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Experimental Investigation and Prediction of Heat Transfer in Turbulent Flow through Tube with Conical Ring Inserts

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Abstract

An experimental investigation has been carried out to study the effect of conical-ring inserts on heat transfer and turbulent flow friction in a horizontal circular tube. The conical-rings were of three different pitch ratios (PR = 0.0, 7.5 and 9.0) corresponding to three different number of conical-rings (N = 1, 2 and 3), respectively. The diameter ratio of the conical-rings was 0.5. The experiments were conducted for air as the working fluid with Reynolds number ranging from about $2x10^4$ to $6x10^4$ under uniform wall heat flux condition. The results revealed that as much as three-fold improvement in heat transfer coefficient might be achieved at the cost of increased pumping power. Over the range investigated, the maximum thermal performance factor of around 1.7 has been found at PR =7.5. Finally, a correlation has been developed for the prediction of Nusselt number in turbulent flow for tubes with conical-ring inserts.

Keywords: conical-ring insert, heat transfer, friction factor, pumping power, thermal performance factor.

1. Introduction

The efficiency and economic competitiveness of industrial processes depend, to a great extent, on the performance of heat exchangers. This performance can be improved by using various enhancement techniques [1]. The uses of turbulence promoters or turbulator devices are heat transfer enhancement techniques that have been widely employed for improving the heat transfer rate in heat exchangers. Typically, the turbulators increase fluid mixing, by increasing turbulence or by limiting the growth of fluid boundary layers close to the heat transfer surfaces [2]. The effects of reverse flow and boundary layer disruption by the turbulators are to enhance the heat transfer coefficient and momentum transfers. Various designs of turbulator devices are being used to improve the heat transfer rate such as V-nozzles, truncated hollow cones, conical nozzles, conical-rings (CR), conical wires, circular cross-sectional rings, etc.

A.A. Jadoaa [3] experimented with drilled conical-ring inserts for enhancing heat transfer rate in a constant heat flux circular tube. He showed that the average Nusselt number increases as the space length between the conical-rings decrease and as the Reynolds number increases. But the friction factor for drilled conical-ring is higher than the plain tube. Kongkaitpaiboon, V. et al [8] experimentally investigated the influences of the perforated conical rings (PCR). The perforated conical rings arranged in the diverging pattern were of three different pitch ratios and three different numbers of perforated holes. It was found that the PCR considerably diminishes the development of thermal boundary layer, leading to the heat transfer rate up to about 137% over that of the plain tube. P. Promvonge and S. Eiamsa-ard [9] experimentally investigated heat transfer and friction characteristics with combined conical-ring and twisted-tape insert in a uniform heat flux tube. The average Nusselt numbers for employing the conical-ring together with the twisted-tape for Y=3.75 and 7.5 respectively, are found to be 10% and 4% over that for using the conical-ring alone or to be about 367% and 350% over the plain tube.

As declared in the literature review, the survey shows that very limited data has been published on the heat transfer performance of tubes with conical-ring inserts in converging arrangement. However, any numerical investigation on conical-rings inserts has hardly been carried out for understanding its action on enhancing thermo-hydraulic performance.

The present work was, therefore, undertaken for fulfilling the following objectives:

1) To study the influence of conical ring inserts on heat transfer located at different pitch ratios inside a tube with properly fabricated experimental facility.
2) To determine experimentally different heat transfer parameters such as heat transfer coefficient, friction factor and pumping power at different Reynolds numbers for tubes with conical ring inserts and compare these parameters with that of the smooth tube.
3) To analyze the heat transfer performance by determining the thermal performance factors.
4) To develop a correlation as a recommendation for prediction of heat transfer in tubes with conical-ring inserts.

