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Experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings ☆

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ABSTRACT

Perforated conical-ring (PCR) is one of the turbulence-promoter/turbulator devices for enhancing the heat transfer rate in a heat exchanger system. In the present paper, the influences of the PCR on the turbulent convective heat transfer (Nu), friction factor (f) and thermal performance factor (η) characteristics have been investigated experimentally. The perforated conical-rings (PCRs) used are of three different pitch ratios (PR$=p/D=4$, 6 and 12) and three different numbers of perforated holes ($N=4$, 6 and 8 holes). The experiment conducted in the range of Reynolds number between 4000 and 20,000, under uniform wall heat flux condition and using air as the testing fluid. The experimental results obtained by using the plain tube and the tube equipped with the typical conical-ring (CR) are also reported for comparison. It is found that the PCR considerably diminishes the development of thermal boundary layer, leading to the heat transfer rate up to about 137% over that in the plain tube. Evidently, the PCRs can enhance heat transfer more efficient than the typical CR on the basis of thermal performance factor of around 0.92 at the same pumping power. Over the range investigated, the maximum thermal performance factor of around 0.92 is found at PR$=4$ and $N=8$ holes with Reynolds number of 4000.

1. Introduction

Heat transfer enhancement (HTE) techniques with a passive method have been developed and applied in many engineering applications. Among the enhancement techniques, the use of turbulence promoters or turbulator devices is one of HTE technique that has 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. Various styles of turbulator devices have been applied to improve the heat transfer rate in a heat exchanger such as truncated hollow cone, V-nozzle, conical-nozzle, conical-ring, coiled wire, circular cross sectional ring, etc. [1–13]. Yakut et al. [9] reported the influence of conical-ring turbulator on the heat transfer enhancement and fluid flow in tubes under a uniform heat flux condition with different pitch ratios. It was observed that the turbulator with the smallest pitch ratio offered the highest heat transfer enhancement and thermal performance factor. Yakut and Sahin [10] demonstrated that the conical-ring also produced the vortices in the tube flow. Promvonge [11] investigated the effect of the conical ring turbulator arrangements (converging conical ring, referred to as CR array, diverging conical-ring, DR array and converging–diverging conical-ring, CDR array) on the heat transfer rate and friction factor. The results revealed that the conical-ring with the DR array provided superior thermal performance compared to those with the CR and CDR arrays.

Durmus [12] used conical turbulators with four conical angles (5°, 10°, 15°and 20°) for heat transfer enhancement. Apparently, the heat transfer rates as well as friction coefficients increased with increasing turbulator angles. Another attempt, Promvonge and Eiamsa-ard [13] combined effects of the reverse flow (conical-ring) together with swirl flow (twisted-tape) to improve the heat transfer in a circular tube. Their finding showed that average heat transfer rate in the tube with the combined devices was approximately 10% over that in the tube with the conical-ring alone.

As declared in the literature review, only typical conical-rings for different arrangements and pitch ratios have been investigated. The present work proposes the newly invented perforated conical-rings (PCR) for heat transfer enhancement in a circular tube. Effect of the number of perforated holes ($N=4$, 6 and 8 holes) and pitch ratios (PR$=4$, 6 and 12) on the heat transfer enhancement are investigated under uniform heat flux conditions. In the experiments, air is used as the testing fluid in turbulent regime with the Reynolds number between 4000 and 20,000.

2. Theoretical analysis

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

$$Q_{\text{conv}} = hA(\tilde{T}_{\text{w}} - T_{\text{b}}) \tag{1}$$

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Nomenclature

A	Heat transfer surface area, m^{-2}
$C_{p,a}$	Specific heat capacity of air, kJ kg^{-1} K^{-1}
D	Diameter of the test tube, m
f	Friction factor
h	Heat transfer coefficient, W m^{-2} K^{-1}
I	Current, A
k	Thermal conductivity of air, W m^{-1} K^{-1}
L	Length of the test tube, m
M	Mass flow rate, kg s^{-1}
N	Number of perforated hole
Nu	Nusselt number
Q	Heat transfer rate, W
p	Pitch length of the PCR arrangement, m
ΔP	Pressure drop, Pa
PR	Pitch ratio $= p/D$
Pr	Prandtl number
Re	Reynolds number
$\tilde{T}$	Mean temperature, K
T	Temperature, K
t	Thickness of the test tube, m
U	Mean axial velocity, m s^{-1}
V	Voltage, V
$\dot{V}$	Volumetric flow rate, m^3 s^{-1}

Greek symbols

μ	Dynamic viscosity, Ns m^{-2}
η	Thermal performance factor
ρ	Fluid density, kg m^{-3}

