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## Heat Transfer Enhancement in a Heat Exchanger Tube with 45° Rectangular-Winglet Pairs

To cite this article: P. Tongyote *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **501** 012060

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# Heat Transfer Enhancement in a Heat Exchanger Tube with 45° Rectangular-Winglet Pairs

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**Abstract.** An experimental investigation on thermal characteristics in a heat exchanger equipped with 45° rectangular winglet pairs (RWP) is conducted using air flowing in a constant heat flux tube with Reynolds number ( $Re$ ) in a range of 5,200 to 23,000. Parameters of RWP included three relative height or blockage ratios ( $BR = 0.06, 0.08$  and  $0.1$ ) and two pitch-spacing ratios ( $PR = 1.0$  and  $2.0$ ). The heat transfer and the pressure drop in the test tube are displayed in the form of Nusselt number ( $Nu$ ) and friction factor ( $f$ ), respectively. The investigation shows that  $Nu$  and  $f$  values of the tube with RWP insert are, respectively, in the range of about 1.55–3.2 and 3.76–6.44 times above the plain tube alone, depending on the operating condition. The maximum thermal performance for using the RWP is found to be 1.7 at the lowest  $Re$ .

**Keywords:** heat transfer, pressure loss, thermal enhancement, rectangular-winglet pairs

## 1. Introduction

In several engineering applications and industrials, various techniques are devised to enhance the convection coefficient between fluid flow and heated walls of thermal systems. Vortex flow devices are among the techniques that are commonly used to increase thermal performance of those systems. The employ of vortex flow in thermal systems has been extensively found in the literature [1–4]. For decades, many investigations have been emphasized on heat transfer augmentation by passive techniques. A tube insertion with vortex-flow devices is one of the passive techniques that have been widely employed for augmenting the heat transfer in a heat exchanger tube [5–9].

The aim of heat transfer enhancement in a heating/cooling system is to reduce the exchanger size and its cost apart from the reduction of pumping/blowing power required in a prescribed thermal system for operating cost saving. Thus, several studies have been conducted to examine the influence of insert devices for increasing the heat exchanger performance. For example, vortex-flow devices to produce the vortex flows included wire coils [10, 11], nozzles [12], twisted tapes [13, 14], fin tapes [15, 16], baffles [17, 18], winglets [19, 20], for augmenting the convection heat transfer are the commonly known in the literature. Ozceyhan [21] studied numerically the heat transfer and thermal stress in a constant heat flux tube with wire-coil insert. The turbulent flow friction and thermal behaviors in a circular tube equipped with winglets was examined experimentally by Chokphoemphun



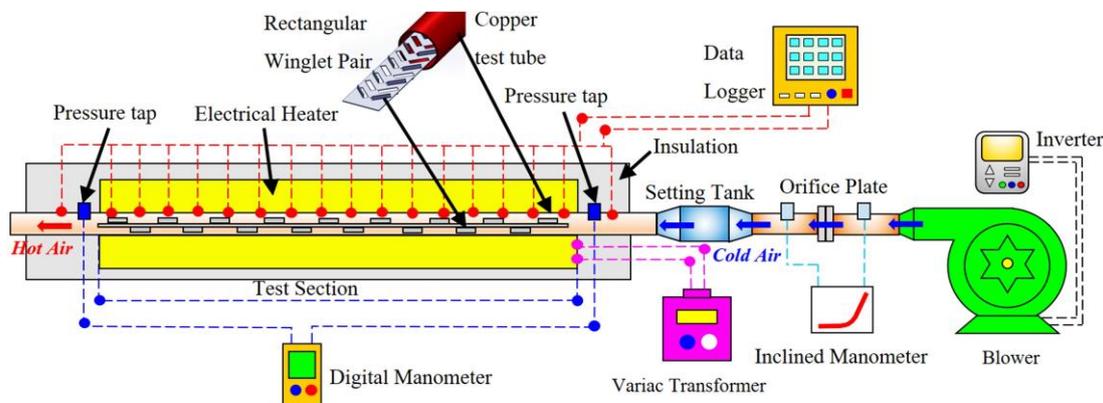
et al. [22]. Recently, Tamna [23] investigated numerically the heat transfer behaviors and flow structure in a tube inserted with winglet pairs. They reported that the use of winglet pairs can induce two main counter-rotating vortices inside the tube leading to greater increase in thermal performance of the heating system

In the literature survey cited earlier, the vortex devices with various configurations have been introduced for augmenting thermal performance of a heat exchanger system.

