The 6th TSME International Conference on Mechanical Engineering
16-18 December 2015

TSF019

Effect of Twist Angle of Freely Rotating Swirl Flow Devices on the Thermal Performance of a Common Circular Tube

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Abstract

The experimental heat transfer performance and flow characteristics for water flowing through a common plain tube fitted with freely rotation swirl flow devices under turbulent flow regime have been studied. A uniformly heating condition was conducted to supply heat load to the test section. Freely rotation swirl flow devices with 20 mm in length and twist angle of 30^{o}, 60^{o}, 90^{o} and 120^{o} are tested and compared. For each twist angle, five pieces of swirl flow devices are installed inside a common plain tube at 0.46 m equally distant along the test tube. A common stainless steel tube with OD of 10 mm, thickness of 0.4 mm and 2,300 mm in length is used as the test section. The experimental data illustrated that the heat transfer performance of swirl flow device with twisted angle of 30^{o} is lowest value than the other. Similarly, swirl flow device with twist angle of 60^{o} give the best value of the performance evaluation criteria. Moreover, the tube with swirl flow devices inserts with twist angle of 120^{o} provided maximum pressure drop compared with the other.

Keywords: swirl generator, heat transfer coefficient, twist angle, pressure drop

1. Introduction

Heat transfer enhancement techniques have been continuously developed and improved over many years to apply in industrial heat transfer equipment such as power generation, petrochemical industry, automotive, metallurgical industry, refrigeration and process industry. It can be classified into two methods: active technique and passive technique. For the active technique, heat transfer performance is enhanced by adding extra power such as jet impingement, stirring of the fluid, fluid and surface vibration, and electric field. In the passive method, the modifications of surface geometry are required such as extended surface, treated surface, surface roughening as well as swirl flow device inserts. Focusing on the passive technique, the common tube with swirl generator inserts is one of the most effective methods to augment the heat transfer performance of the heat exchangers. The main reason of this enhancement is swirling flow which leads to an increase in fluid mixing and decrease in the thermal boundary layer thickness. Twisted tape swirl generator fitted in the common plain tube is the basically heat transfer enhancement technique to augment the thermal performance of the heat exchangers. This is due to many reasons such as easy to install, simple configurations and steady performance [1]. Heat transfer enhancement and pressure drop characteristics of plain tube with twisted tape inserts under laminar flow condition were carried out by Pathipakka and Sivashanmugam [2], Naphon [3], and Bhattacharya and Saha [4]. The experimental investigations for twisted tape inserts were also conducted by Eiamsa-ard et al. [5] and Bas and Ozceyhan [6]. However, additional of some devices in the main flow can promote penalty drop in pressure. Thus, a number of researchers aimed to develop and improve the new type of swirl generator configurations for maximizing the thermal performance to pressure drop ratio. Some recent examples of experimental approach are described as follows:

