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Thermal Augmentation in Circular Tube with Inclined-Baffle Vortex Generators

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Abstract

The influence of inclined-baffle vortex generators (BVGs) on heat transfer and turbulent flow friction characteristics in a uniform heat-fluxed tube is experimentally investigated in the present work. The aim at using the BVG is to create counter-rotating vortices in the tube to help increase the turbulence intensity leading to higher heat transfer enhancement. The effect of insertion of the 60° BVGs with geometry parameters, such as two baffle-to-tube height ratios or blockage ratios (BR=b/D=0.1 and 0.2) and three baffle-pitch to tube-height ratios (PR=P/D=0.5, 1.0, and 1.5) is examined. In the experiment, air as the test fluid was passed through the test tube for the Reynolds number from 5000 to 24,000. The experimental results of heat transfer and pressure loss in terms of Nusselt number and friction factor are compared between the inserted tube and the smooth tube. The 60° BVG with larger BR provides higher heat transfer and friction loss than the one with smaller BR while the smaller PR performs better than the larger. However, the BVG at BR=0.1 and PR=1.0 yields the highest thermal performance at lower Reynolds number.

***Keywords**:* inclined-baffle vortex generator; tube; heat transfer; pressure loss; thermal performance.

1. Introduction

The technology of enhanced heat transfer has received strong attention over the past 3–5 decades; heat transfer augmentation techniques can be applied mainly in the design of more compact heat exchangers found in various industries, especially refrigeration, automotive and chemical processes. Swirl/vortex flow devices form an important group of the passive augmentation technique. Insertion of swirl/vortex generators in a circular tube is a simple passive technique for enhancing the convective heat transfer coefficient on the tube of a heat exchanger due to their advantages of easy fabrication and operation as well as low maintenance. In addition, the performance of using the turbulators strongly depends on their geometries. In earlier investigations, several turbulators were utilized to promote heat transfer in the tube flow. There have been many types of vortex generators employed in the heat exchanger tubes such as twisted tapes [1], coiled wires [2,3], ribs/baffles [4,5] and winglets [6].

A comparison of the thermal and hydraulic performances between twisted-tape inserts and coiled-wire inserts was introduced by Wang and Sunden [7] for both laminar and turbulent flow regions. They found that the coiled wire performs effectively in enhancing heat transfer in a turbulent flow region, whereas the twisted tape yields a poorer overall efficiency. Rahai et al. [8] experimentally investigated the influences of wire-coil pitch spacing on the mixing enhancement of a turbulent jet from a Bunsen burner. Heat transfer enhancement of flowing water in a tube with flow drag reduction additives by inserting wire coils was presented by Inaba and Haruki [9]. Apart from experimental investigations, the numerical studies on heat transfer enhancement by means of the circular ring turbulators were also reported [10,11]. Ozceyhan et al. [10] numerically studied effect of space between the circular cross sectional rings on heat transfer rate and friction factor. Similarly, Akansu [11] numerically investigated the effect of pitch spacing of porous rings and found that the heat transfer rate and friction factor increase with decreasing the ring spacing. Promvonge et al. [12,13] studied experimentally and numerically the turbulent square-duct flow over the 30° angle-finned tapes inserted diagonally. They reported that at smaller fin pitch spacing, the finned tape at e/H=0.2 and P/H=1 gives the best thermal performance.

In the present work, another type of inserted devices has been developed by insertion of multiple 60°-inclined BVGs into a tube. The present inserted device has been developed from a combination of the merits of rib, baffle, winglet and twisted tape turbulators. This means that the present inserted device is expected to provide a drastically high heat transfer rate like baffles, low pressure drop like angled ribs, swirl/vortex flow as winglets and ease of practical use like twisted tapes. Therefore, a new 60° inclined BVG insert is offered to yield higher thermal performance than the wire-coil insert as reported in Ref. [2]. The experimental results using air as the test fluid for the 60°-inclined BVGs inserted in the tube are presented for turbulent flows in Reynolds number from 5000 to 24,000 in the current work.