The heat transfer augmentation is strongly dependent of the device geometry and the proper device geometry can provide higher performance of thermal systems. Among the mentioned devices, the winglet is one of the best efficient vortex-flow devices because of its lowest pressure drop penalty and ease of use by placing on flat surfaces of ducts/channels. In the current study, the  $45^\circ$  winglet pairs mounted inside the test section with two different pitch-spacing ( $P$ ) ratios ( $PR=P/D=1.0$  and  $2.0$ ) and three relative winglet heights ( $b$ ), called blockage ratio ( $BR=b/D = 0.06, 0.08$  and  $0.1$ ) were offered. Owing to curved wall of the tube, the winglets were alternately arranged and mounted repeatedly on a straight tape employed for supporting the winglets before insertion. In the present experiment, air as the working fluid entered the test tube for the  $Re$  ranging from 5,200 to 23,000.

## 2. Experimental Setup

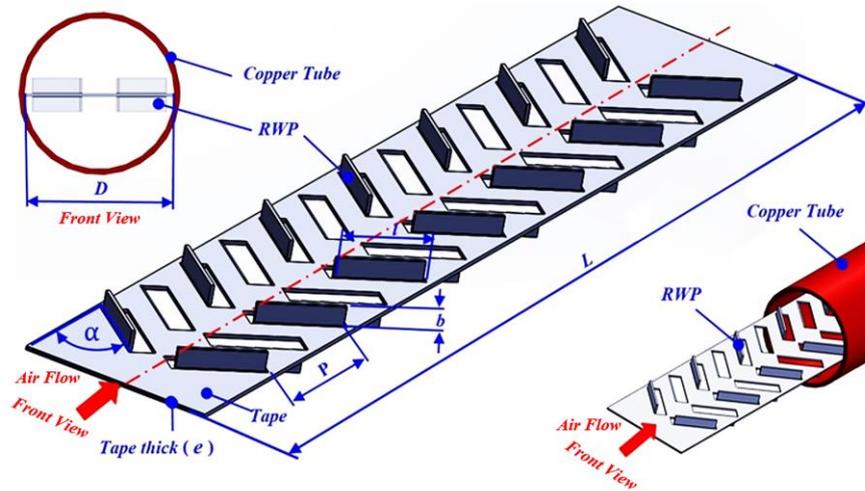
The experimental apparatus is schematically sketched as depicted in figure 1. As can be seen, air induced by a blower was directed through the orifice-type flowmeter, settling tank and then, the test section. The flowmeter used to measure the volumetric airflow rate in the tube was built as per ASME standard [24] where the pressure drop of the orifice plate was measured by an inclined manometer filled by manometric fluid with specific gravity of 0.826. The varied airflow rate in the tube was made using an inverter for controlling the blower-motor speed.



**Figure 1.** Schematic of experimental setup.

The system tube made of copper had a 50-mm inner diameter ( $D$ ) and 3000 mm in total length that included a 1600-mm length ( $L$ ) of the test tube as shown in figure 1. To provide a constant wall heat-flux condition, a continually winding flexible electrical wire was utilized to wrap around the tube. Also, a duct insulation was applied and covered the outermost wall of the test tube to reduce the convection heat loss to the surrounding air. The bulk temperatures of air at the inlet and the outlet ( $T_i$  and  $T_o$ ) were measured by a data acquisition unit in common with the RTD-type PT-100 thermocouples while wall temperature measurements ( $T_w$ ) along the test section were made by 16 thermocouples (T-type) attached on the outer wall tube and equally positioned along the tube wall. The measurement of pressure drop of the test tube was made by using a digital manometer coupled to two static pressure taps across the test tube.

Figure 2 depicts the arrangements of rectangular-winglet pairs (RWP) employed in the current work. The formation of RWP was done by cutting three sides of a preset rectangle marked on a 0.5-mm thick aluminum tape before punching it to become the rectangular winglet as displayed in figure 2. Each of the winglet had a length ( $l$ ) of 19.8 mm whereas the tape had 1600-mm in length. The RWPs were mounted alternately on double sides of the tape as can be seen in figure 2. The RWPs were arranged with a fixed angle of attack ( $\alpha$ ) of  $45^\circ$ , two relative winglet pitches or pitch ratios ( $PR=P/D$ ) and three relative winglet heights ( $b$ ) or blockage ratios ( $BR=b/D$ ). The insertion of the RWP-attached perforated-tape into the test tube is shown in figure 2.



