement unit in conjunction with type K thermocouples. Five thermocouples were placed on the local wall of the tube and thermocouples
were placed around the tube to measure the circumferential temperature variation, which was found to be negligible. The mean local wall temperature was determined by means of calculations based on the readings of the multi-channel thermocouples.

Air flow rate was measured by an anemometer. Water was used in U-tube manometers to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. Also, the pressure drop of the heat transfer test tube U-tube manometers. The volumetric air flow rates from the blower were adjusted by varying motor speed. During the experiments, the bulk air was heated by an adjustable electrical heater wrapping along the test section. Both the inlet and outlet temperatures of the bulk air from the tube were measured by thermocouples, calibrated within ±0.2°C deviation by thermostat before being used. It was necessary to measure the temperature at 3 stations altogether on the outer surface of the heat transfer test pipe for finding out the averaged Nusselt number. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk air at steady state conditions in which the inlet air temperature was maintained at 25°C. The various characteristics of the flow, the Nusselt's number, and the Reynolds numbers were based on the average of tube wall temperature and outlet air temperature. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature.

EXPERIMENTAL PROCEDURE

In the apparatus setting above, the inlet bulk air at 25 °C from a 7.5 kW blower was directed through the orifice meter and passed to the heat transfer test section. The air flow rate was measured by an anemometer. Water was used in U-tube manometers with specific gravity (SG.) of 0.826 to ensure reasonably accurate measurement of the low pressure drop encountered at low Reynolds numbers. Also, the pressure drop of the heat transfer test tube was measured with inclined U-tube manometers. The volumetric air flow rates from the blower were adjusted by varying motor speed through the inverter, situated before the inlet of test tube. During the experiments, the bulk air was heated by an adjustable electrical heater wrapping along the test section. Both the inlet and outlet temperatures of the bulk air from the tube were measured by multi-channel Chromel constantan thermocouples, calibrated within ±0.2°C deviation by thermostat before being used. It was necessary to measure the temperature at 3 stations altogether on the outer surface of the heat transfer test pipe for finding out the averaged Nusselt number. For each test run, it was necessary to record the data of temperature, volumetric flow rate and pressure drop of the bulk air at steady state conditions in which the inlet air temperature was maintained at 25°C. The various characteristics of the flow, the Nusselt's number, and the Reynolds numbers were based on the average of tube wall temperature and outlet air temperature. The local wall temperature, inlet and outlet air temperature, the pressure drop across the test section and air flow velocity were measured for heat transfer of the heated tube. The average Nusselt numbers were calculated and discussed where all fluid properties were determined at the overall bulk mean temperature.

HEAT TRANSFER CALCULATION

1. Mass = density of air × c/s area × air flow rate (kg/s)
2. \( Q_{\text{air}} = m \times C_{p_{\text{air}}} \times \Delta T_{\text{air}} \)
3. \( Q_{\text{conv.}} = h \ A \Delta T_{\text{conv.}} \)
4. \( \Delta T_{\text{air}} = T_{\text{out}} - T_{\text{in}} \)
5. \( T_{w} = (T_{1} + T_{2} + T_{3}) / 3 \)
6. \( T_{b} = (T_{\text{out}} + T_{\text{in}}) / 2 \)
7. \( \Delta T_{\text{conv.}} = T_{w} - T_{b} \)
8. \( h_{\text{exp.}} = (m \times C_{p} \times \Delta T_{\text{air}}) / (A \Delta T_{\text{conv.}}) \)
9. \( Re = (\rho v d) / \mu \)
10. \( Nu = 0.02 \times Re^{0.8} \times Pr^{0.4} \)
11. \( h_{\text{theoretical}} = (Nu \times K) / D \)
12. \( f = 2 \Delta P / (L/D^2) \rho v^2 \)

**RESULT & DISCUSSION**

**Fig No 2**

The Nusselt number of various coil spring pitches increases when compared with those from the plain tube. In the figure No. 2 coiled wire turbulators yield a considerable heat transfer enhancement with a similar trend in comparison with the smooth tube, and the Nusselt number from coils increases for rising Reynolds number. This is because the coiled wire turbulators interrupt the development of the boundary layer of the fluid flow and increase the degree of flow turbulence. It is worth noting that the coiled spring with lesser pitches provides higher heat transfer than the plane pipe for all Reynolds number values.

