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Research Article

EXPERIMENTAL STUDY OF THERMAL PERFORMANCE IN A TUBULAR HEAT EXCHANGER USING INCLINED PERFORATED VORTEX RINGS

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ABSTRACT:
This paper presents an experimental investigation on thermal and friction characteristics in turbulent airflow through a tubular heat exchanger inserted with 45° inclined perforated vortex rings (P-VRs). For the present work, the P-VR elements employed to create the streamwise vortices were mounted on the tube wall using two small straight wires to link the P-VR elements together. Influences of three relative ring-height or blockage ratios (e/D = BR = 0.20, 0.15 and 0.10) and three relative ring-pitches or pitch ratios (P/D = PR =2.0, 1.0 and 0.5) for a fixed angle of attack of 45° on the Nusselt number (Nu), friction factor (f) and thermal enhancement factor (TEF) for turbulent tube flows, Reynolds number (Re) ranging from 4200 to 26,000 were explored. The measured result has shown that the rate of heat transfer and friction loss rise considerably with increasing BR but declining PR. The highest values of Nu and f around 3.37–4.38 and 14.07–39.42 times above the smooth tube, respectively are at BR = 0.2 and PR = 0.5. The highest TEF is seen to be about 1.55 at PR = 1.0, BR = 0.10. Further, the Nu and f correlations were established by fitting the experimental data having the deviations from the measured data by ±7% each.

Keywords: *Heat exchanger, Vortex ring, Friction loss, Thermal performance*

1. INTRODUCTION

Heat transfer augmentation techniques have been developed and widely utilized in several industries for example, locomotives, air conditioners, aerospace, heat exchangers, etc. As a simple and effective passive heat transfer augmentation method, various vortex generators (VG) have been used to alter the flow direction of fluid, and to reduce the thickness of the boundary layer, leading to the increase in the convective heat transfer coefficient. Nevertheless, inserting the VG device into a tube results in the rise of flow resistance in the heat exchanger tube. Thus, lots of researchers have engaged themselves in researches on this area and various numerical and experimental investigations on the influences of using VGs on the heat transfer rate and pressure loss in a tubular heat exchanger were carried out [1-8].

Man et al. [9] studied comparatively the effect of flow resistance and the heat transfer in turbulent flow between the alternation of clockwise and counter-clockwise twisted tape (ACCT tape) and the typical twisted tape (TT tape). They showed that the modified twisted tapes provided the heat transfer better than the typical twisted tapes. Chiu and Jang

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[10] numerically investigated the heat transfer and flow friction characteristics in a round tube with various tube inserts, such as streamwise strip inserts (with and without holes) and twisted-tape inserts. Fan et al. [11] studied numerically the effect of the louvered strip insert on the thermal and flow friction characteristics in a round tube. Skullong et al. [12] examined experimentally the flow friction and heat transfer behaviors in a round tube with staggered-winglet perforated-tapes. The heat transfer rate and friction loss in a round tube with curved delta wing vortex generators were reported by Deshmukh et al. [13]. Tamna et al. [14] explored the application of the double twisted-tapes with V-ribs on the edges to enhance the heat transfer in a tubular heat exchanger and showed that the highest TEF at about 1.4 was achieved.

Ibrahim et al. [15] performed a CFD study on the friction and heat transfer behaviors in a round tube with conical ring turbulators. Skullong et al. [16] conducted an experiment to study the effect of quadruple perforated-delta-winglet pairs on thermal and fluid flow characteristics at two blockage ratio (BR) and pitch ratio (PR). Promvonge et al. [17] studied the effect of the inclined vortex rings on the thermal performance friction loss, and the heat transfer rate in a circular tube. Chingtuaythong et al. [18] studied experimentally the turbulent convection heat transfer and flow resistant in a heat exchanger tube inserted with V-shaped rings.

Previous research works have showed that the insertion of twisted tape/wire coil into a round tube provided poor thermal performance (TEF < 1) owing to lower vortex strength of the flow resulting in a slight increase in heat transfer with considerably higher pressure loss while the application of wings/winglets/vortex rings yielded a realizable thermal performance with relatively low-pressure loss penalty. Thus, in the present work another type of the insert device, the 45° inclined perforated vortex ring (P-VR) is introduced to rectify this matter. The current vortex-flow device has been developed by using the combined merits of baffle and winglet vortex generators. This implies that the VG device mentioned above will provide a much higher heat transfer like the baffle and a vortex/swirl flow as the winglet. Hence, a novel 45° inclined P-VR is offered with the expectation to give higher thermal performance than the typical twisted tape/wire coil owing to the vortex-induced impingement/reattachment effect as reported in Refs. [7, 19]. The experimental result using air as the test fluid for the P-VR inserted tube is presented for turbulent flows in the Reynolds number range of 4200 to 26,000 in the current study.

