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Performance assessment in a heat exchanger tube fitted with perforated-rectangular-winglet tape inserts

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Abstract

An experimental work has been conducted to examine the turbulent flow and heat transfer characteristics in a constant heat-fluxed round tube with perforated-rectangular-winglet tape (PRWT) inserts. The fluid flow and thermal characteristics are presented for Reynolds numbers (Re) ranging from 4100 to 25,400. In the current study, a straight tape with 45° PRWT mounted repeatedly on both tape sides is inserted into the test tube to generate two pair of longitudinal vortex flow throughout. The influences of punched hole or pore diameter on the winglet (d = 1, 1.5, 2 and 3 mm) having a single winglet width ratio (b/D=BR= 0.24) and a winglet pitch ratio (P/D=PR= 3.0) on heat transfer and friction loss behaviors in the tube are investigated. The experimental results show that the PRWT provides higher heat transfer rate and friction loss than the smooth tube. The reduction of pore diameter, d results in the increase in Nusselt number (Nu) and friction factor (f) values. The inserted tube with d = 2 mm yields the highest thermal performance of about 1.46. Comparing thermal performance at a similar operating condition, the PRWT with d= 2 mm performs around 1%, 1.8% and 3% higher than the one with d = 3, 1.5 and 1 mm, respectively.

Keywords: Winglet; Heat exchanger; Turbulent flow; Vortex generator; Thermal performance

1. Introduction

Heat transfer enhancement (HTE) technology has been developed and widely applied to efficient utilization of heat exchanger systems. Swirl flow devices, as one of the passive HTE technology [1], have been extensively studied due to ease of installation, manufacturing and maintenance. Insertion of turbulator/swirl generator device have been widely used because the reduction in overall thermal resistance can lead to a smaller size and saving energy of heat exchanger systems. Many investigations have been carried out to study the effect of several swirl flow devices on HTE in heat transfer systems both tubular and flat-plate duct heat exchanger.

For tubular heat exchangers, the wire coils and twisted tapes have been widely used for enhancing the rate of heat transfer. The conjugate heat transfer and thermal stress in a uniform wall heat-fluxed tube with coiled-wire inserts was numerically investigated by Ozceyhan [2]. Eiamsa-ard and Promvonge [3] reported the effect of serrated twisted tape (STT) insert on fluid flow and heat transfer behaviors in a constant heat-fluxed tube by considering two STT geometry parameters (serration width and depth ratios). They found that the thermal performance factor of all STT is better than that of the typical twisted tape in a range of 2–12%. A comparison of the thermal and hydraulic performances of twisted tape or wire coil inserts was introduced by Wang and Sunden [4]. They found that the coiled wire performs effectively in enhancing heat transfer in a higher turbulent flow region whereas the twisted tape yields a poorer overall efficiency. Promvonge [5] investigated the effects of the wire coils in conjunction with twisted tapes on heat transfer and friction characteristics in a uniform heat-fluxed tube and found that for using the combined devices, the increase in heat transfer rate and performance are about 200–350% of using a single enhancement device.

The thermal performance can be enhanced by the use of winglets and baffles for heat exchanger ducts or channels. Zhou and Ye [6] presented experimental data of thermal and flow characteristics for rectangular-channel flow having curved trapezoidal, rectangular, trapezoidal and delta winglets. Skullong et al. [7] investigated the fluid flow and heat transfer coefficient in a solar air heater fitted with combined wavy-groove and perforated-delta wing vortex generators. Kwankaomeng and Promvonge [8], and Promvonge et al. [9] studied numerically the periodic flows through 30° and 45° angled baffles mounted in a square channel, respectively. In the literature review above, for thermal performance improvement in tubular heat exchangers, the wire coils and twisted tapes have been widely applied while in duct/channel heat exchangers, winglets/baffles have been extensively employed. The optimum thermal performance for a channel with vortex generator (VG) in the form of winglet-type is much higher than that for a tube inserted with swirl flow devices. However, the VG inserts with different punched pore diameters on rectangular winglets have not been investigated in the past, especially for a tubular heat exchanger. Therefore, considering the advantages of VG devices, a newly designed VG device in the form of perforated-rectangular-winglet tape (PRWT) is introduced in the current work.