**Figure 2.** Rectangular Winglet Pair (RWP) arrangements.

### 3. Data Processing

In this section, the measured data are rearranged to be in terms of Reynolds number,  $Re$ , Nusselt number ( $Nu$ ), pressure loss in the test tube or friction factor ( $f$ ), and thermal enhancement factor (TEF). The  $Re$  is defined as

$$Re = UD / \nu \quad (1)$$

where  $\nu$  and  $U$  are, respectively, the kinematic viscosity and mean velocity in the tube of air. The  $f$  calculated from the pressure drop ( $\Delta p$ ) of the tube length ( $L$ ) is obtained by

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

In the current work, air entered the insulated tube and thus, the steady-state heat transfer rate is assumed to be equal to the heat absorbed from the tube which can be expressed as:

$$Q_{\text{air}} = Q_{\text{conv}} \quad (3)$$

where

$$Q_{\text{air}} = \dot{m} C_{p,\text{air}} (T_o - T_i) \quad (4)$$

in which  $\dot{m}$  and  $C_{p,\text{air}}$  are the mass flow rate and the specific heat of air, respectively. The convection heat absorbed from the test section is estimated by

$$Q_{\text{conv}} = hA(\tilde{T}_w - T_b) \quad (5)$$

where  $A$  is the heat transfer inner surface area of the test tube and

$$T_b = (T_o + T_i) / 2 \quad (6)$$

$$\tilde{T}_w = \sum T_w / 16 \quad (7)$$

$T_w$  represents the local wall temperature located equally on the top of the test section and the mean wall temperature,  $\tilde{T}_w$  is evaluated by using 16 points of the wall temperatures. The average convection coefficient ( $h$ ) and Nusselt number (Nu) are determined via

$$h = \dot{m} C_{p,air} (T_o - T_i) / A(\tilde{T}_w - T_b) \quad (8)$$

and

$$Nu = hD / k \quad (9)$$

Thermal-physical properties of fluid are evaluated at the bulk air temperature ( $T_b$ ) as given in equation (6).

TEF is defined as the ratio of  $h$  of the tube with RWP insert to  $h$  of the smooth/plain tube alone at an identical pumping/blowing power (pp) and is presented by

$$TEF = \frac{h}{h_{0,pp}} = \frac{Nu}{Nu_{0,pp}} = \left( \frac{Nu}{Nu_0} \right) \left( \frac{f_0}{f} \right)^{1/3} \quad (10)$$

where the subscript “0” stands for the smooth tube acting alone.

## 4. Results and Discussion

### 4.1. Verification of smooth tube

Nu and  $f$  data of the present smooth tube are verified with those from Dittus-Boelter and Blasius correlations [25] as prescribed in equations (11) and (12), respectively, as displayed in figure 3. It can be observed in the figure that the present results agree well within  $\pm 6\%$  with both Nu and  $f$  from the correlations.

Dittus-Boelter correlation,

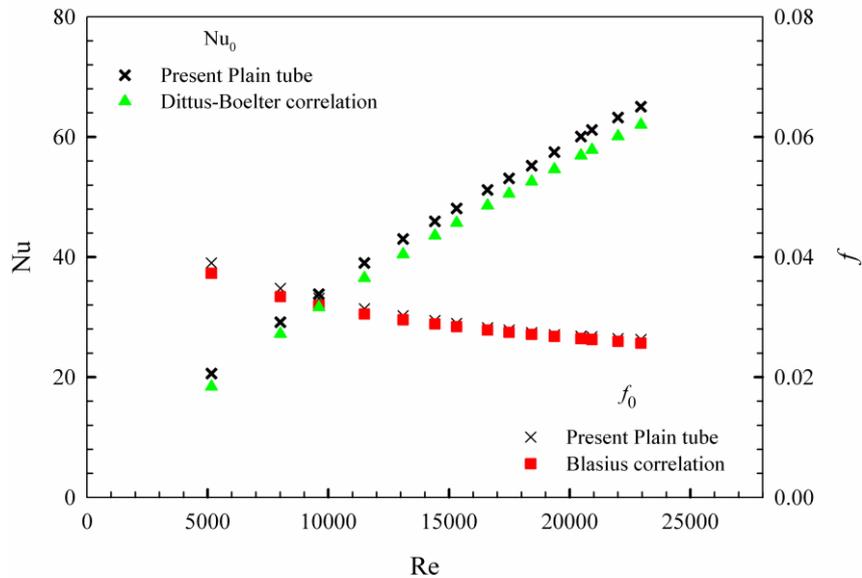
$$Nu = 0.023 Re^{4/5} Pr^{0.4} \quad (11)$$