2. Experimental apparatus and procedure

Experimental apparatus

All Fig. 1 depicts the schematic view of an experimental setup. It mainly consists of a heat transfer test section with proper insulation, data acquisition system for displaying temperature values, a pitot tube and a high pressure induced draft fan. For the test section, the test tube was heated by continually winding flexible nichrome wire to provide a uniform wall heat flux boundary condition. The test tube is made of brass with a dimension of L = 1500 mm for length, D = 70 mm for inner diameter, D_o = 80 mm for outer diameter, and t = 5 mm for tube thickness. For keeping a uniform wall heat flux conditions along the entire length of the test section, the electrical power was controlled by a variable voltage transformer. The temperature distribution at the inner tube wall was measured using type K thermocouples which were tapped on the local wall of the tube. The outer surface of the test tube was well insulated to minimize convective heat loss to surroundings, and necessary precautions were taken to prevent leakages from the system. A traversing pitot tube was used to measure the air flow. The pressure drop was measured using a U-tube manometer.

In the experiments, all of the conical-ring inserts were located in converging conical-ring arrangements (CR array). The arrangement of these enhancement devices in the tube and their geometrical details are shown in Fig. 2. The conical-ring inserts were made of aluminum with 65 mm in length and its throat diameter was 32.5 mm (0.5D), with 2 mm thickness. Single conical-ring located at 30 mm from the test tube inlet, two conical-rings (N = 2) with pitch length, p = 630 mm (PR = 9.0) and three conical-rings (N =3) with pitch length, p = 525 mm (PR = 7.5) were used in the present work for comparison. It should be noted that to fix the conical-rings in the tube, they were fastened with two very smooth wire rods.

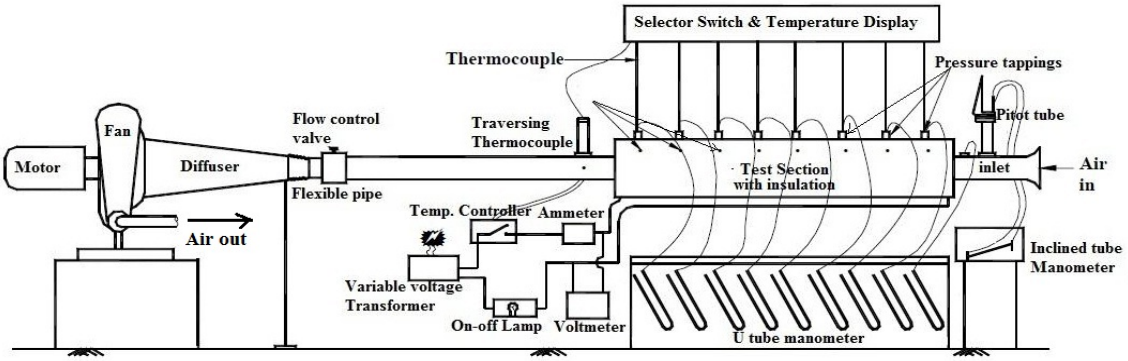

Fig. 1. Schematic diagram of the experimental setup

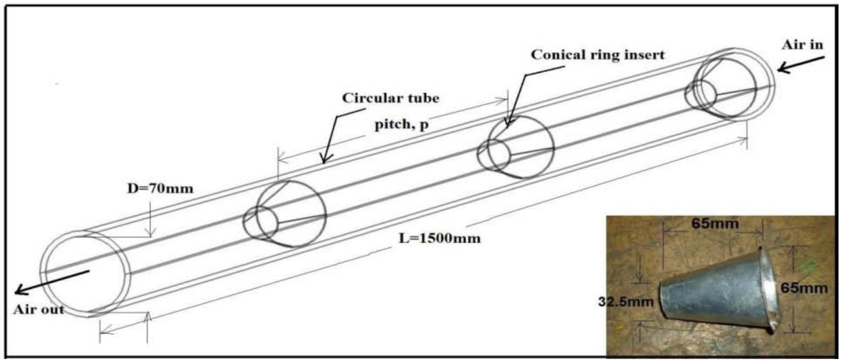

Fig. 2. Converging arrangement of three conical rings inside the test section

Experimental procedure

The fan was first switched on and allowed to run for about ten minutes to neutralize the transient characteristics. The flow of air through the test section was set to a desired value and was kept constant with the help of a butterfly valve. The electric heater was then switched on, adjusting the output level with the help of a regulating transformer, if necessary. At the steady-state condition thermocouple readings were recorded manually with the help of selector switch and at the same time, manometric readings were taken by the inclined manometer.