A new type of the vortex ring is mounted periodically in a round tube having a uniform heat flux. This is much fruitful in designing and upgrading the heat exchanger tubes. Hence, the main purpose of this work is divided into two aspects: (i) to investigate the heat transfer augmentation and the friction loss in a heat exchanger tube inserted with P-VRs at different size/arrangement in the form of BR and PR values, (ii) to give reliable correlations based on the experimental data of the friction factor (f) and Nusselt number (Nu).

2. EXPERIMENTATION

2.1 Technical details of inclined P-VRs

The detail and layout of the 45° inclined P-VR elements is presented in Fig. 1. An aluminum strip with thickness (t) of 3 mm was used to form a perforated vortex ring whose punched hole diameter (d) was set to constant at 2 mm (fixed). A 3-mm aluminum sheet was formed to be an elliptical ring and then punched to become a P-VR element as seen in Fig. 1. Each of the inclined P-VRs was fastened to each other by inserting the two straight wires (3-mm diameter) into the holes drilled on two sides of P-VRs (see Fig. 1). The P-VRs included three ring height ratios (e/D=BR= 0.1, 0.15 and 0.2) and pitch ratios (P/D=PR= 2.0, 1.0 and 0.5) were placed by the wall-attached arrangement.

2.2 Experimental setup

The schematic representation of the experimental setup used in the present work is shown in Fig. 2. In the test apparatus, the 2-mm thick (t) copper tube having a 50.8-mm inner-diameter (D) was totally 3000-mm long, included the 1200-mm tested tube length (L) and the 200-mm exit-section whereas the inlet section of 1600 mm was adopted to have the flow to be fully developed as it entered the tested tube. A Nichrome heating wire of 21-gauge and resistance of 46 Ohms was wound all over the entire length of the test tube. The heater wire was covered with ceramic beads to act as electrical insulation to prevent the short circuit of electric current. The Nichrome wire terminals were connected to a variac transformer (AC power supply) to achieve a uniform heat-flux tube wall (q'' = 600 W/m^2). The outermost surface of the test tube was wrapped with ceramic insulations to reduce the convection loss to the surrounding air. In the test runs, air was flowed through the tube to obtain Re in a range of 4200 to 26,000.

A 2.3-kW blower was employed to force the air at 25 °C (room temperature) into the heat exchanger tube. Air flowed through an orifice meter mounted between the blower and the settling tank to measure the volume airflow rate and an inclined manometer was employed to measure the pressure drops across the orifice flow-meter. The volume rates of airflow were varied by controlling the speed of blower-motor via an inverter. Two RTD (Resistance Temperature Detectors) were employed to measure the air inlet and exit temperatures whereas the wall temperatures were monitored by using sixteen T-type thermocouples placed equally along the top wall of the tube. Whenever, temperatures of the exit air and tube walls remained unchanged for the interval of 20 minutes implying that it was in a steady state, all the temperatures were collected via a Fluke 2680A data acquisition system. The pressure loss of the tested section was measured using a digital manometer (Dwyer Mark III). According to the methodology introduced by references [20], the measurement uncertainties of pressure, temperature and velocity were, respectively, obtained by ±3.2, ±0.3 and ±4.6%. The highest values of uncertainties in other instruments varied by ±1.2% for the flowmeter and ±1.1% for the manometer while the dimensionless parameters had the uncertainties around 3.5% for Nu, 6.8% for *f* and 3.2% for Re. More details of the experimental uncertainty calculation can be seen in Refs. [21].