Subscripts

a	Air
b	Bulk
conv	Convection
i	Inlet
o	Outlet
p	Plain tube
t	Turbulator
w	Wall

where h is the local heat transfer coefficient, T_w is the local surface wall temperature and T_b is the bulk air temperature in test section that is assumed to be linearly risen along the test tube, whereas,

$$T_b = (T_o + T_i) / 2 \tag{2}$$

In the present work air is used as a working fluid and flowed through an insulated tube under uniform heat flux conditions. The heat transfer rate at steady state is assumed to be equal to the heat loss from the test section which can be expressed as:

$$Q_a = Q_{conv} \tag{3}$$

where

$$Q_a = MC_{p,a}(T_o - T_i) = VI \tag{4}$$

As found, the heat absorbed by the fluid for thermal equilibrium test is within 5% lower than the heat supplied by electrical winding in the test tube due to convection heat losses from the test section to surroundings. For data analysis, only the heat transfer rate absorbed by the air is taken for internal convective heat transfer coefficient calculation. The heat transfer coefficient can be written as follows:

$$h = MC_{p,a}(T_o - T_i) / A(\tilde{T}_w - T_b) \tag{5}$$

$$h = \sum h_{1-15} / 15 \tag{6}$$

where h is the mean heat transfer coefficient, which is average value of the 15 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 mean heat transfer coefficient is reported in term of Nusselt number, Nu is defined as

$$Nu = hD / k \tag{7}$$

Reynolds number is defined as

$$\mathrm{Re} = \rho UD / \mu \tag{8}$$

An apparent friction factor, f can be evaluated from the following equation:

$$f = (\Delta P / L)D / (\rho U^2 / 2) \tag{9}$$

where L is the axial distance between the two pressure taps and U is mean velocity in the tube given by:

$$U = M / \rho A \tag{10}$$

Apparent friction factors are based on the isothermal condition or without heating condition. All of thermo-physical properties of air are determined at the overall bulk air temperature.

3. Experimental rig

In the experiments, all of the perforated conical-ring (PCR) turbulators were located in diverging conical-ring arrangements (DR array). The arrangement of these enhancement devices in the tube and their geometrical details is shown in Fig. 1. The conical-ring was made of aluminum with 62 mm in length and its throat diameter was 31 mm (0.5D), with 2 mm thickness. The PCRs with different numbers of perforated holes, $N = 4$, 6 and 8 holes with 5 mm in diameter, and also different pitch lengths, $p = 248$ mm (PR = 4), $p = 372$ mm (PR = 6), and $p = 744$ mm (PR = 12) were used in the present work for comparison. It should be noted that to fix the PCRs in the tube, the PCRs were fastened with four small wire rods.

Fig. 2 depicts the schematic view of an open-loop experimental facility for the present study. The experimental facility consists of a heat transfer test section, enhancement devices, data acquisition system, an orifice meter and a high pressure blower. For the test section, the test tube was heated by continually winding flexible electrical wire with constant wire pitch length to provide a uniform wall heat flux boundary condition. The test tube is made of copper with a dimension of $L = 1500$ mm for length, $D = 62$ mm for inner diameter, $D_o = 65$ mm for outer diameter, and $t = 1.5$ mm for tube thickness. For keeping a uniform wall heat flux conditions along the entire length of the test section, the electrical output power was controlled by a variac 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. Thermocouples were also placed round the tube to measure the circumferential temperature variation, which was found to be negligible. The inner temperature of the air was measure at the entry the test tube or the end of the calming section using resistance temperature detector (RTD). At the exit of the test section, temperature of the outlet air was measured with three RTDs which located in the mixing section.

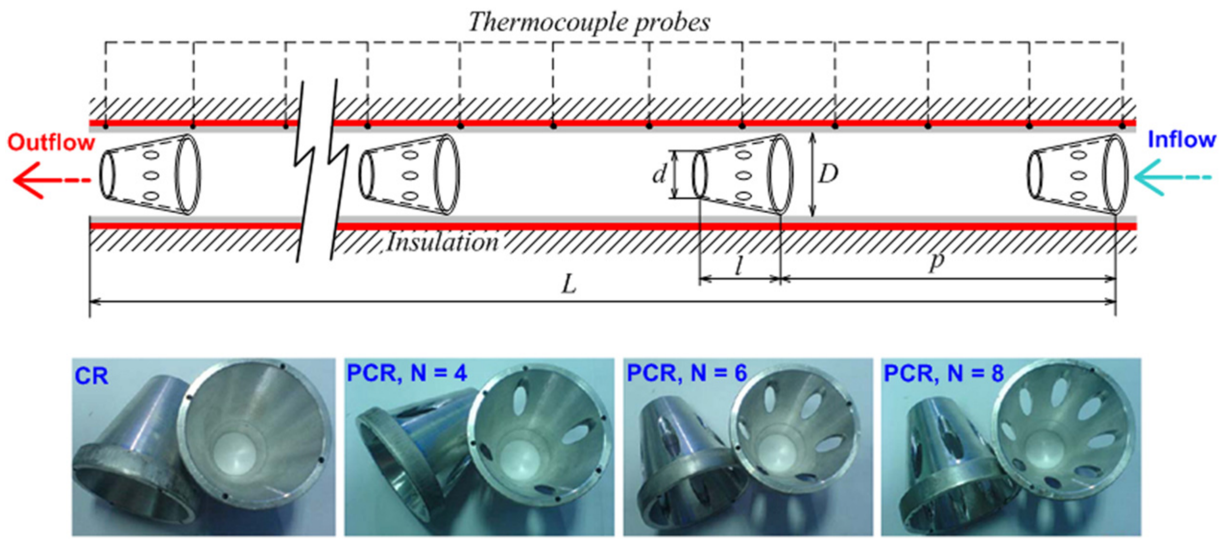

Fig. 1. The arrangement of CR/PCR in a round tube.