2. Experimental setup

The experimental work was conducted in an open-loop experimental facility as shown in Fig. 1. The loop consisted of a circular pipe connected a high-pressure blower to a settling tank, and an orifice flow-meter was placed in this pipeline. A tube including a calming tube (3000 mm), test section (1 m) and exit (0.1 m) was employed after the settling tank. The copper test tube has a length of L = 1000 mm, with 50.8 mm inner diameter (D), 53 mm outer diameter (D_o), and 1.5 mm thickness (t) as depicted in Fig. 2. The inclined BVGs made of 0.5 mm aluminum strips also given in Fig. 2 were connected to each other by a straight steel rod placed on the center of a semicircle steel rod used as a support on both sides of the BVG connecting rod. The 60° BVGs with various geometry parameters, including two different baffle-to-tube height ratios or blockage ratios (BR=b/D=0.1 and 0.2) and three baffle pitch to tube diameter ratios (PR=P/D=0.5, 1.0, and 1.5) were inserted into the tube. The tube was heated by continually winding flexible electrical wire to provide a uniform heat-flux boundary condition. The electrical output power was controlled by a variac transformer to obtain a constant heat-flux along the entire length of the test section. The outer surface of the test tube was well insulated to minimize heat loss to surroundings. The inlet and outlet temperatures of the bulk air were measured at certain points with a data logger in conjunction with the RTD thermocouples calibrated within ±0.2 °C deviation by thermostat before being used while twenty four type-K thermocouples were tapped equally along the local top and side walls of the grooved outer tube surface.

The inlet bulk air from a 1.5 kW high pressure blower was directed through the orifice flow-meter and passed to the heat transfer test section in turbulent region, Reynolds number from 5000 to 24,000. Manometric fluid was used in a U-tube manometer with specific gravity of 0.826 to ensure reasonably accurate measurement of pressure drop. Also, the low pressure drop of the tested tube was measured with an inclined/U-tube manometer. The volumetric air flow rates from the blower were adjusted by varying motor speed through an inverter. During the experiments, the bulk air was heated by an adjustable electrical heater wrapping along the test section.

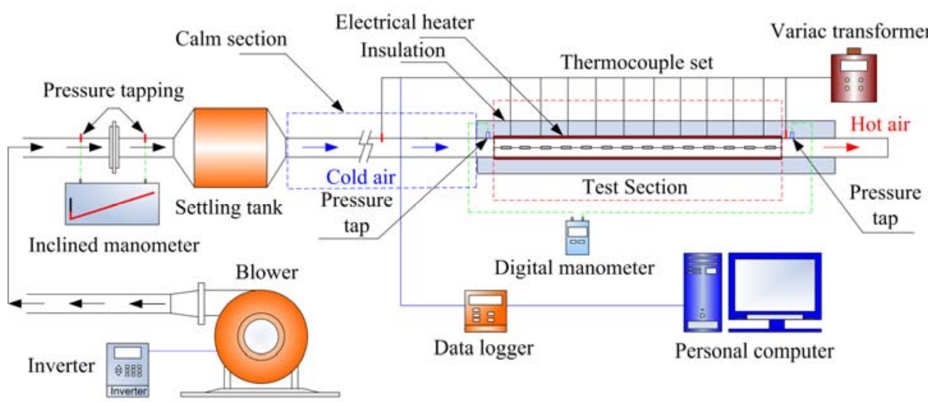


Fig. 1. Schematic diagram of experimental apparatus.

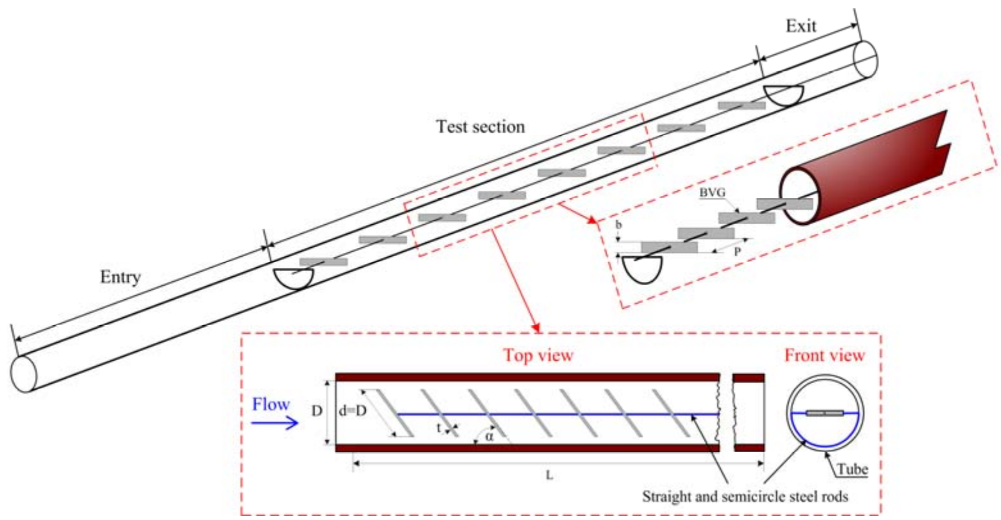


Fig. 2. Test section with 60°-inclined BVGs.