2. Experimental facility

Fig. 1 presents a schematic of experimental set-up used in the current work. In the figure, a blower used for supplying air to the test system was connected by a settling tank with a clam section to become a fully developed flow. The volumetric air flow rate was varied by adjusting motor speed of the blower to achieve desired Reynolds number between 4100 and 25,400. The copper test tube had a length of L = 1000 mm, with 50.1 mm inner diameter (D). The tube was heated by a flexible electrical wire continuously wounding around the tube to provide a uniform heat flux boundary condition. The outer surface of the test tube was well insulated to minimize convective heat loss to the surrounding. The wall temperatures (T_w) were measured by 16 chromel-constantan (K-Type) thermocouples located equally on each of the top wall and the side wall along the test tube while the inlet and outlet temperatures of air in the test tube were measured by two RTD-type thermocouples. All the temperature values were read through a fluke 2680A data acquisition system. Also, the pressure drop across the test tube with PRWT insert was measured with a digital manometer. More details on the experimental set-up, method and uncertainty analysis are similar as reported in an earlier paper [10]. The detail of the 45° PRWT is depicted in Fig. 2. The winglet elements made of 0.5 mm thick aluminum strips were attached on both sides of a straight tape using superglue while the tape was made of aluminum with its dimension of 50 mm × 1000 mm × 0.5 mm ($W \times L \times t$). The 45° PRWT was inserted into the tube with four different punched hole diameters, d = 1, 1.5, 2 and d = 3 mm at a single winglet-width ratio (b/D=BR = 0.24) and winglet-pitch ratio (P/D=PR = 3.0). The geometric parameters and test conditions are listed in Table 1.

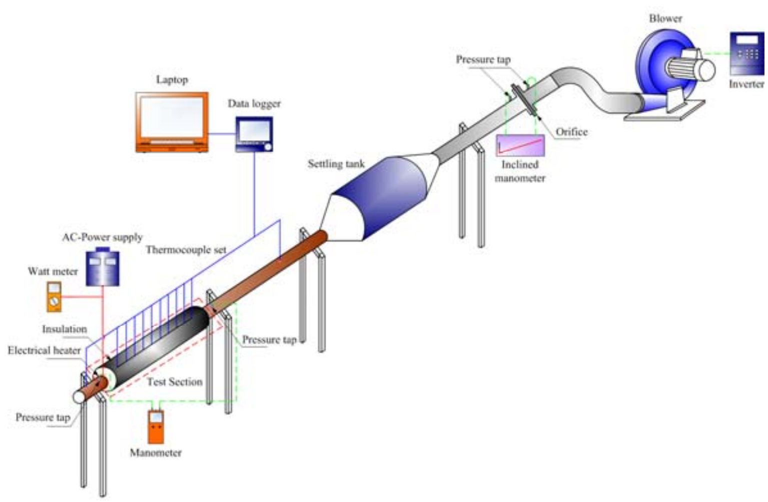

Fig. 1 Schematic diagram of experimental setup.

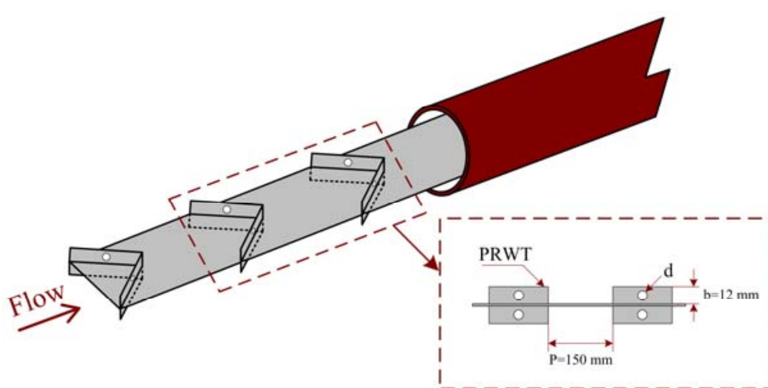

Fig. 2 Test section with 45° PRWT inserts.