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. The pressure drop was measured using static pressure device, one static pressure tap was located upstream of the test section while another tap is located downstream of the test section. In the experiments, an orifice meter was employed to measure the volumetric flow rate at the entry of the test section in which the fluid has the density of around 826 kg/m^3. The Reynolds number of the working fluid (air) ranged from 4000 to 20,000 based on the bulk mean properties and the internal diameter of the tube. The detail of the experimental procedure used in this study was described elsewhere [13].

In order to quantify the uncertainties of measurements, the reduced data obtained experimentally were determined based on Ref. [14]. The maximum uncertainties of non-dimensional parameters are $\pm 5\%$, $\pm 10\%$ and $\pm 14\%$ for Reynolds number, Nusselt number and friction factor, respectively. The uncertainty in the axial velocity measurement was estimated to be less than $\pm 7\%$, and pressure has a corresponding estimated uncertainty of $\pm 5\%$, whereas the uncertainty in temperature measurement at the tube wall was about $\pm 0.5\%$. The experimental results were reproducible within these uncertainty ranges.

4. Results and discussion

4.1. Heat transfer results

Prior to the main experiment, Nusselt numbers for the plain tube are measured under a uniform heat flux condition and then compared with those obtained from the fundamental Eq. (11) by Dittus and Boelter [15] in order to validate the present plain tube.

$$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{11}$$

Fig. 3 reveals that the heat transfer results of the present work agree well with those obtained from Eq. (11) with the discrepancies of less than $\pm 6\%$. In addition, the experimental results of the present plain tube in term of the Nusselt number can be expressed as

$$Nu = 0.057\mathrm{Re}^{0.709}\mathrm{Pr}^{0.4} \tag{12}$$

It is noteworthy that relationships between Reynolds numbers and Nusselt numbers from both equations are in the same trend in which, the Nusselt numbers increase with increasing Reynolds numbers due to the rise of mass transfer within the tube.

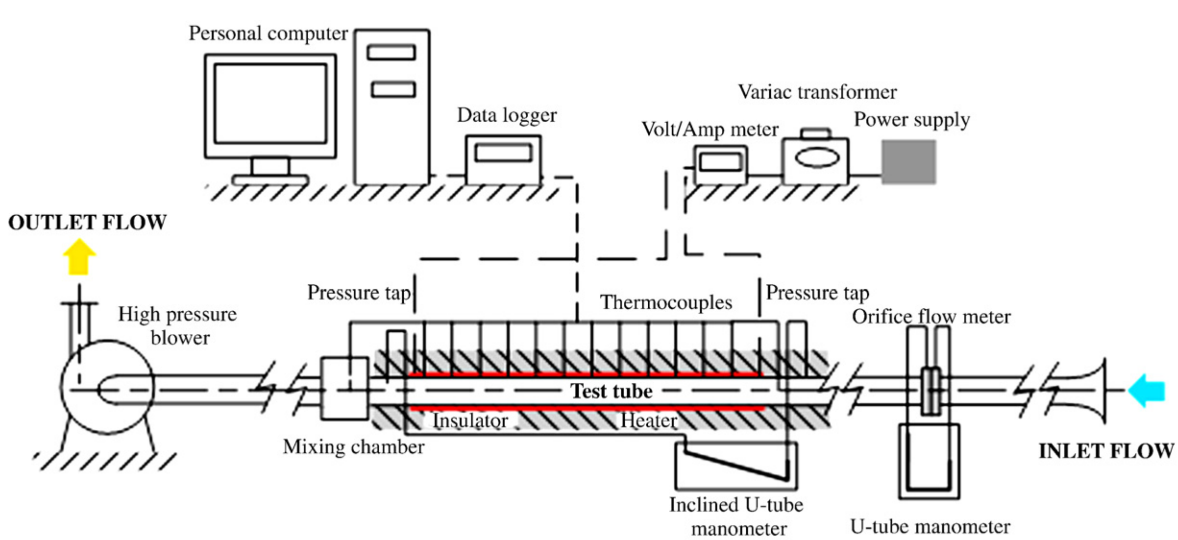

Fig. 2. Schematic diagram of experimental heat transfer apparatus.