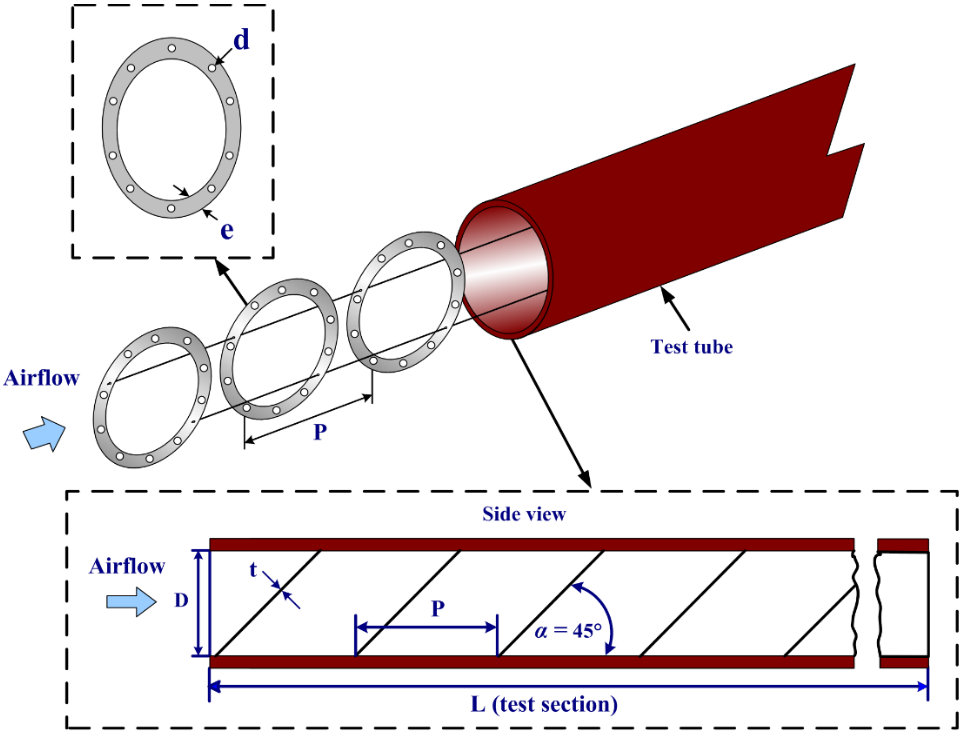


Fig. 1. Test tube inserted with 45° inclined P-VR.

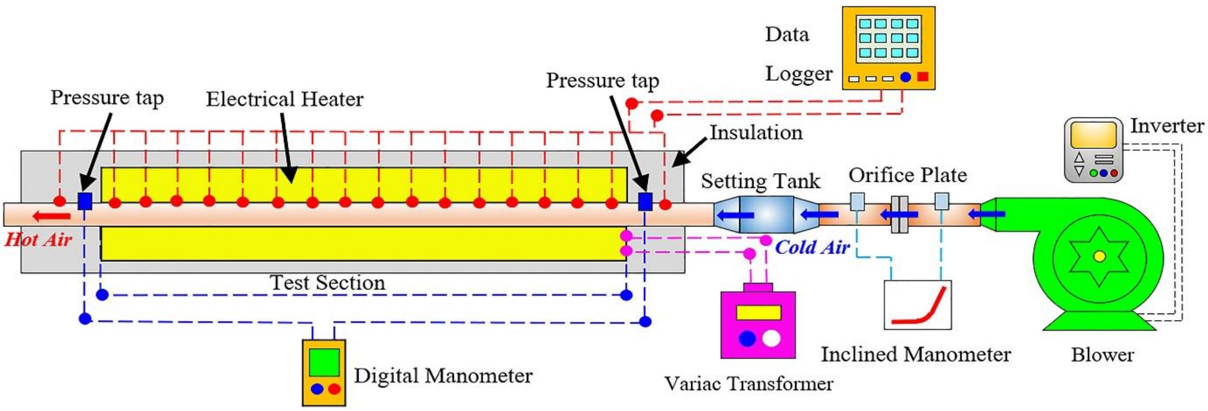


Fig. 2. Schematic representation of the experimental system.

3. THEORETICAL ANALYSIS

The current work is carried out to investigate the friction factor (*f*), Nusselt number (Nu) and thermal enhancement factor (TEF) at various Reynolds numbers (Re) in a heated tube fitted with inclined P-VRs.

The rate of airflow displayed in terms of Reynolds numbers (Re) is given by

$$\mathrm{Re} = UD/\nu \tag{1}$$

The pressure drop (ΔP) of the tested tube is employed for estimating the friction factor (*f*) as denoted via

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

in which U is defined as the average velocity.