3. Data reduction

The convective heat flux is assumed to be uniform distribution over the heated wall tube and evaluated as:

$$Q = hA(T_w - T_b) = h(\pi D_i L)(T_w - T_b) \tag{1}$$

Where, h is the local heat transfer coefficient, Di is the inside diameter of the tube, L is the tube length, T_w is the local temperature of the inner wall surface, T_b is the bulk air temperature in the test section that is assumed to be linearly rising along the test section, whereas,

$$T_b = \frac{T_o + T_i}{2} \tag{2}$$

Thus the heat transfer coefficient can be written as follows,

$$h_x \frac{MC_{p,a}(T_o - T_i)}{\pi D_i L(T_w - T_b)}; h = \frac{\sum h_{1-8}}{8} \tag{3}$$

Where h is the average heat transfer coefficient, which is mean value of the 8 local points lined between the inlet and the exit of the test section and evaluated at the outer wall surface of the inner tube. The mass flow rate is calculated as follows,

$$M = \rho_a A_x v_m = \rho \frac{\pi D_i^{\,2}}{4} v_m \tag{4}$$

Where ρ_a is the density of air at room temperature, v_m is the mean velocity of air inside the tube and Ax is the cross-sectional area at the inlet of the tube. The average heat transfer coefficient is reported in term of Nusselt number, Nu which is defined as follows,

$$Nu = \frac{hD_i}{k} \tag{5}$$

Reynolds number is defined as,

$$\mathrm{Re} = \frac{\rho_b v_m D_i}{\mu_b} \tag{6}$$

Where ρ_b is the bulk air density, is the mean velocity and μ_b is the bulk air dynamic viscosity. An apparent friction factor, *f* can be evaluated from the following equation,

$$f = \frac{(\Delta P / L) D_i}{(1/2)\rho_b {v_m}^2} \tag{7}$$

Where L is the axial distance between two pressure taps and v_m is the mean velocity. All of the thermo-physical properties of air are determined at the overall bulk air temperature.

4. Results and discussion

Heat transfer results

Prior to the present work, Nusselt numbers for the smooth tube have been measured under a uniform heat flux condition and then compared with those obtained from the fundamental equation (8) by Dittus and Boelter in order to validate the present results.

$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{8}$$

The heat transfer results of the present work agree well with little discrepancies from 5% to 13%. In addition, the experimental results of the present smooth tube in term of the Nusselt number can be expressed as

$$Nu = 0.02838\,\mathrm{Re}^{0.769}\,\mathrm{Pr}^{0.33} \tag{9}$$

The relationship between Nusselt number and Reynolds number signifies that the Nusselt number increases with the Reynolds number due to the rise of mass transfer within the tube. Fig. 3 shows that the overall Nusselt number for inserted tube with single conical ring is about 27% to 200% higher than that of smooth tube over the Re range investigated. Conical-rings located at PR=7.5 inside the tube gives around 1.3 to 2.0 times higher heat transfer over the Reynolds number range investigated. The present effect of the pitch ratio (PR=0.0, 7.5 and 9.0) of the conical-ring inserts on the heat transfer enhancement are correlated in term of the Nusselt number for different Reynolds numbers as,

$$Nu = [10^{-5}.e^{0.412(PR)}].\,\mathrm{Re}^{[-0.021(PR)^2+0.147(PR)+1.531]}\,.\,\mathrm{Pr}^{0.33} \tag{10}$$