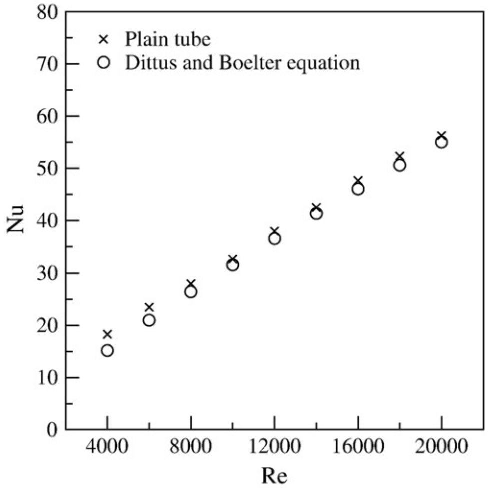

Fig. 3. Nusselt number verification for plain tube.

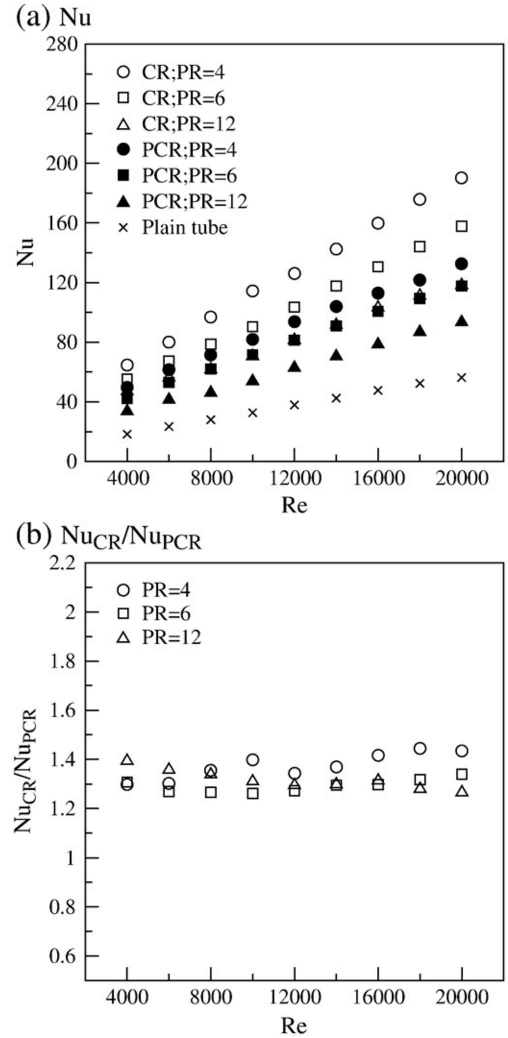

Fig. 4. Effect of pitch ratio (PR) on heat transfer enhancement for PCR with $N=4$ holes: (a) Nu and (b) Nu_{CR}/Nu_{PCR}.

4.1.1. Effect of pitch ratio

Effect of the tube with PCR with $N=4$ holes at different pitch ratios (PR = 4, 6 and 12) on the heat transfer rate (Nu) is demonstrated in Fig. 4(a–b). From Fig. 4(a), the heat transfer rates for the tube equipped with PCRs are higher than those found from the plain tube for a given Reynolds number. This is due to the interruption of flow by the turbulators which results in the destruction of thermal boundary layer near the tube wall. It is also found that the heat transfer rate increases with the decreasing pitch ratio (PR) for all turbulator arrangements. The quantitative results reveal that the mean heat transfer rate the in the tube with PCR at smallest pitch ratio (PR = 4) is 18.8%, 53.9% and 137% higher than those in the tube with PCRs at PR = 6, 12 and plain tube, respectively. In addition, the typical conical-rings (CR) provide higher heat transfer rates than those offered by the PCRs (Fig. 4(b)). The mean heat transfer rates the in the tube with CRs for all pitch ratios studied are around 1.26–1.44 times of those in the tube with PCRs.

4.1.2. Effect of number of hole in PCR

Effect of the number of the perforated holes ($N=4$, 6 and 8 holes) on the heat transfer rate is depicted in Fig. 5. Obviously, the PCR with larger number of perforated holes (N) provides a lower heat transfer rate due to the lower turbulence intensity in the tube. The Nusselt numbers in the tube equipped with the PCRs of $N=4$, 6 and 8 holes, are respectively 90–239%, 69–220% and 65–172%, higher than those in the plain tube. In contrast, the Nusselt numbers in the tube equipped with the PCRs of the same range of the number of the perforated hole (N) mentioned above are respectively, 2–11.6%, 9.5–21.3% and 20.7–30.8% lower than those in the tube with typical CR, depending on pitch ratio. The present effect of the pitch ratio (PR = 4, 6 and 12) and number of the perforated holes ($N=4$, 6 and 8 holes) of the PCRs on the heat transfer enhancement are correlated in term of the Nusselt number for different Reynolds numbers as

$$Nu = 1.258\mathrm{Re}^{0.606}\mathrm{PR}^{-0.39}N^{-0.32}\mathrm{Pr}^{0.4} \quad (13)$$