3. Data reduction

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

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

in which

$$Q_{air} = \dot{m}C_{p,air}(T_o - T_i) = VI - heat\ losses \quad (2)$$

The heat supplied by an electrical heater in the test tube is found to be 3 to 5% higher than the heat absorbed by the fluid for thermal equilibrium test due to convection and radiation heat losses from the test tube to surroundings. Thus, only the heat transfer rate absorbed by the fluid is taken for internal convective heat transfer coefficient calculation. The convection heat transfer from the test tube can be written by

$$Q_{conv} = hA(\tilde{T}_s - T_b) \quad (3)$$

in which

$$T_b = (T_o + T_i)/2 \quad (4)$$

$$\tilde{T}_s = \sum T_s / 24 \quad (5)$$

where for a constant heat-flux, the average surface temperature $\tilde{T}_s$ can be calculated from 24 points of the local surface temperatures, T_s, lined equally apart between the inlet and the exit on the top and the side walls of the test tube. The average heat transfer coefficient, h and the average Nusselt number, Nu are estimated as follows:

$$h = \dot{m}C_{p,air}(T_o - T_i)/A(\tilde{T}_s - T_b) \quad (6)$$

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

The Reynolds number is given by

$$Re = UD/\nu \quad (8)$$

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 from Eq. (4).

The thermal enhancement factor, *TEF*, defined as the ratio of the, h of an augmented surface to that of a smooth surface, h_0, at an identical pumping power:

$$TEF = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f_0}{f}\right)^{1/3} \quad (10)$$

4. Results and discussion

4.1 Verification of smooth tube

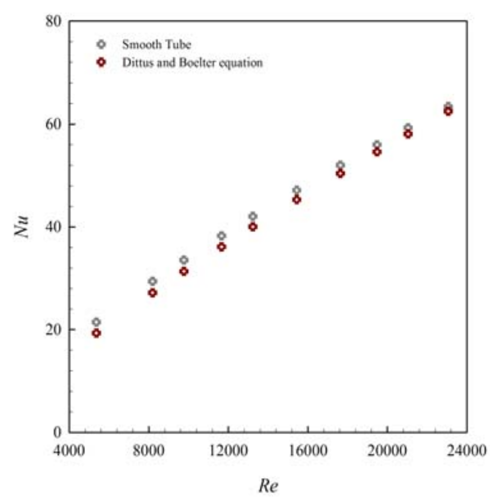
The present experimental results on heat transfer and friction characteristics in a smooth wall tube are first validated in terms of Nusselt number, (*Nu*) and friction factor, (*f*). The *Nu* and *f* obtained from the present smooth tube are, respectively, compared with the correlations of Dittus–Boelter for *Nu*, and of Blasius for *f* found in the open literature [14] for turbulent flow in ducts.

Correlation of Dittus–Boelter,

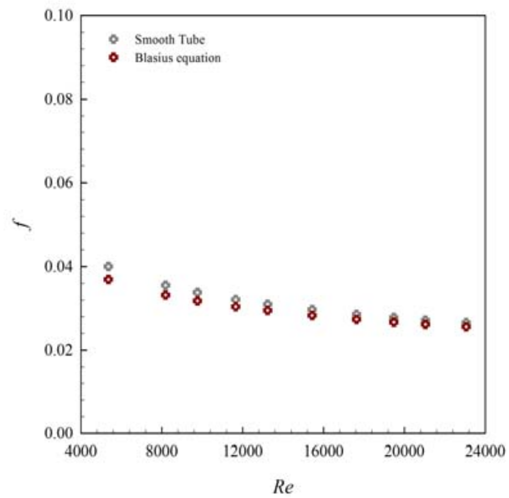
$$Nu = 0.023Re^{0.8}Pr^{0.4} \quad (11)$$

Correlation of Blasius,

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



(a)



(b)

Fig. 3. Verification of (a) *Nu* and (b) *f* for smooth tube.

Figure 3a and b shows a comparison of the *Nu* and *f* obtained from the present work with those from correlations of Eqs. (11) and (12). In the figure, the present results agree very well within $\pm$5% for *Nu* and $\pm$6% for *f*, respectively.