In this work, air enters steadily the tested section and the heat transfer from the tube to air is prescribed by

$$Q_{\mathrm{a}} = \dot{m}C_{\mathrm{p,a}}(T_{\mathrm{o}} - T_{\mathrm{i}}) = VI - Q_{\mathrm{loss}}$$

where Q_{loss} is the tube heat loss by convection to the surroundings. With regard to a thermal resistance network for the cylindrical layer concept [22], the Q_{loss} can be written as:

$$Q_{\mathrm{loss}} = (T_{\mathrm{w}} - T_{\infty}) \Big/ \frac{\ln(r_{\mathrm{o}}/r_{\mathrm{i}})}{2\pi Lk} + \frac{1}{2\pi L r_{\mathrm{o}} h_{\mathrm{o}}}$$

where h_0,T_w, T_∞ are respectively, the free convection coefficient of air from the outermost tube surface and the surrounding, the local temperature of the tube wall, and the surrounding temperature, while k, r_i, r_o, are respectively, the insulation thermal conductivity, the inner and outer radii of the tube.

At thermal equilibrium condition, Q_{loss} is seen beneath 4% of the electrical input power (*VI*). Owing to air entering steadily the test section, the rate of heat transfer is assumed to be equal to the heat from convection and given by

$$Q_{\mathrm{a}} = Q_{\mathrm{conv}} \tag{3}$$

The convective heat transfer inside the tube is estimated via

$$Q_{\mathrm{conv}} = hA(\tilde{T}_{\mathrm{w}} - T_{\mathrm{b}}) \tag{4}$$

in which

$$T_{\mathrm{b}} = (T_{\mathrm{o}} + T_{\mathrm{i}})/2 \tag{5}$$

and

$$\tilde{T}_{\mathrm{w}} = \sum T_{\mathrm{w}}/16 \tag{6}$$

The mean wall temperature, $\tilde{T}_w$, is calculated by 16 local wall temperatures lined equally along the test tube.
The mean Nusselt number (Nu) can be evaluated using the mean heat transfer coefficient (h) as rewritten by

$$h = \dot{m}C_{\mathrm{p,a}}(T_{\mathrm{o}} - T_{\mathrm{i}})/A(\tilde{T}_{\mathrm{w}} - T_{\mathrm{b}}) \tag{7}$$

where, A is the heat transfer surface area inside the tube, and determined by

$$A = \pi DL \tag{8}$$

where D is the inner diameter of tube.

Thus, the average Nu is from the following relations:

$$\mathrm{Nu} = hD/k \tag{9}$$

Thermo-physical property values of air can be obtained at the bulk air temperature (T_b) given by Eq. (5).
Thermal enhancement factor (TEF) is given by

$$\mathrm{TEF} = \left(\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right)\left(\frac{f}{f_0}\right)^{-1/3} \tag{10}$$

where Nu_0 and f_0 are defined for the plain tube while the others stand for the inserted tube.

4. Results and Discussion

4.1 Verification of smooth tube

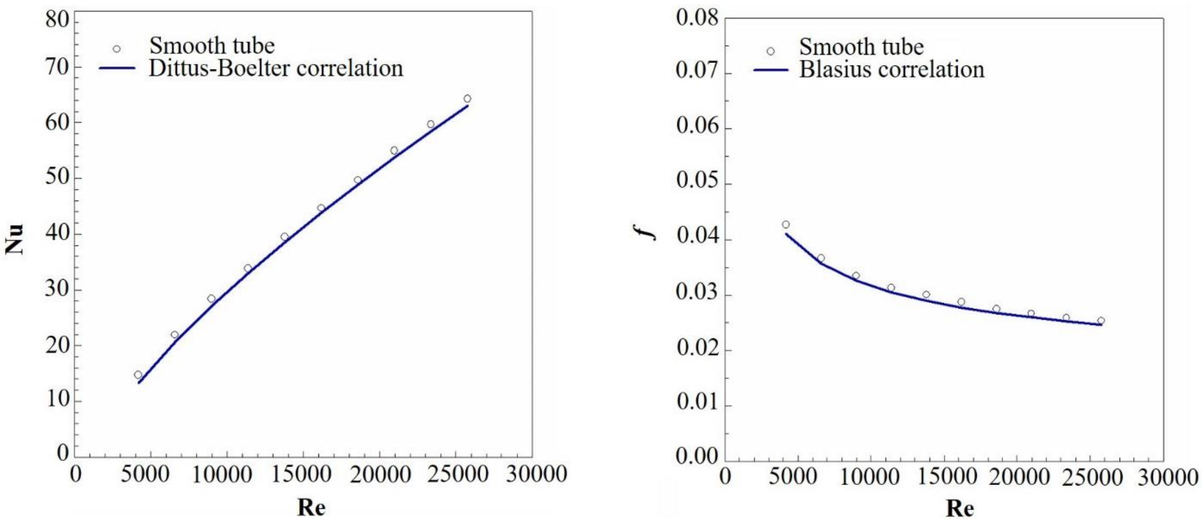
In comparison, Nu and f from the present smoot tube and those from the correlations in previous work [22], Dittus-Boelter correlation (Eq. 11) for Nu and Blasius correlation (Eq. 12) for f, are plotted against each other as presented in Fig. 3(a) and (b), respectively. In the figure, the measured results are in good agreement with those from the published correlations. The average deviations of both the results are within ±4.9% and 5.6% for Nu and f, respectively.