4.2. Friction factor results

Similarly to the verification for the Nusselt number mentioned in Section 4.1, verification of the present plain tube for the friction factor is made by comparison of the present data with those from the previous studies, under similar test condition. The verification result is shown in Fig. 6. As seen, the friction factor values of the present plain tube are in good agreement with previous correlations of Blasius [15] with the deviations within ±5.9% for $4000 \leq Re \leq 20{,}000$, $Pr \approx 0.7$ and $L/D = 24.2$. The correlation of friction factor for the present plain tube can be drawn as

$$f = 0.458\mathrm{Re}^{-0.284} \quad (14)$$

4.2.1. Effect of pitch ratio

Effect of the pitch ratio (PR = 4, 6 and 12) of the PCR on the friction factor under isothermal turbulent flow condition is displayed in Fig. 7 (a–b). The results obtained for plain tube are also plotted for comparison. Evidently, friction factor noticeably increases with decreasing pitch ratio. In the other word, a decrease of the distance between each pair of the turbulators causes an increase in friction factor. This is due to the simple fact that the smaller distance between each pair of the PCRs, the more numbers of PCRs available in the tube, thus the more blockage against the flowing stream. In addition, it can be observed that the friction factors in the tube with the PCR are significantly lower than those in the tube with the typical CR, which are around 72.2% for PR = 4, 68.1% for PR = 6, and 72.5% for PR = 12. This indicates that the presence of the perforated hole in the conical-ring possesses high potential for reduction of friction in the tube.

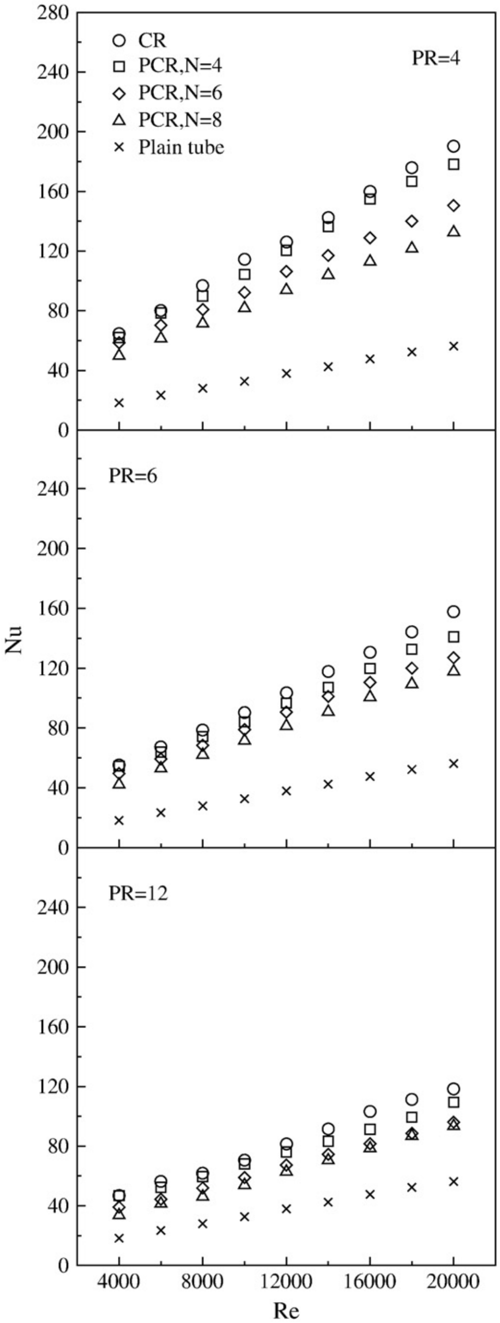

Fig. 5. Effect of number of perforated hole (N) on heat transfer enhancement.

4.2.2. Effect of number of hole in PCR

Effect of the number of perforated hole ($N=4$, 6 and 8 holes) of the PCRs on the friction factor is presented in Fig. 8. Obviously, friction factor decreases with the increase of the number of perforated hole due to the reduction of turbulent fluctuation or eddy motion and the appearance of reverse flow between the each pair of the turbulators. The friction factors of using the PCRs with $N=4$, 6 and 8 holes, are respectively around 57.2%, 73.6% and 82%, lower that of using the typical CR. The correlation of friction factor for the tube with PCRs in the present work can be expressed as

$$f = 985.48\mathrm{Re}^{-0.368}\mathrm{PR}^{-0.747}N^{-1.253} \tag{15}$$

Comparison between the present data (Nu and f) with those calculated by the present correlations for the Nusselt number and friction factor are portrayed in Figs. 9 and 10. In the figures, the