4.2 Heat transfer

The experimental results on heat transfer and friction characteristics in a uniform heat-fluxed tube fitted repeatedly with 60° BVGs at two baffle-to-tube height or blockage ratios (BR=0.1 and 0.2) are presented in the form of *Nu* and *f*, respectively. The *Nu* values for all cases investigated are presented in Fig. 4. In the figure, the BVG inserts yield the considerable heat transfer enhancement with similar trends in comparison with the smooth tube and the *Nu* increases with the increment of Reynolds number (*Re*). This is because the BVG interrupt the development of thermal boundary layer of the fluid flow and increase the turbulence intensity of the flow. It is worth noting that the *Nu* for the BVG with BR=0.2 is considerably higher than the one with BR=0.1. This is caused by higher blockage of using BR=0.2, interrupting the flow and diverting its direction thus promoting high levels of mixing over the others.

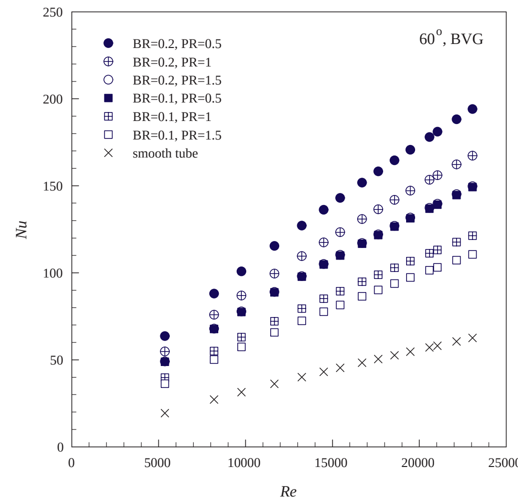


Fig. 4 Variation of *Nu* with *Re* for 60° BVGs.

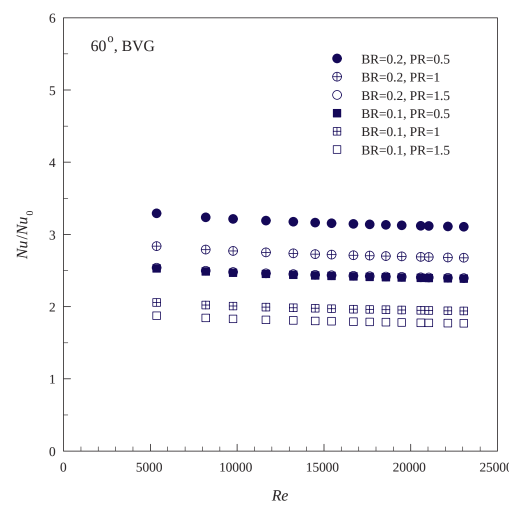


Fig. 5 Variation of Nu/Nu_0 with *Re*.

The Nusselt number ratio, Nu/Nu_0, defined as a ratio of augmented Nusselt number to Nusselt number of smooth tube, plotted against the *Re* is displayed in Fig. 5. In the figure, it is found that the Nu/Nu_0 increases with reducing the PR, apart from the *Re*. This is caused by higher flow blockage of using BR=0.2 and smaller PR interrupting the flow leading to stronger swirl/vortex flow strength and thus promoting high levels of mixing over others. For the BVGs with PR=0.5, 1.0 and 1.5, the increases in Nu/Nu_0 values at BR=0.2 and 0.1 are, respectively, about 3.10–3.3, 2.68–2.84 and 2.39–2.54; and 2.38–2.53, 1.94–2.06 and 1.77–1.88 times above the smooth tube, depending on *Re* values.

4.3 Friction factor

The effect of the BVG insert on the isothermal pressure drop across the tested tube is depicted in Fig. 6. The variation of the pressure drop is shown in terms of *f* with *Re*. In the figure, it is apparent that the use of the BVG inserts leads to a substantial increase in the *f* over the smooth tube. As expected, the obtained *f* from the BVG at BR=0.2 is substantially higher than that at smaller BR. The mean increase in the *f* of the BVGs is in a range of 3.3 to 16.4 times over the smooth tube. This can be attributed to flow blockage, higher surface area and the act caused by the vortex flow.