Garcia et al. [7] studied the thermal and hydraulic performance of a common plain tube fitted with various helical-wire-coil inserts. Tandiroglu [8] investigated the effect of flow geometry parameters of common tube with baffle inserts on the heat transfer performance and pressure drop in transient conditions. Later, the thermal performance and pressure drop of a common circular tube fitted with a full-length helical screw element with different twist ratios and spacer have been studied by Sivashanmugam and Suresh [9]. Chang et al. [10] reported on the local heat transfer coefficient and flow characteristic of a round tube equipped with a broken twisted tape. Garcia et al. [11] studied the thermal performance and pressure drop characteristic of a circular tube fitted with three different wire coils. Similarly, effect of propeller-type turbulators fitted into a circular tube on the entropy generation rate and exergy loss rate were reported by Kurtbas et al. [12]. Sarac and Bali [8] heat transfer and flow characteristics of a common round tube equipped with a decaying swirl flow. Gul and Evin [13] investigated the heat transfer and pressure drop features of a short helical tape fitted at the entrance of the common tube. Eiamsa-ard et al. [14] studied the thermal performance and flow characteristic of a round tube equipped with a propeller-type swirl generator inserts. Kurtbas and colleges [15] reported the heat transfer and friction characteristics of a circular tube with a conical injector-type swirl generator (CITSG) inserts. Akhavan-Behabadi et al. [16] presented an experimental investigation on the heat transfer coefficient and pressure drop of a common tube implanted with coiled wire during the heating of engine oil. Khalil et al. [17] studied the heat transfer and pressure drop characteristics of turbulent swirling air flow through a sudden expansion pipe, experimentally. Similarly, Zohir et al. [18] reported the thermal performance and flow characteristics of a sudden expansion pipe fitted with a spiral spring and a propeller-type swirl generator. Ibrahim [19] studied the heat transfer enhancement and pressure drop of flat tubes implanted with full-length helical screw-tape. Likewise, Patil and Babu [20] investigated the heat transfer and friction characteristics of a double-pipe counter flow heat exchanger fitted with full-length twisted tape and full-length screw tape inserts. Recently, Duangthongsuk and Wongwises [21, 22] investigated the thermal and hydraulic performances of a common plain tube fitted with turbine type swirl generators. The effect of free rotation of swirl flow devices was reported.

As aforementioned, many researchers reported that the common tube fitted with some swirl flow devices gave better heat transfer enhancement than the plain tube. However, higher pressure drops were also obtained due to their almost being fixed or their not freely rotating. In the present study, heat transfer enhancement and pressure drop characteristics of a common circular tube fitted with freely rotating turbine-type swirl generator are studied. This type of swirl generator is new design. The effect of twist angle on the heat transfer performance and pressure drop characteristics is presented.

2. Experimental Apparatus and Method

In the present study, the effect of twist angle of freely rotating turbine type swirl generator on the thermal performance and pressure drop is investigated be means of an experimental approach. The experimental system is schematically depicted as shown in Fig. 1. It consisted of two test sections: the first test section was the common round tube, and the second was the tube fitted with rotating turbine type swirl generators ($RTSG_S$) at five position along the test tube. $RTSG_S$ with twist angle of 30º, 60º, 90º and 120º are tested and compared. Furthermore, it comprised of a pump with an inverter, two cooling tanks, a DC power supply, and a rotameter. DI water was used as working medium and flowing under constant heat flux condition. Stainless steel tubes with 10 mm in outer diameter, 0.4 mm in thickness and 2.3 m in length were used as the test tube. For the common tube fitted with $RTSG_S$ inserts, five $RTSG_S$ were installed at x/L of 0.0, 0.2, 0.4, 0.6 and 0.8, respectively. $RTSG_S$ with twist angles of 30º, 60º, 90º and 120º were tested and compared. Each $RTSG_S$ had four blades and was installed inside the transparency tube to observe the rotation behavior. They were made from aluminum material with 2 cm in length, 0.5 mm thickness, and 6.6 mm in outer diameter. The configuration of the $RTSG_S$ is shown in Fig. 2. A differential pressure transmitter is used to measure the pressure drop across the test section. T-type thermocouples were mounted at both ends of the test section for measuring the bulk fluid temperature. Also, 10 T-type thermocouples were installed at different longitudinal positions on the outer surface for measuring the wall temperatures. The locations of the wall temperature measurements and the RTSG positions are illustrated in Fig. 3. In order to simulate wall heat flux, a DC power supply was used. For cooling tank No. 1, it consisted of a cooling coil with 1 TR cooling capacity, a 4 kW electric heater, and an RTD with a temperature controller for setting the fluid temperature at a desired value. Likewise, cooling tank No. 2 was used to reduce the fluid temperature leaving from the test tube to the temperature of tank No. 1. It consisted of with cooling coil with 1.5 TR cooling capacity, and an RTD with a temperature controller. A rotameter was used to measure the fluid flow rate. The flow rate was controlled by adjusting the rotation speed of the pump.