Table. 1 Geometric parameters of vortex generators and tested conditions

Working fluid	Air
Reynolds number	4100 - 25,400
Tape width, *W*	50 mm
Tape length, *L*	1000 mm
Relative winglet pitch ratio, *P/D*	3
Relative winglet width ratio, *b/D*	0.24
Punched hole diameter of winglet, *d*	1, 1.5, 2, 3 mm

3. Data reduction

In the present work, air is used as the test fluid. The steady state heat transfer rate is assumed to be equal to the heat loss in the test tube:

$$Q_{\mathrm{air}} = Q_{\mathrm{conv}} \quad (1)$$

in which

$$Q_{\mathrm{air}} = \dot{m}C_{\mathrm{p,air}}(T_{\mathrm{o}} - T_{\mathrm{i}}) = VI \quad (2)$$

The convection heat transfer from the test tube can be written by

$$Q_{\mathrm{conv}} = hA(\tilde{T}_{\mathrm{w}} - T_{\mathrm{b}}) \quad (3)$$

where

$$T_{\mathrm{b}} = (T_{\mathrm{o}} + T_{\mathrm{i}})/2 \quad (4)$$

$$\tilde{T}_{\mathrm{w}} = 16 \quad (5)$$

where for a constant heat flux, the average surface temperature $\tilde{T}_{\mathrm{w}}$ can be calculated from 16 points of the local wall temperatures, T_{w}, lined equally apart between the inlet and the exit of the test tube. The average heat transfer coefficient, *h* and the average Nusselt number, Nu are estimated as follows:

$$h = \dot{m}C_{\mathrm{p,air}}(T_{\mathrm{o}} - T_{\mathrm{i}}) / A\tilde{T}_{\mathrm{w}} - T_{\mathrm{b}}) \quad (6)$$

$$Nu = hD/k \quad (7)$$

The Reynolds number, Re, is given by

$$\mathrm{Re} = UD/\nu \quad (8)$$

Friction factor, *f*, can be written as

$$f = \frac{\Delta p}{(L/D)\rho U^2/2} \quad (9)$$

in which *U* is mean air velocity in the tube.
All thermo-physical properties of air are determined at the overall bulk air temperature (T_b) from Eq. (4).
The thermal enhancement factor (TEF) defined as the ratio of the, *h* of an inserted tube to that of a smooth tube, h_0, at an identical pumping power:

$$\mathrm{TEF} = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right|_{pp} = \left(\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right)\left(\frac{f_0}{f}\right)^{1/3} \quad (10)$$

4. Results and discussion

4.1 Confirmatory test

The present experimental results on the heat and fluid flow characteristics in a uniform heat-fluxed tube without PRWT inserts are initially reported in the form of Nu and *f*. The results of the plain (smooth) tube in the present work are compared with the published correlations of Dittus–Boelter and Petukhov [11] for fully developed turbulent flow in circular tubes.
Correlation of Dittus–Boelter,

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{4/5}\,\mathrm{Pr}^{0.4} \quad (13)$$

Correlation of Petukhov,

$$f = (0.79\ln\mathrm{Re} - 1.64)^{-2} \quad (14)$$

Figs. 3 and 4 show the comparison between the present experimental work and earlier correlations from the previous work. In the figures, the results of the present work agree reasonably well within ±6% for Nu of Dittus–Boelter and ±7% for *f* of Petukhov correlations.

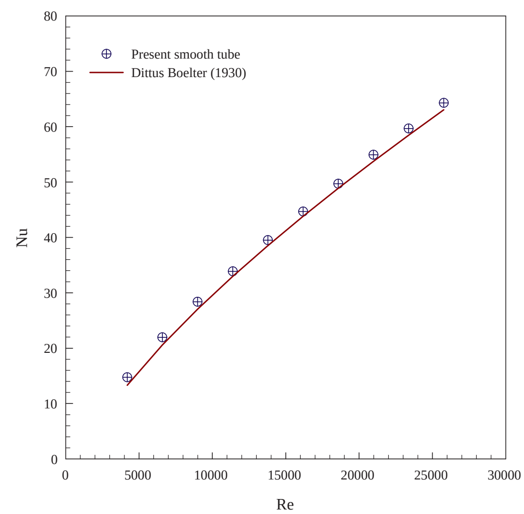

Fig. 3. Confirmatory test of Nu of plain tube.

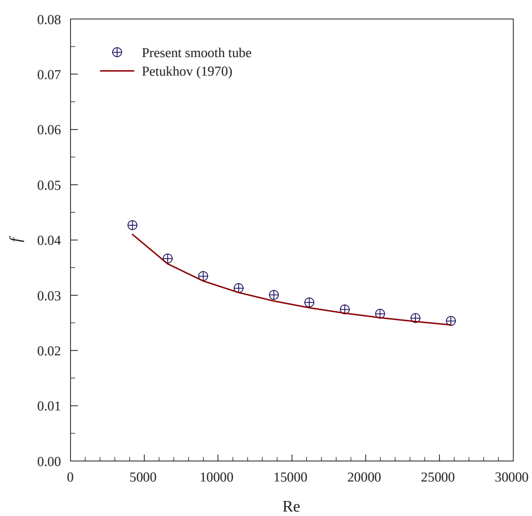

Fig. 4. Confirmatory test of *f* of plain tube.