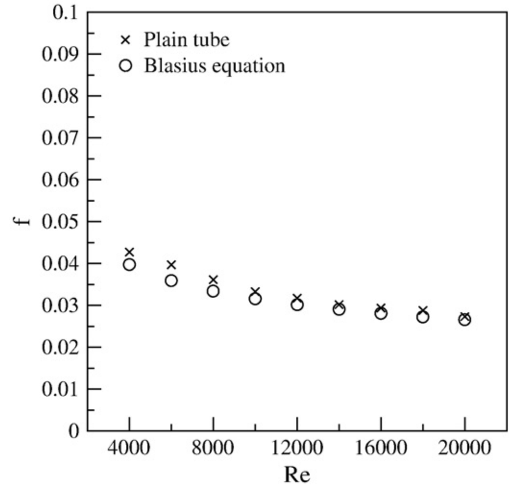

Fig. 6. Friction factor verification for plain tube.

deviations of the measured data fall within ±10% and ±12% from the present correlations of the Nusselt number and the friction factor, respectively.

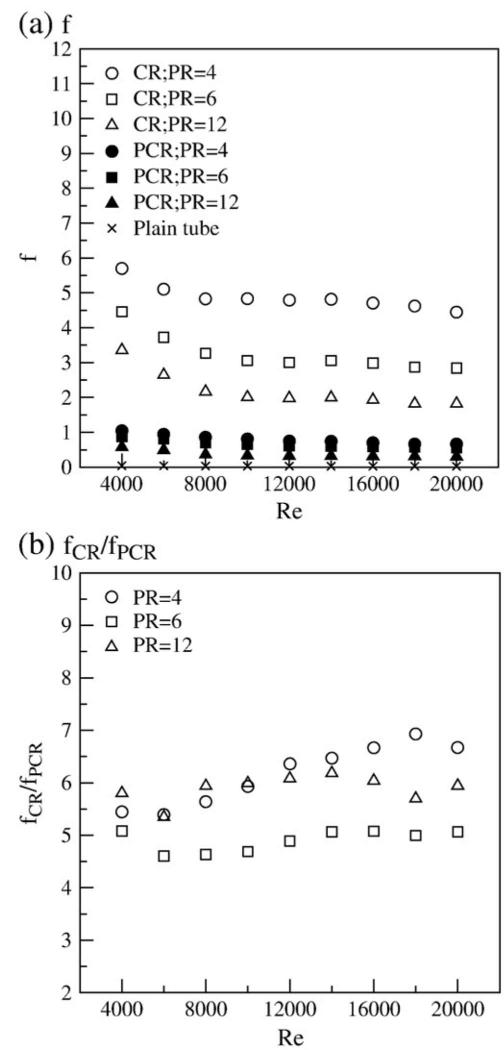

Fig. 7. Effect of pitch ratio (PR) on friction factor for PCR with $N=4$ holes: (a) f and (b) f_{CR}/f_{PCR}.

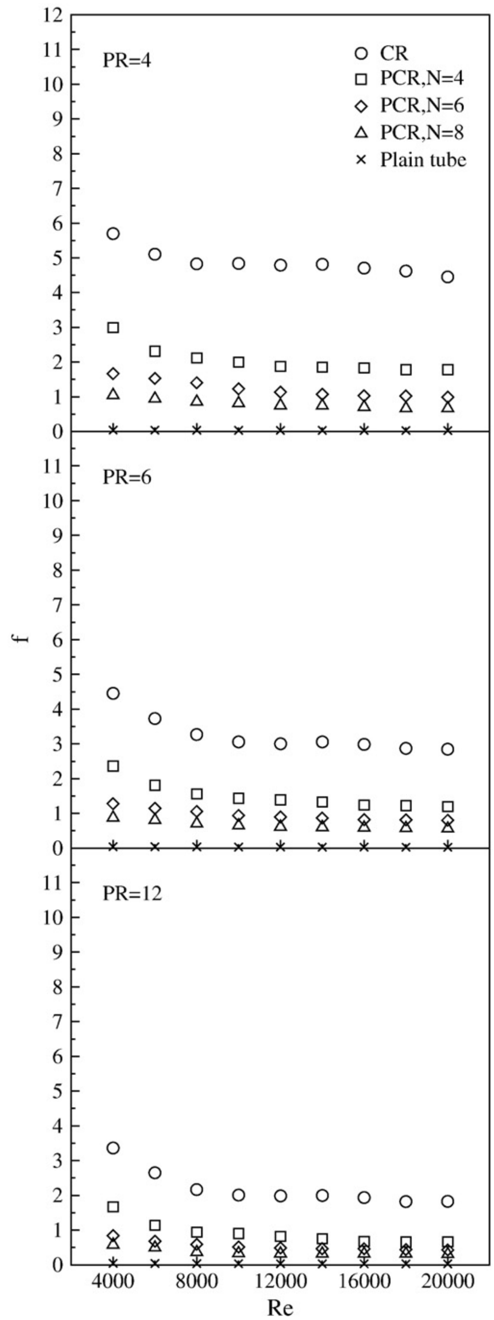

Fig. 8. Effect of number of perforated hole (N) on friction factor.