In order to ensure the reliability of the measured data, the YOKOKAWA differential pressure transmitter was calibrated using an air-operated dead weight tester. The uncertainty of the pressure measurement was ± 0.030 kPa. All of the temperature-measuring devices were well calibrated using standard precision mercury glass thermometers. The uncertainty of the temperature measurements was ± 0.1 ºC. The rotameter was specially calibrated by the manufacturer, and its accuracy was ± 7% of full scale.

For testing conditions, Reynolds number ranged between 4,500 to 9,500 was tested. Heat flux was kept constant at 15 kW/m^2 by adjusting the voltage regulator. The fluid temperature was kept constant at 15 ºC. When a steady state reached, any data such as the wall temperatures and the inlet and outlet temperatures of the water and the pressure drop and flow rates of working fluid were recorded.

3. Data Reduction

Before evaluating the thermal performance and pressure drop characteristic of the common round tube implanted with swirl generator, the reliability and accuracy of the experimental system were necessary verified. A common circular tube was tested. The heat transfer coefficient and friction factor obtained from the measured data were compared with the calculated values achieved from the Gnielinski's equation [23] and the Colebrook's equation [24], respectively.

The Gnielinski equation is computed using the Eq. (1) as follows:

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\,\mathrm{Pr}}{1+12.7(f/8)^{0.5}(\mathrm{Pr}^{2/3}-1)} \qquad (1)$$

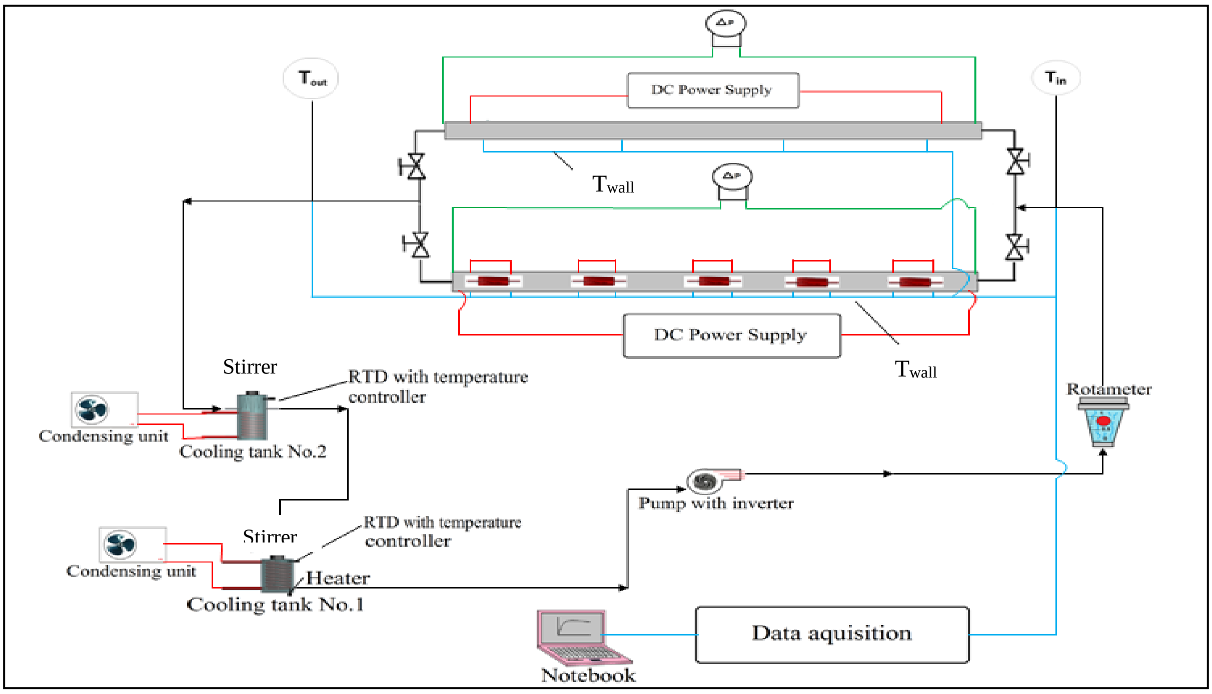
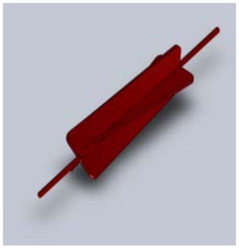
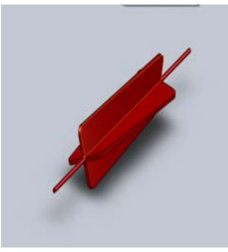