4.2 Heat transfer results

The comparison of heat transfer (Nu) and Nu/Nu$_0$ of the tube with/without PRWT is demonstrated in Figs. 5(a) and (b), respectively. In Fig. 5(a), Nu of the tube with PRWT was found to be consistently higher than that of the smooth tube. The Nu increases with decreasing *d*, but with the increment in Re. This is because the smaller *d* causing flow circulation and separation provides higher turbulence intensity of the flow than the larger one.

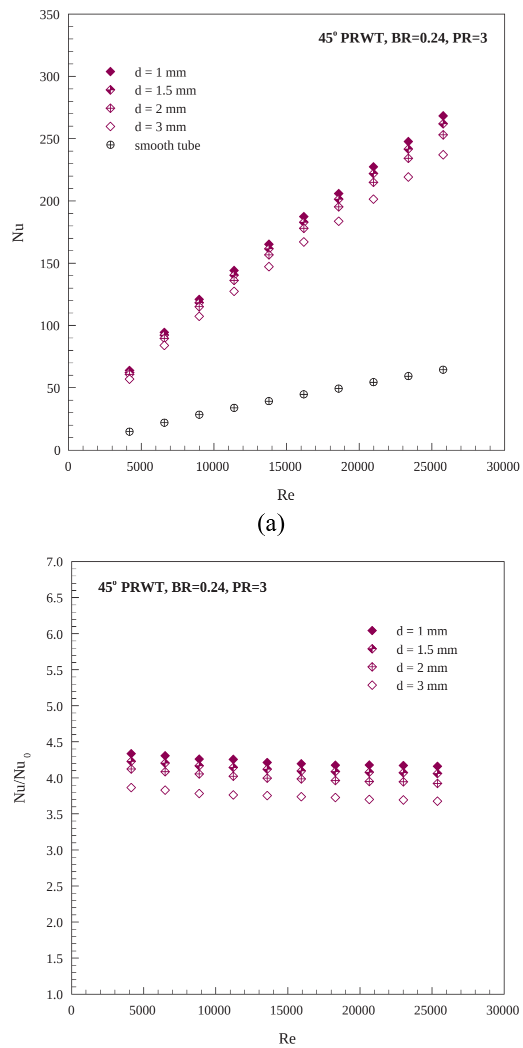

Fig. 5. Comparison of (a) Nu and (b) Nu/Nu_0 with Re for various *d* values.

Fig. 5(b) shows the normalized average Nusselt number ratio, Nu/Nu_0 (defined as a ratio of augmented Nu to Nu of smooth tube). It is noted that Nu/Nu_0 tends to slightly decrease with the rise of Re for all values of *d*. The mean Nu/Nu_0 values are found to be about 4.23, 4.13, 4.01 and 3.75 times above the smooth tube for *d* = 1, 1.5, 2 and 3 mm, respectively.

4.3 Friction factor results

The experimental friction loss in the form of *f* and f/f_0 of the tube fitted with the PRWT as well as a smooth tube under isothermal flow condition are shown in Figs. 6(a) and (b), respectively. It can be seen that the *f* values obtained from the four different *d* values are in a similar trend and decrease with increasing Re and *d*. The increase in friction factor with vortex flow, in general, is much higher than that with axial flow (smooth tube) as can be seen in Fig. 6a.

Fig. 6b presents the variation of friction factor ratio (f/f_0) with Re for various PRWT inserts. It is seen from the figure that the f/f_0 shows the increasing trend with the increment of Re for all cases. As expected, the f/f_0 of the *d* = 1 mm is the highest at a similar operating condition. This can be attributed to higher flow blockage, larger surface area and the act caused by the reverse flow. The PRWT yields the f/f_0 at about 29–45, 26–41, 23–35 and 19–29 times for *d* = 1, 1.5, 2 and 3 mm, respectively.