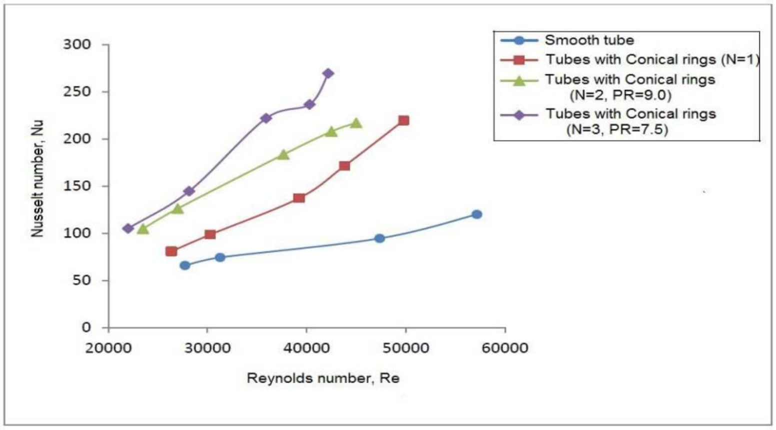

Fig. 3. Variation of overall Nusselt number with Reynolds number for both smooth tube and tubes with conical-ring inserts

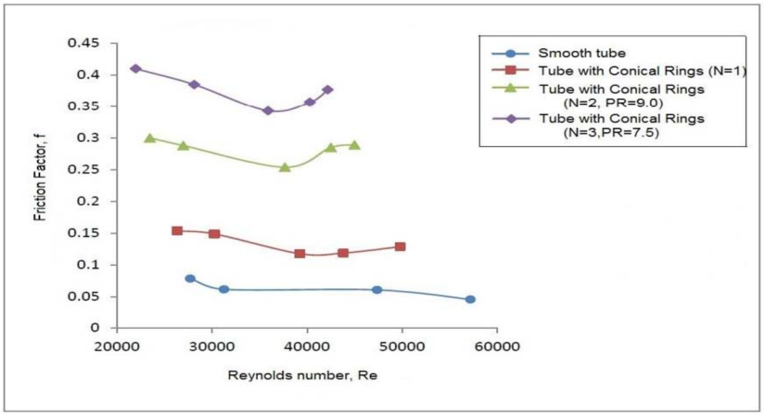

Fig. 4. Friction factor variation with Reynolds number for both smooth tube and tubes with conical-ring inserts

Friction factor results

Fig. 4 shows that the average friction factors for tubes with conical ring inserts are higher than that for the smooth tube at comparable Reynolds numbers. Evidently friction factor noticeably increases with increasing number of conical rings i.e. decreasing pitch ratio. This is due to the simple fact that the smaller distance between each pair of the conical rings, the more number of conical rings available in the tube, thus the more blockage against the flowing stream. The quantitative results show that the friction factor of the tube equipped with three conical rings (N=3) at PR=7.5 is approximately above 400% higher than that of the smooth tube.

Pumping power results

Fig. 5 exhibits that the conical-ring inserts enhances heat transfer at the cost of increased pumping power. It shows that the pumping power increases with the Reynolds number. Conical ring inserts with different pitch ratio also affects the pumping power requirement. It is observed from the Fig. 5 that the pumping power is required significantly higher, maximum in the case of maximum number of conical rings at lower pitch ratio. Extra blower power is required to overcome adversely increasing.

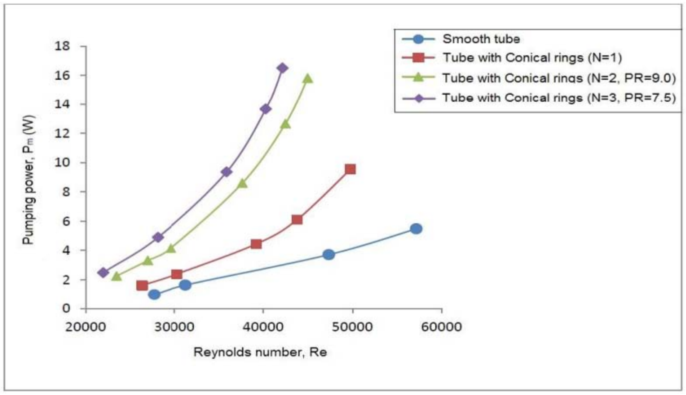

Fig. 5. Pumping power variation with Reynolds number for both smooth tube and tubes with conical-ring inserts