Correlation of Dittus and Boelter for heating

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

Correlation of Blasius:

$$f = 0.316\mathrm{Re}^{-0.25} \tag{12}$$



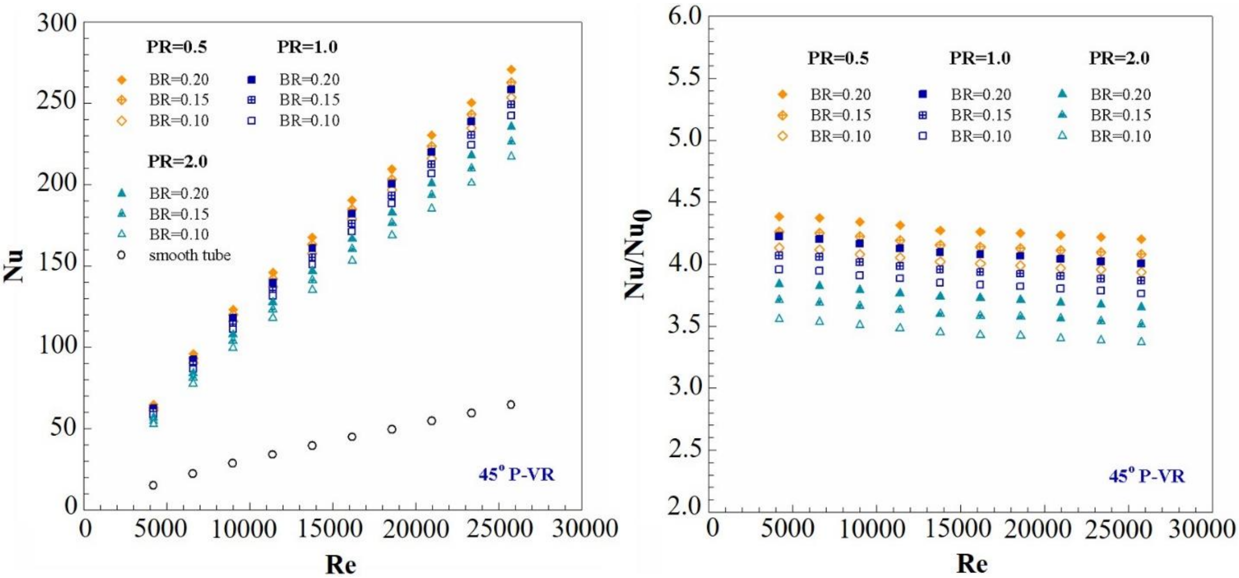
(a) (b)

Fig. 3. Comparison of measured (a) Nu and (b) f with published data for the smooth tube.

4.2 Perforated vortex ring effect on Nusselt number

The profiles of Nu and the augmented Nu or Nu/Nu_0 along with Re for the inclined P-VRs at varied BR and PR are, respectively, depicted in Fig. 4(a) and (b). From Fig. 4(a), the Nu values of all inserted tubes increase considerably with rising Re. Obviously, Nu of the inclined P-VR is considerably higher than the smooth tube alone and shows the uptrend with rising BR but with the decline of PR. This can be explained from the fact that the appearance of inclined P-VRs assists to produce the vortex flow along the tube, aside from increasing a high turbulence level. The use of the P-VR at large BR gives rise to higher flow blockage in the tube, hence, resulting in stronger strength of vortex flow than that at small BR. The increase in heat transfer using the inclined P-VR is around 236–338% compared to the smooth tube and for PR=0.5 and BR= 0.2, gives the highest Nu at about 338% above the smooth tube.