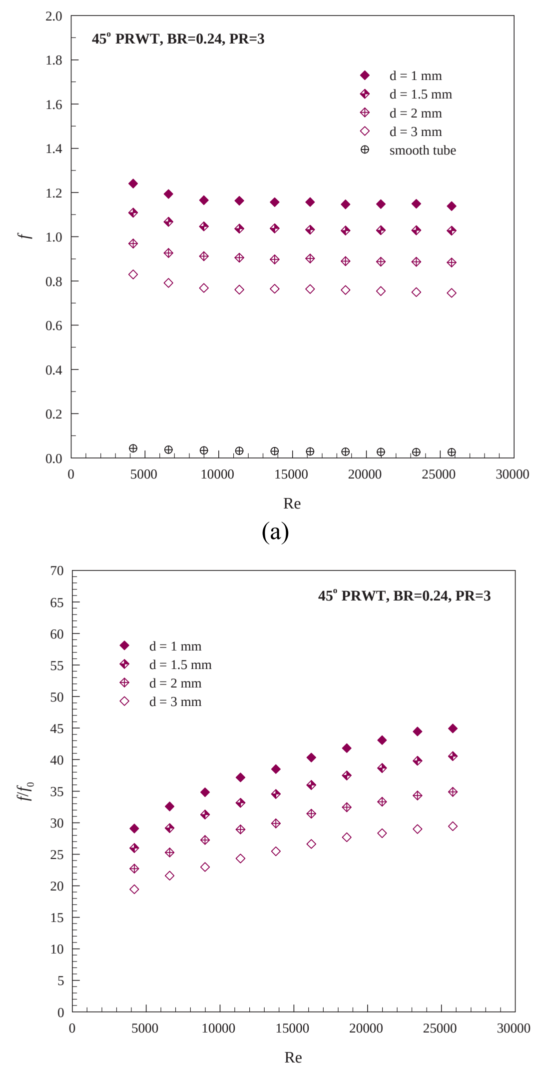

Fig. 6. Comparison of (a) *f* and (b) f/f_0 with Re for four types of PRWT.

4.4 Performance evaluation

As preliminary design guidance, the thermal enhancement factor (TEF) can be introduced based on the same power consumption per unit mass of fluid, is taken into account by using Eq. (10). Fig. 7 shows the variation of TEF with Re. In the figure, the TEF found to be above unity tends to reduce with the increase in Re for all cases investigated. The d=2 mm gives the highest TEF at lower Re due to lower f than the d = 1 and 1.5 mm but yields higher Nu than the d = 3 mm. The TEF values for the 45° PRWT with d = 1, 1.5, 2 and 3 mm are, respectively, about 1.17–1.41, 1.18–1.43, 1.2–1.46 and 1.19–1.44.

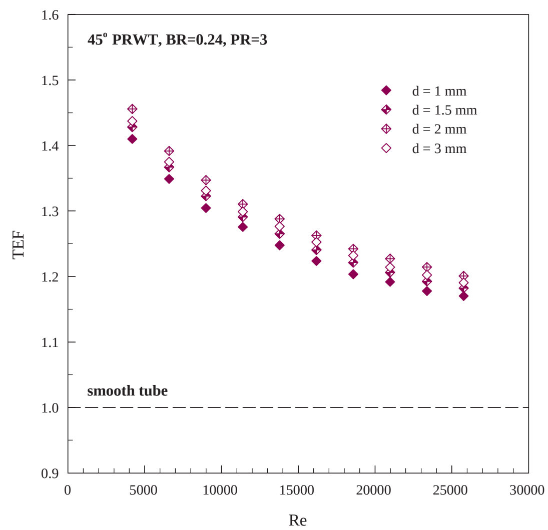

Fig. 7. Effect of TEF on Re.

5. Conclusions

In the present work, an experimental investigation has been conducted to examine the effect of punched pore diameter of the winglet on fluid flow and heat transfer characteristics in a heat exchanger tube. Major findings can be summarized as follows:

- The insertion of the 45° PRWT leads to a considerable increase in the heat transfer rate and pressure loss penalty.
- The heat transfer in the PRWT tube is found to be about 3.68–4.34 times higher than the smooth tube.
- The heat transfer rate and friction factor due to the presence of PRWT increase as the punched pore diameter increases.
- The maximum TEF of 1.46 is found for the PRWT with d = 2 mm at Re= 4100 where Nu and f, respectively increase around 4.12 and 22.71 times of those for the smooth tube.

6. References

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