Fig.1 Schematic diagram of the experimental system used in the experimental apparatus

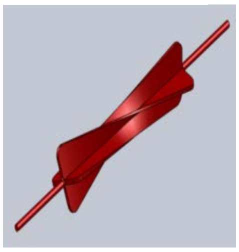


a) 30°



b) 60°

c) 90°



d) 120°

Fig. 2 Configuration of the swirl flow generator used in the present work

where *Nu* is the Nusselt number, *Re* is the Reynolds number, *Pr* is the Prandtl number, and *f* is the friction factor.

For friction factor, The Colebrook equation is used as follows:

$$\frac{1}{\sqrt{f}} = -2.0\ell og\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \qquad (2)$$

where *D* is the inner diameter of the test section, and ε is the roughness of the tube.

The pressure drop across the test section is computed from:

$$\Delta P = f\frac{L}{D}\frac{u_m^2}{2}\rho \qquad (3)$$

where *ΔP* is the measured pressure drop, *L* is the length of the tube, ρ is the density of fluid and u_m is the mean velocity of fluid.

Then, the heat transfer performance of plain tube and tube fitted with $RTSG_S$ can be calculated from the following equation.

The heat load to the test section can be calculated from:

$$Q_{ts} = VI \qquad (4)$$

The heat transfer rate into the fluid was computed from:

$$Q_f = \dot{m}Cp(T_{out}-T_{in}) \qquad (5)$$

Then, the average heat transfer rate was computed from:

$$Q_{ave} = \frac{(Q_{ts}+Q_f)}{2} \qquad (6)$$

where Q_{ts} is the heat transfer rate to the test section, Q_f is the heat transfer rate of the fluid, Q_{ave} is the average heat transfer rate, V is the electric voltage, I is the electric current, $\dot{m}$ is the mass flow rate of the fluid, Cp is the specific heat, and T_{in} and T_{out} are the inlet and exit temperature of the fluid, respectively.

The average heat transfer coefficient is calculated as follows:

$$h = \frac{Q_{ave}}{A(T_w - T_f)} \qquad (7)$$

$$Nu = \frac{hD}{k} \qquad (8)$$

where h is the average heat transfer coefficient, T_w is the average wall temperature of the test section, T_f is the bulk temperature of the fluid, and k is the thermal conductivity of the fluid.

Finally, the Reynolds number and Prandtl number are defined as follows:

$$\mathrm{Re} = \frac{\rho u_m D}{\mu} \qquad (9)$$

$$\mathrm{Pr} = \frac{\mu Cp}{k} \qquad (10)$$

where, μ is the fluid viscosity.

Finally, performance evaluation criteria (PEC) can be calculated as follows:

$$PEC = \frac{(Nu_e / Nu_o)}{(f_e / f_o)^{1/3}} \qquad (11)$$

where, subscript e and o are enhance tube and common tube, respectively.

For pressure drop data, the measured data for the common round tube were compared with the data for the common tube with RTSGS inserts.

4. Result and Discussion

To validate the reliability and accuracy of the experimental system, the thermal and hydraulic performances of the common circular tube are tested. Then, the measured data for Nusselt number and friction factor are compared with the predicted values from the Gnielinski equation and Colebrook equation, respectively.