The variation of Nu/Nu_0 for the inclined P-VR insert with Re is depicted in Fig. 4(b). In the figure, the general tendency of Nu/Nu_0 for the inclined P-VR is found to decrease slightly for increasing Re. Nu/Nu_0 shows the uptrend with rising BR but with reducing PR. At BR = 0.2, 0.15 and 0.1, the average values of Nu/Nu_0 are respectively, about 4.29, 4.16 and 4.03; 4.1, 3.96 and 3.86; and 3.74, 3.61 and 3.45 times for PR = 0.5, 1.0 and 2.0. This implies that using the larger BR and smaller PR causes a significantly strong flow-circulation aside from more intensity of turbulence and interruption of the boundary layer development, leading to the rise in heat transfer. Nonetheless, at this condition, it comes together with substantially higher friction loss, as being seen in the next section.



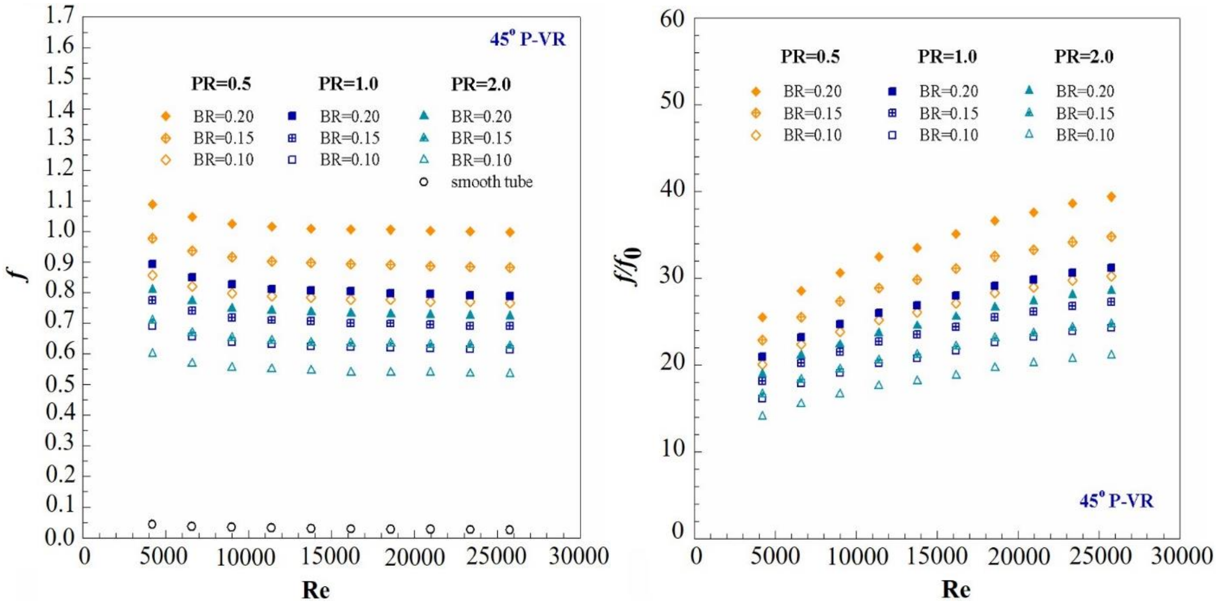
(a) (b)

Fig. 4. (a) Nu and (b) Nu/Nu_0 versus Re for inclined P-VRs.

4.3 Perforated vortex ring effect on friction factor

The plots of friction factor (f) and friction factor ratio (f/f_0) against Re for various BR and PR values are depicted in Figs. 5(a) and (b), respectively. It can be noted in Fig. 5(a) that the P-VR yields much greater f value above the plain tube and its trend displays a slight decrease with rising Re. A close examination reveals that f tends to rise with the increment in BR while shows the reversing trend with increasing PR. This is due to higher flow obstruction, the fluid-viscous dissipation from the increased surface area and the drag by the longitudinal vortex flow, leading to substantially higher flow friction and form drag, especially for large BR and small PR. In comparison with the smooth tube, f of the inclined P-VR is increased around 14.07–39.42 times.

As illustrated in Fig. 5(b), it is seen that f/f_0 has the up tendency with rising Re. The employ of the inclined P-VRs gives an extreme increase of f/f_0 with rising Re. It is obvious that f/f_0 shows the uptrend with increasing BR while exhibits the reversing trend with rising PR. The average f/f_0 values with BR = 0.2, 0.15 and 0.1 are some 33.83, 30.05 and 26.22; 27.05, 23.63 and 21; and 24.65, 21.39 and 18.23 times at PR = 0.5, 1.0 and 2.0, respectively.