Blasius correlation,

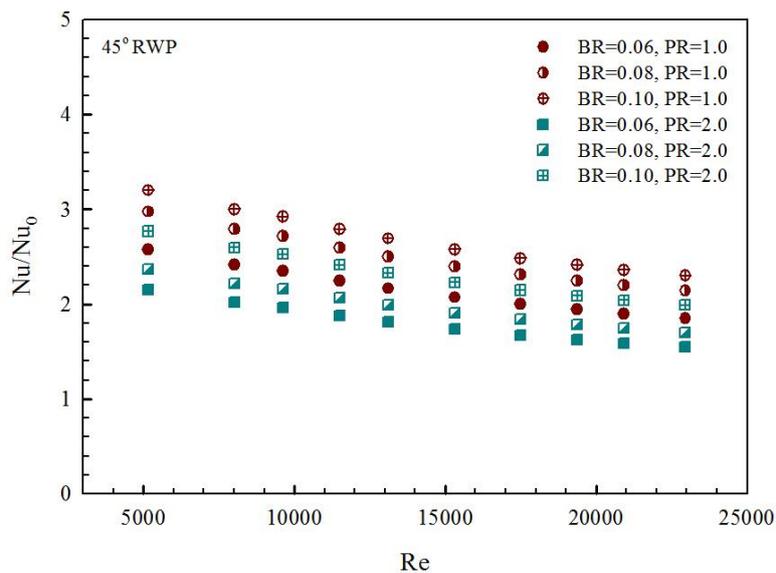
$$f = 0.316 Re^{-0.25} \quad (12)$$

### 4.2. Effect on heat transfer

Figure 4 shows the relationship between Nusselt number ratio,  $Nu/Nu_0$  and Re for using RWPs. In the figure,  $Nu/Nu_0$  tends to decrease slightly with the rise of Re for all cases. It is noted that the case of BR = 0.1 and PR = 1.0 gives the highest  $Nu/Nu_0$ . This is because of higher blockage of flow at BR = 0.1, more flow interruption resulting in higher vortex-flow strength and, then, more efficient convection coefficients between air and the tube surface. The  $Nu/Nu_0$  values are varied in the range of 1.55–3.2 depending on Re, BR and PR, and the mean increases in  $Nu/Nu_0$  for BR=0.06, 0.08 and 0.1 are about 2.15, 2.49, and 2.68; and 1.8, 1.98 and 2.31 times for PR=1.0 and 2.0, respectively.



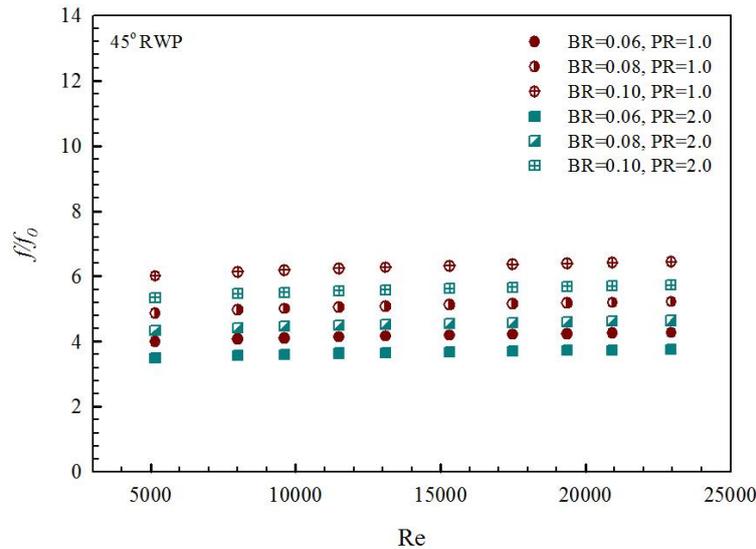
**Figure 3.**  $Nu_0$  and  $f_0$  of the present work with those of published correlations [25].