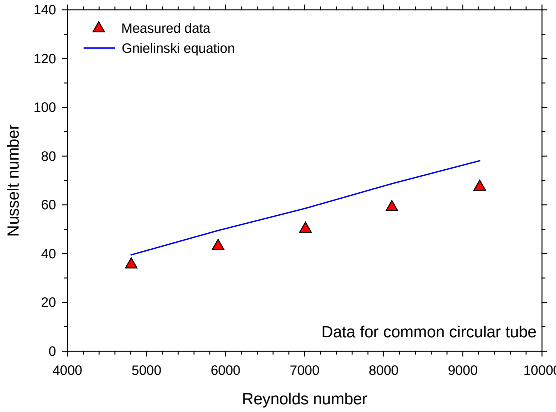


Fig. 3 Measured data and the Gnielinski equation for Nusselt number of common circular tube

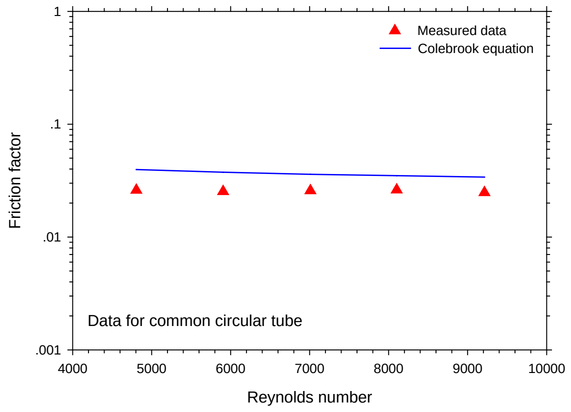


Fig. 4 The measured data and the Colebrook equation for friction factor of common circular tube

As shown in Figs. 3 and 4, the measured data show good agreement between the experimental values for the common circular tube and the predicted values.

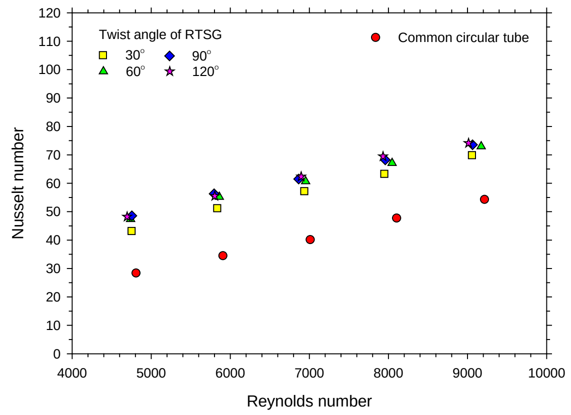


Fig. 5 Comparison of the Nusselt number between the common circular tube and the tube fitted with RTSG at different twist angle

Fig. 5 shows the comparison of the Nusselt number obtained from the common circular tube and that obtained from the common tube implanted with RTSG$_S$ at various twist angle. The data indicate that the thermal performance of the common tube fitted with RTSG$_S$ is higher than that of the common circular tube and slightly increased with increasing twist angle. This behavior is due to high swirling flow of fluid after the swirl generator is crated, which lead to increase in the heat transfer performance. The RTSG with twist angle of 30° give smallest Nusselt number compared with the other twist angle. The results also show that RTSG with twist angle of 60° provide higher heat transfer performance than the RTSG with 30° twist angle by average 6.22% whereas the RTSG with 90° and 120° twist angle give larger heat transfer performance than that of 60° twist angle RTSG about 1.92% and 2.78%, respectively. This mean that using of RTSG with twist angle over than 60° could increase the heat transfer performance by a few percent compared with twist angle of 60°.