(a) (b)

Fig. 5. Variation of (a) f and (b) f/f_0 versus Re for inclined P-VRs.

4.4 Thermal enhancement factor

The potentiality of the inclined P-VRs in practical can be assessed via an indicator in terms of thermal enhancement factor (TEF) as denoted in Eq. (10). This result is directly pertinent to the trade-off of the enhanced Nu with the increased f penalty. TEF plotted against Re at various PR and BR is shown in Fig. 6 below. As seen, all values of TEF are above unity, implying the merit of the inclined P-VR over the smooth tube alone. TEF shows the down tendency with the rise in Re for the P-VR insert while the maximum around 1.55 is seen at BR = 0.1, PR = 1.0. The maximum TEF is found at the lowest BR (BR=0.1), due to lowest f. Also, it is observed that the highest TEF values are in the range of 1.44 to 1.55 for different BR and PR values at the lowest Re. Hence, the best choice of this P-VR roughness is at BR = 0.1, PR = 1.0 to achieve the superior thermal performance.

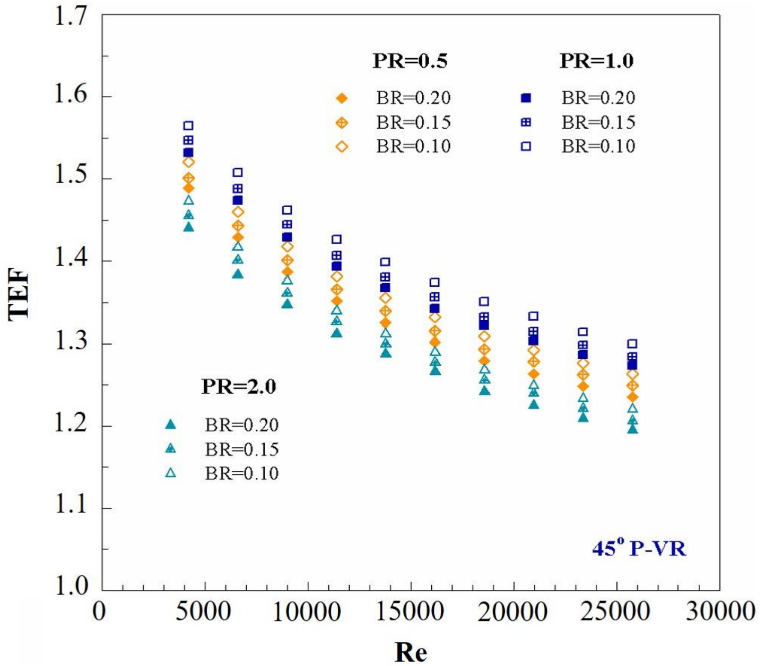


Fig. 6. Variation of TEF with Re for various inclined P-VRs.