**Figure 4.**  $Nu/Nu_0$  versus  $Re$  for RWP insert.

#### 4.3. Effect on friction factor

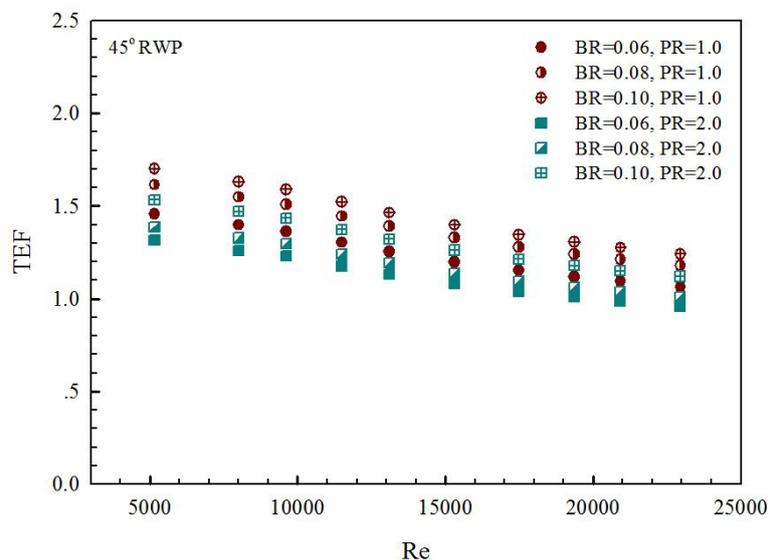
Figure 5 depicts the variation of friction factor ratio,  $f/f_0$  with  $Re$  for RWP inserts. It is clearly observed that the use of RWP leads to the extreme increase of  $f$  higher than the smooth tube acting alone, owing to the fluid dissipation of kinetic energy and larger surface friction from the presence of RWP including the appearance of reversing flow. The increase of  $f/f_0$  for using the RWP is in a range of 3.76–6.44 times and its values are found to be nearly free from  $Re$  at a given BR and PR. The RWP with PR=1.0 provides considerably higher  $f/f_0$  than that with PR=2.0 and also  $f/f_0$  shows the increasing tendency with rising  $Re$  for all RWPs used. The maximum  $f/f_0$  around 6.44 times can be seen for the RWP with BR=0.1, PR=1.0. The mean  $f/f_0$  for the RWP with PR=1.0 is about 12 % higher than the one with PR= 2.0, at a prescribed BR.



**Figure 5.**  $f/f_0$  versus Re for RWP insert.

#### 4.4. Assessment of thermal performance

The TEF variation with Re values for the tube inserted with RWPs is displayed in figure 6. TEF was evaluated from the Nu and  $f$  of the present tube with RWP and those of the smooth tube at a similar pumping/blowing power condition as defined by equation (10). It is interesting to note that the TEF shows the decreasing tendency with the increment in Re for all RWP cases. The maximum TEF of 1.7 can be observed at BR = 0.1, PR = 1.0 at lower Re whereas the lowest one of 0.96 is seen at BR = 0.06, PR = 2.0.



**Figure 6.** Variation of TEF with Re for RWP insert.

## 5. Conclusion

Thermal characteristics in a heat exchanger tube equipped with 45° RWP tapes have been examined experimentally and the current experiment has been carried out for the turbulent tube flow,  $Re = 5,200\text{--}23,000$  for a uniform tube-wall heat-flux. The study shows that  $Nu/Nu_0$  displays the decreasing tendency while  $f/f_0$  exhibits the increasing trend with rising Re. The application of 45° RWP

insert results in a considerable increase in the heat transfer rate above the smooth tube alone around 1.55–3.2 times depending on PR and BR. Also, the increase of  $f/f_0$  for the RWP is seen around 3.76–6.44 times. As a performance indicator, the highest TEF around 1.7 can be observed at BR=0.1 and PR=1.0.

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