Thermal performance factor

According to the results shown above, the CR inserts offers heat transfer enhancement in accompany with the increase of friction factor. The increase of friction causes a rise of pumping power. Therefore, the actual effectiveness of the CR inserts depends upon the weight of the increase in heat transfer and the increase in friction which can be determined from performance evaluation. The thermal performance factor (η) at constant pumping power is the ratio of the convective heat transfer coefficient of the tube with heat transfer enhancement device (CR) to the plain tube.

$$\eta = \frac{(Nu_{CR} / Nu_s)}{(f_{CR} / f_s)^{1/3}} \tag{11}$$

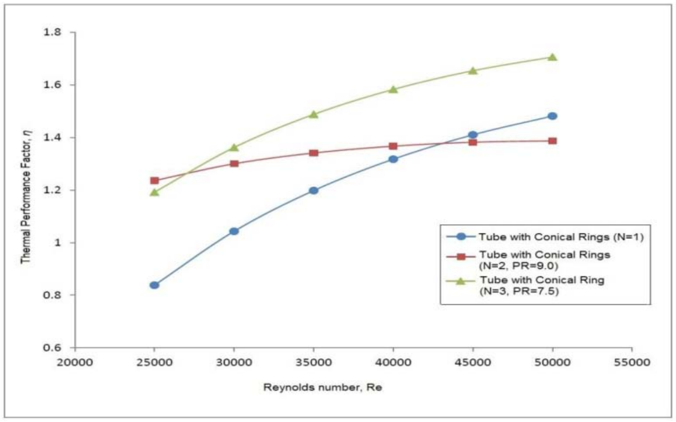

Fig. 6. Comparison of thermal performance factors among tubes with different arrangements of conical-ring inserts with respect to Reynolds number

It is observed from Fig. 6 that the thermal performance factor follows an increasing trend with the Reynolds number. For single conical ring insertion, the thermal performance factor has been found below unity signifying uneconomical enhancement at the lower Reynolds number regime, which is not the case for conical rings at PR=9.0 and 7.5. At the higher Reynolds number regime, thermal performance factor has been found maximum of above 1.6 for three conical ring inserts (N=3) located at PR=7.5. From Fig. 6 it is also seen that the thermal performance factor for two conical rings (N=2) is neither lesser than three rings (N=3) nor better than single ring (N=1) insertion in between the range of about Re=27000 to 43000, respectively.

5. Summary

Hydro-dynamically fully developed and thermally partially developed flow through circular tube has been experimentally investigated in order to test the heat transfer enhancement efficiency of conical ring inserts. Performance of the conical ring inserts has also been evaluated. The gists of the present study are given below:

1) The average heat transfer coefficient and overall Nusselt number increases with Reynolds number. The overall Nusselt number has been found to increase by about 140% to 300% revealing significant heat transfer enhancement with conical ring inserts compared to smooth tube.

2) The friction factor is found to increase with conical ring inserts by 1 to 4 times compared to the smooth tube.

The pumping power is found to increase to 1.67 to 5.33 times with conical ring inserts compared to that required for smooth tube.

3) The thermal performance factor is found maximum of above 1.6 for three conical ring inserts with pitch ratio 7.5. The thermal performance is found to be better for two conical ring inserts with pitch ratio 9.0 than that of 1 conical ring insert in the Reynolds number of range of about 27000 to 43000.

The conical-rings increase fluid mixing, by increasing turbulence or by limiting the growth of fluid boundary layers close to the heat transfer surfaces. The effects of reverse flow and boundary layer disruption by the rings enhance the heat transfer coefficient and momentum transfers at the expense of pressure drops.

6. Recommendation for future work

The present study has been carried out for conical-ring inserts with converging arrangements. These investigations may also be done with diverging or converging-diverging arrangements. Conical ring inserts with different diameter ratios may be implemented to study the heat transfer enhancement as well.

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