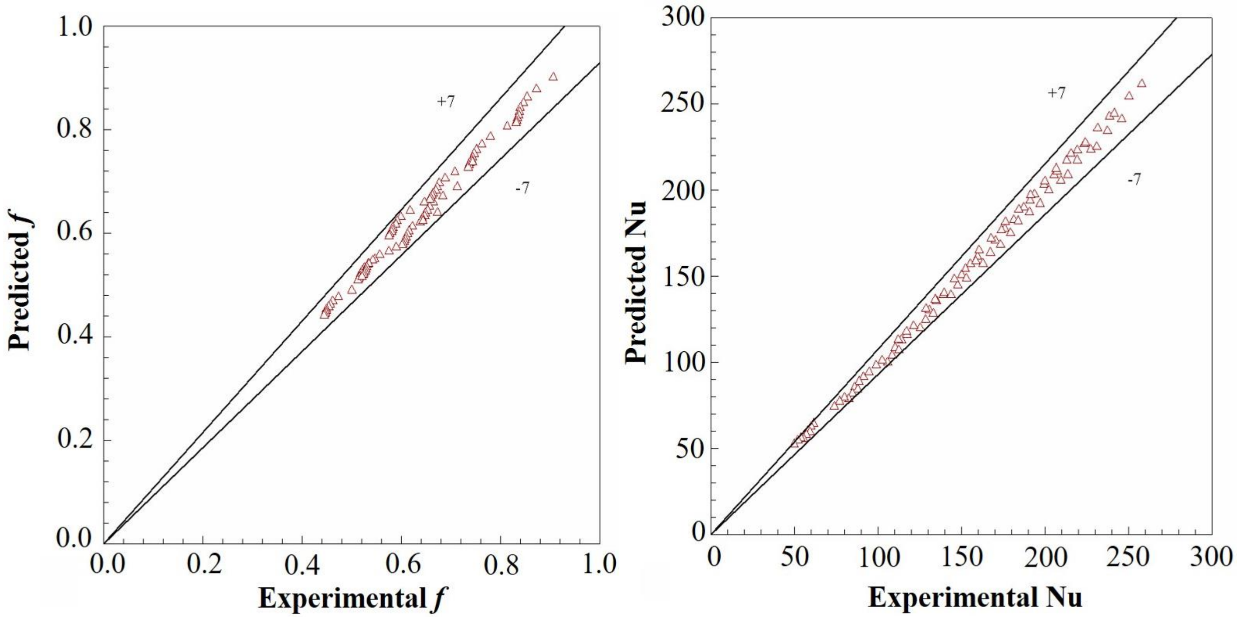
5. Nu AND f CORRELATIONS

The f and Nu empirical correlations for using the 45° inclined P-VRs are determined from the measured data and they are correlated as given in equations (13) and (14). The correlation of Nu is displayed as a function of Re, PR, BR and Prandtl number (Pr) but f is independent of Pr. The f and Nu correlations are obtained as follows:

$$f = 3.808\mathrm{Re}^{-0.057}\mathrm{BR}^{0.386}\mathrm{PR}^{-0.494} \tag{13}$$

$$\mathrm{Nu} = 0.16\mathrm{Re}^{0.772}\mathrm{Pr}^{0.4}\mathrm{BR}^{0.097}\mathrm{PR}^{-0.202} \tag{14}$$

The reliability of the above correlations is verified by plotting the values of f and Nu from their correlations with the measured data as shown in Fig. 7(a) and (b) respectively. It is apparent that the discrepancies between the predicted and the measured data of Nu and f are by ±7% each.



(a) (b)

Fig. 7. Comparisons of the predicted (a) f and (b) Nu with the experimental data.

6. CONCLUSIONS

An experimental study of thermal behaviors in a tubular heat exchanger fitted with a novel designed vortex ring in the form of inclined perforated vortex ring (P-VR) under turbulent flow, Re = 4200–26,000, using air as the working fluid has been conducted. The effects of BR, PR and Re on thermal behaviors of the inclined P-VR insert in the tube are investigated. Key findings in the current study are as follows:

- The use of inclined P-VR has a significant effect to thermal and flow behaviors in the tube resulting in enhancing the heat transfer as well as the friction loss.
- All the inclined P-VRs provide considerably higher heat transfer than the smooth tube alone. The increase in Nu/Nu_0 is ranging from 3.37 to 4.38 times while that in the f/f_0 is around 14.07 to 39.42, depending on PR, BR and Re.
- For performance comparison, the highest TEF around 1.55 is seen at BR = 0.1 and PR = 1.0, pointing that in practical, the P-VR is regarded as a superior device for upgrading a tubular heat exchanger.
- The Nu and f correlations are determined and their values are found to agree favorably with the experimental data. Their deviations are within ±7% each.

Nomenclature

A	[m^2]	heat transfer surface area
BR	[=e/H]	blockage ratio of ring
$C_{p,a}$	[J/kg K]	specific heat of air
D	[m]	tube diameter
d	[m]	punched hole diameter on ring
e	[m]	height of ring
f	[-]	friction factor
k	[W/m K]	fluid thermal conductivity
L	[m]	length of test section
$\dot{m}$	[kg/s]	mass rate of flow
Nu	[-]	Nusselt number
PR	[=P/H]	ring pitch ratio
Pr	[-]	Prandtl number
P	[m]	pitch length between rings
Q	[W]	heat transfer rate
Pr	[-]	Prandtl number
Re	[-]	Reynolds number
T_i	[^oC]	inlet air temperature
T_o	[^oC]	outlet air temperature
T_b	[^oC]	bulk air temperature
TEF	[-]	thermal enhancement factor

Greek symbols

α	[degree]	inclination angle
ν	[N s/m^2]	fluid kinematic viscosity

Subscripts

0	plain/smooth tube
a	air
w	wall

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