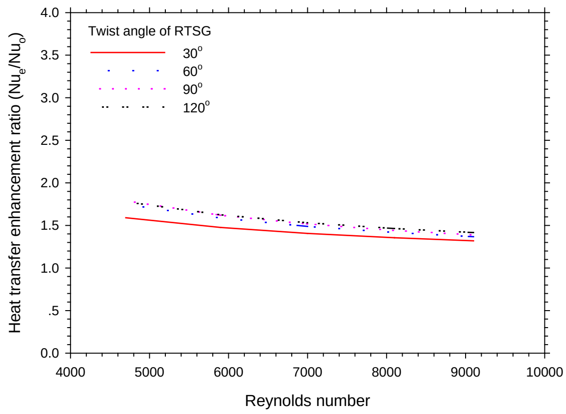


Fig. 6 Heat transfer enhancement ratio as a function of twist angle

Fig. 6 shows the effect of twist angle of RTSG on the heat transfer enhancement ratio. The measured data demonstrate that the use of RTSG fitted in the common circular tube can increase the heat transfer performance by about 1.5 times compared with the common bar tube.

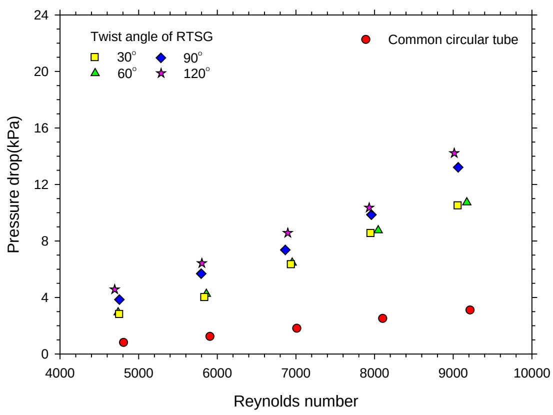


Fig. 7 Comparison of the pressure drop between the common circular tube and the tube fitted with RTSG at different twist angle

Comparison of the pressure drop across the test section between common circular tube and RTSG with different twist angles inserts is show in Fig. 7. The results show that the pressure drop of the common bare tube is less than that of the tube equipped with RTSG. This is due to the fact that the addition of the swirl flow device into the common tube will provide high flow resistance which led to a higher penalty in the pressure drop. Moreover, pressure drop of RTSG with 60° twist angle is quite equal to the RTSG with 30° twist angle and significantly increased can be seen for twist angle of 90° and 120°, respectively. This mean that the use of RTSG with 60° twist angle provide higher heat transfer performance and moderate pressure drop is obtained, when compare to the other twist angles.

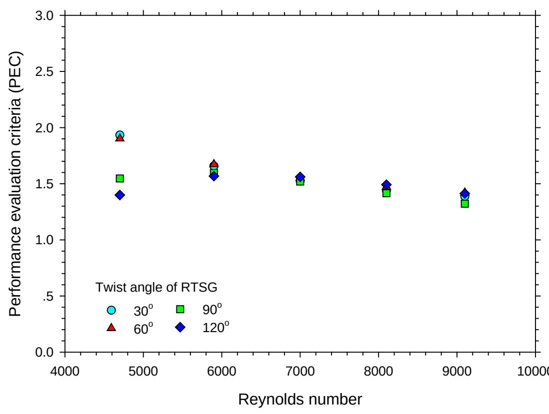


Fig. 8 Variation of the PEC value with Reynolds number for different twist angles

The variation of PEC as a function of twist angle as well as the Reynolds number is show in Fig. 8. From this figure, it is clearly seen that the PEC values of RTSG inserts lies in the range of 1.32 – 1.93. The data also indicate that the tube fitted with RTSG with twist angle of 60° provide higher heat transfer performance compare to the other twist angle.

5. Conclusions

The paper reports the thermal and hydraulic performances of a common circular tube fitted with freely rotating turbine-type swirl generator (RTSG). Effect of twist angle is presented and then compared with the data for common circular tube. Major findings are observed as follows:

- The heat transfer performance of the common tube with RTSG inserts is greater than that of the common bare tube and slightly increase as twist angle increases.
- RTSG with twist angle of 60° provide higher PEC than that of the other twist angles. High heat transfer performance is obtained. On the contrary, moderate pressure drop is observed.

6. Acknowledgement

The authors would like to express their appreciation to the Thailand Research Fund (TRF), the National Science and Technology Development Agency and the National Research University Project for providing financial support.

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