4.3. Thermal performance factor results

According to the results shown above, the PCR turbulator offers heat transfer rate enhancement in accompany with the increase of friction factor. The increase of friction causes a rise of pumping energy. Therefore, the actual effectiveness of the turbulator depends upon the weight of the increase in heat transfer and the increase in friction which can be determined from performance evaluation. Generally, the performance evaluation the turbulator (enhancement device) is made using the data of the plain tube as reference and usually considered at the same pumping power, since this is relevant to the operation cost. For constant pumping power

$$(\dot{V}\Delta P)_p = (\dot{V}\Delta P)_t \quad (16)$$

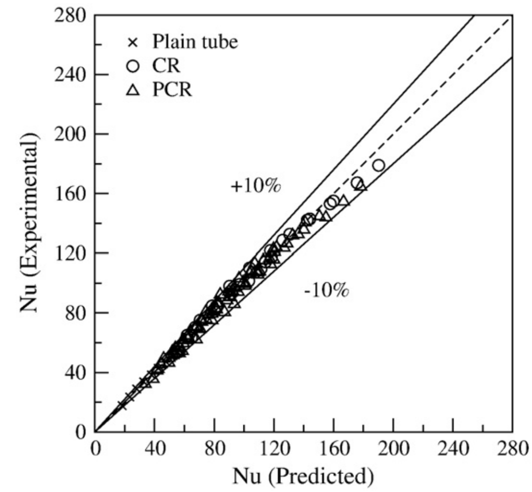

Fig. 9. Comparison of experimental data with those calculated from the correlation for Nusselt number.

The relationship between friction and Reynolds number can be written as

$$(f\mathrm{Re}^3)_p = (f\mathrm{Re}^3)_t \quad (17)$$

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 (PCR) to the plain tube that can be found below.

$$\eta = \left(Nu_t / Nu_p\right) / \left(f_t / f_p\right)^{1/3} \quad (18)$$

The present results from Eq. (18) correlated with the thermal performance factor can be drawn as

$$\eta = 1.596\mathrm{Re}^{-0.067}\mathrm{PR}^{-0.142}N^{-0.095} \quad (19)$$

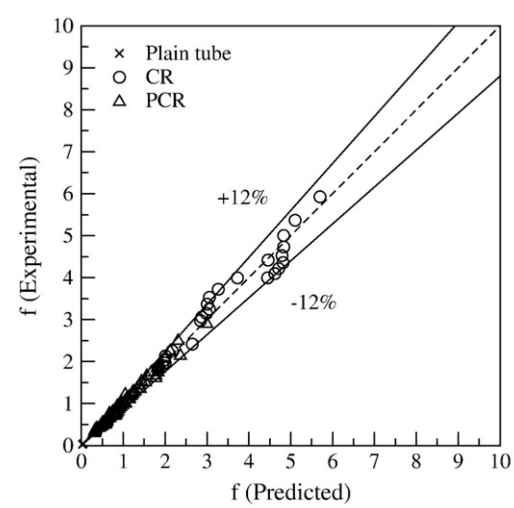

Fig. 10. Comparison of experimental data with those calculated from the correlation for friction factor.

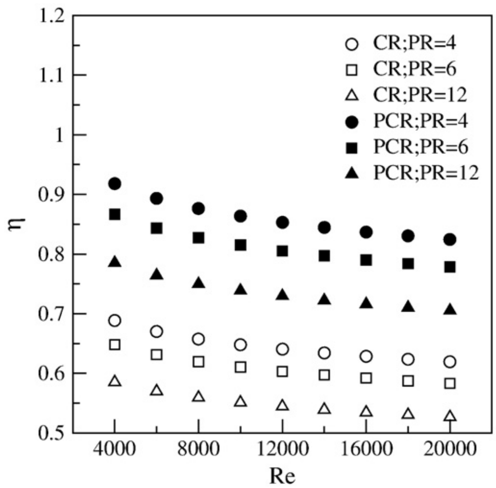

Fig. 11. Effect of pitch ratio (PR) on thermal performance factor.

Effects of the pitch ratio (PR) and number of perforated hole (N) of the turbulators on the thermal performance factor (from Eq. (19)) of the tube fitted with the turbulators are presented in Figs. 11 and 12. It is found that the thermal performance factor decreases with increasing Reynolds number and it is also observed that the turbulator with the smallest pitch ratio (PR) provide the highest thermal performance factor for both of the PCR and CR turbulators. The maximum thermal performance factors of tube fitted with PCRs at PR = 4, 6, and 12, are found to be 0.92, 0.87, and 0.79, respectively. Interestingly, the performance factors of the tube fitted with the CRs are lower than those of the tube with PCRs around 19.8–25.5%. This indicates that at the same pumping power the PCRs can enhance heat transfer more efficient than the CRs and this confirms the greater benefit of the use of PCRs as energy saving device over the CRs. Additionally, the PCRs with the larger number of the perforated hole (N) offer the higher thermal performance factor because they generate lower pressure drop. Fig. 13 presents the comparison of the thermal performance factor between experimental data ($Nu_t/Nu_p/((f_t/f_p)^{1/3})$) and those calculated from the Eq. (19). It is found that the majority of the measured data falls within ±6% for the present correlation.