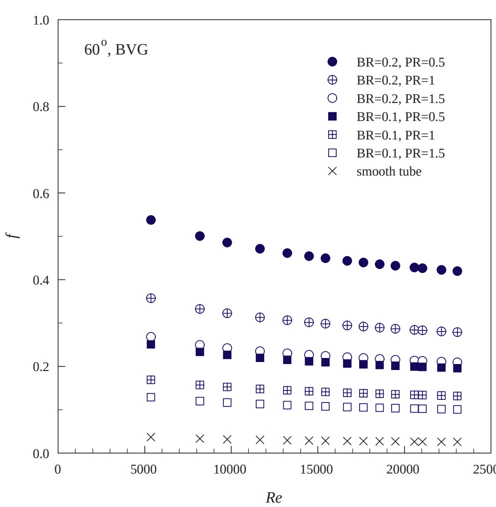


Fig. 6 Variation of *f* with *Re* for 60° BVGs.

The effect of the BVG insert on the isothermal friction factor ratio, f/f_0 with *Re* values is depicted in Fig. 7. In the figure, the f/f_0 value is found to be increased with the rise of *Re*. It is visible that the use of the BVG inserts leads to a substantial increase in f/f_0 with the rise of *Re*. As expected, the f/f_0 of the BR=0.2 is much higher than that of the BR=0.1 at a similar operating condition. For the BVG with PR=0.5, 1.0 and 1.5, the increases in f/f_0 values at BR=0.2 and 0.1 are, respectively, about 14.56–16.37, 9.67–10.87 and 7.26–8.15; and 6.79–7.63, 4.57–5.14 and 3.49–3.92 times above the smooth tube.

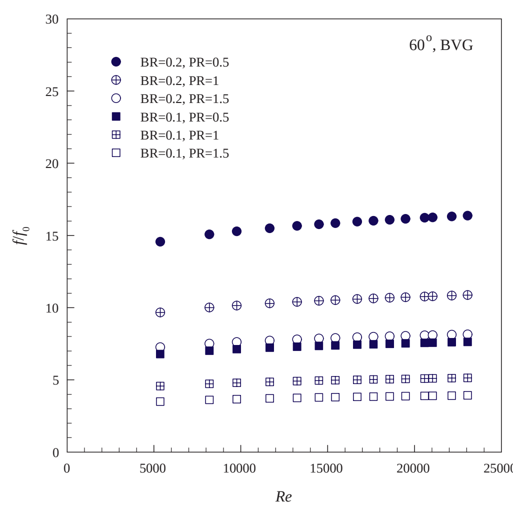


Fig. 7 Variation of f/f_0 with *Re*.

4.4 Thermal performance

Figure 8 shows the variation of the thermal enhancement factor, *TEF* with *Re* for all cases. The data obtained by *Nu* and *f* values are compared at a similar pumping power condition. In the figure, the *TEF* tends to reduce with the increase of *Re* for all. It is seen that the BVG inserted tube at BR=0.2 and PR=0.5 gives the highest *TEF* at lower *Re*. For the BR=0.2 BVGs at PR=0.5, 1.0 and 1.5, the maximum *TEF*s are, respectively, about 1.35, 1.29 and 1.31.

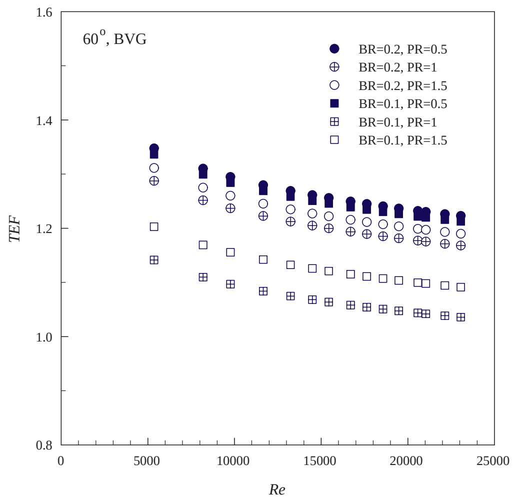


Fig. 8 Variation of *TEF* with *Re* for 60° BVGs.

5. Conclusions

An experimental work has been conducted to investigate airflow friction and heat transfer characteristics in a round tube inserted with the 60°-inclined BVGs at various BRs and PRs for turbulent regime, *Re* from 5000 to 24,000. The summary results can be drawn as follows:

1. The presence of the BVGs at BR=0.2 causes a much high pressure drop increase, f/f_0=3.49–16.37 but also provides a considerable heat transfer augmentation in the tube, Nu/Nu_0=1.77-3.3.

2. The 60° BVG with larger BR gives higher heat transfer and friction loss than the one with smaller BR but the smaller PR performs better than the larger.

3. The 60° BVG at BR=0.2, PR=0.5 yields the highest *TEF* of about 1.35 at lower *Re*.

6. References

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