**Fig No 3**
The figure No. 3 shows that heat transfer coefficient have a gradual increase with the increase in the velocity of flow of air. The plain pipe has the lowest heat transfer coefficient and increase when the coil spring pitch decreases. The maximum heat transfer takes place in the conical set-2 as compare to the other coil spring. It shows that higher the turbulence higher will be the heat transfer rate.

Fig No. 4
This figure No. 4 shows the characteristic curves b/w the friction factor and the Reynolds number. It shows that the friction factor decreases with the increase of the Reynolds number. The conical set-1 is the curve on the top with respect to the other curves and it shows that with the increase of pitches the friction factor curve decreases. The friction factor is highest around at a Reynolds number 2200-3000.

Comparison of Temperature & Reynold Number

Fig No. 5
This figure No.5 shows the variation of the temperature and the Reynolds no. for a plain pipe. It shows the temperature at various locations at one point and at different Reynolds number. It indicates that when the temperature goes on decreasing from the inlet to outlet there is a heat transfer take place along the test tube.
This figure No. 6 shows the variation of the temperature and the Reynolds no. for a coil pitch 12mm. It shows the temperature at various locations at one point and at different Reynolds number. It indicates that when the temperature goes on decreasing from the inlet to outlet there is a heat transfer take place along the test tube.

This figure No. 7 shows the variation of the temperature and the Reynolds no. for a coil pitch 9mm. It shows the temperature at various locations at one point and at different Reynolds number. It indicates that when the temperature goes on decreasing from the inlet to outlet there is a heat transfer take place along the test tube.
This figure No.8 shows the variation of the temperature and the Reynolds no. for a coil pitch 6mm. It shows the temperature at various locations at one point and at different Reynolds number. It indicates that when the temperature goes on decreasing from the inlet to outlet there is heat transfer take place along the test tube.

![Temperature vs Reynolds No. (conical)](image)

**Fig No 9**

This figure No. 9 shows the variation of the temperature and the Reynolds no. for a conical spring. It shows the temperature at various locations at one point and at different Reynolds number. It indicates that when the temperature goes on decreasing from the inlet to outlet there is a heat transfer take place along the test tube.

Enhancement Efficiency v/s Reynolds Number

![Enhancement Efficiency vs Reynolds No.](image)

**Fig No 10**

The enhancement efficiency tends to increase with the increase of Reynolds number. Enhancement efficiencies varied between 0.78 and .98 for plain pipe and various coil spring. There is a small increase in the enhancement efficiency with the increase of the Reynolds no.
CONCLUSION

An experimental study has been performed to investigate the air flow friction and heat transfer characteristics in a round tube fitted with coiled square wire turbulators for the turbulent regime, Re = 2000 –10,000 and Pr = 0.7. The use of coiled circular wire causes a high pressure drop increase, which depends mainly on spring pitches and wire thickness, and also provides considerable heat transfer augmentations. However, the Nusselt number augmentation tends to decrease rapidly with the rise of Reynolds number. If wire coils are compared with a smooth tube at constant pumping power, an increase in heat transfer is obtained, especially at low Reynolds number. Although fairly large differences have been observed among the analyzed coil wires, their evaluated performance is quite similar. The coiled circular wire should be applied instead of the smooth one to obtain higher heat transfer and performance, leading to more compact heat exchanger.

We observed that the heat transfer in case of the conical coil is highest as compare to the plain pipe and the pipe containing the coil of different pitches.

The enhancement efficiency increases with the decreasing pitches and found highest in the conical sets. The friction factor decreases with the increase of the Reynolds number. The conical set -1 is the curve on the top with respect to the other curves and it shows that with the increase of pitches the friction factor curve decreases. The friction factor is highest around at a Reynolds number 2200- 3000. The heat transfer enhancement is well established and is used routinely in the power industry, process industry, and heating, ventilation, and air-conditioning.

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