5. Conclusions

Heat transfer and friction factor characteristics in the tube with the perforated conical-rings (PCRs) are investigated for Reynolds numbers ranging from 4000 to 20,000. The effects of the pitch ratio (PR) and number of perforated hole (N) on the heat transfer enhancement in a tube are also considered. The concluding remarks can be described as follows:

(1) At the similar test conditions, the PCRs offers lower heat transfer enhancement than the CRs. However, they generate friction factor only around 25% of that produced by the PCRs. Consequently, the thermal performance factor of all PCRs arrangements is higher than those of the CRs over the range studied.
(2) The heat transfer rate and friction factor of PCRs increase with decreasing pitch ratio (PR) and decreasing number of perforated hole (N). However, the thermal performance factor increases with increasing number of perforated hole and decreasing pitch ratio.

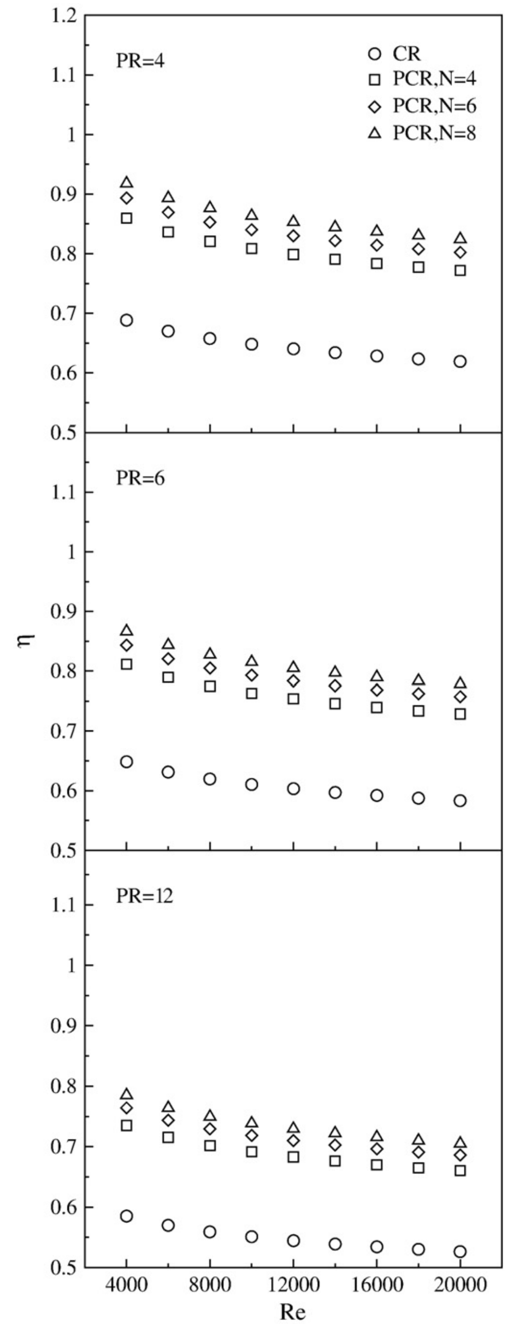

Fig. 12. Effect of number of perforated hole (N) on thermal performance factor.

(3) The mean heat transfer rates obtained from using the PCR with PR = 4, 6, and 12 are found to be respectively, 185%, 140%, and 86%, over the plain tube. Over the range investigated, the maximum thermal performance factor of around 0.92 is found at PR = 4 and N = 8 holes with the Reynolds number of 4000.

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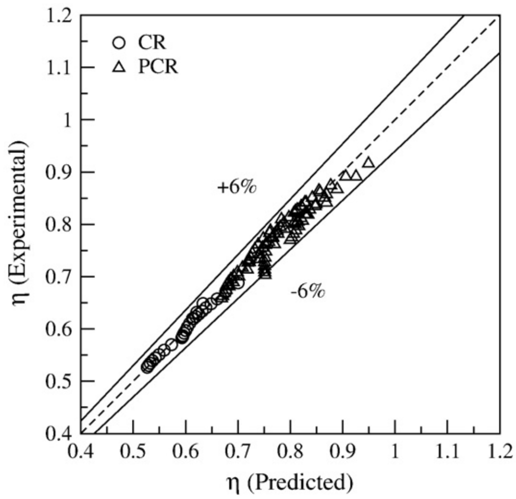

Fig. 13. Comparison of experimental data with those calculated from the correlation for thermal performance factor.

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