

EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER USING INCLINED VORTEX RINGS IN A CIRCULAR TUBE WITH VARYING INCLINATION

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Abstract: *The influence of inclined vortex rings (VR) on heat transfer augmentation in a uniform heat-fluxed tube has been investigated experimentally. In the present work, the inclined elliptical vortex rings were mounted repeatedly in the tube with various angle ($25^{\circ}, 30^{\circ}, 35^{\circ}$) and geometry parameters of the VR, three relative ring width ratios ($BR = b/D = 0.1, 0.15$ and 0.2) and at constant relative ring pitch ratios ($PR = P/D = 0.5$). Air was employed as the test fluid in the tube for the Reynolds number from 12000 to 24,500. The aim at using the VRs is to create counter-rotating vortices inside the tube to help increase the turbulence intensity as well as to convey the colder fluid from the core regime to the heated-wall region. The experimental results show a significant effect of the presence of the VRs on the heat transfer and pressure loss over the smooth tube. The larger BR value provides higher heat transfer and pressure loss than the smaller. The Nusselt number is maximum for vortex ring angle 35° , but due to the large increase in the friction factor, thermal performance decreases. However, the VR at $BR = 0.1$ and $PR = 0.5$ with 30° yields the best thermal performance.*

Keywords: Heat transfer, Inclined vortex ring, Tube Turbulators, Nusselt number.

I. INTRODUCTION

Heat transfer augmentation techniques are needed in heat exchanger systems to enhance heat transfer rate and improve their thermal performance. The techniques for heat transfer augmentation can be divided into two main groups. One is the active method requiring extra external power sources such as mechanical aids, surface-fluid vibration, injection and suction of the fluid, jet impingement, and electrostatic fields. The other is the passive method that requires no external power for the systems. In general, the passive method is more popular and it includes a surface coating, wavy surfaces, rough and extended surfaces, convoluted (twisted) tube, additives for liquid and gases, turbulators (coiled wire and conical ring) and vortex/swirl generators. Most inserted devices mentioned above that form an important group of the passive augmentation technique, are frequently used to generate vortex/swirl flow in the thermal system. Insertion of vortex/swirl generators in a circular tube is a simple technique for enhancing the convective heat transfer coefficient on the tube side of a heat exchanger due to their advantages of easy fabrication and operation as well as low maintenance. In addition, the performance of those turbulators or vortex generators strongly depends on their

geometries. There are many types of vortex generators employed in the heat exchanger ducts/tubes such as helical/twisted tapes [1-3], coiled wires [4-6], circular/twisted-rings [7,8] and angle-finned tapes [9,10]. Yakut et al. [11, 12] reported the effect of conical-ring turbulators on the heat transfer, pressure drop, flow-induced vibration and vortices. In addition, the thermal performance for employing the turbulators at constant pumping and entropy generation was evaluated. From the evaluation of entropy generation, the conical-ring turbulators showed the merit as energy saving device only at low Reynolds number since low pressure drop was generated in that flow region. Promvonge [13] studied the insertion effect of the conical ring arrangements, namely, converging conical ring, diverging conical ring, and converging-diverging conical ring on the heat transfer rate, friction factor and thermal performance in a round tube. The study showed that the diverging conical ring offered higher thermal performance than the converging and converging-diverging ones. Durmus [14] investigated the effect of angle arrangement of the conical-ring type turbulators on the heat transfer and friction loss. The investigated results revealed that heat transfer rate as well as friction factor increased with the increase in the conical-ring angle. Promvonge and Eiamsaard [15] examined the combined effect of the conical-ring and twisted-tape for heat transfer enhancement in a circular tube. As reported, the use of the conical-ring in common with the twisted-tape provided an average heat transfer rate up to 10% over that of the conical-ring alone. Taslim et al. [16] conducted a measurement of the heat transfer in a V-ribbed square channel with three relative rib height ratios ($e/D = 0.083, 0.125$ and 0.167) at a fixed $P/e = 10$ using a liquid crystal technique. Various staggered rib configurations were studied, especially for the angle of 45° and experimental data showed a significant increase in average Nusselt number with increasing the e/D value. Chandra et al. [17] studied the heat transfer behaviors in a square channel with continuous ribs on four walls where ribs were placed superimposed on walls. They found that the heat transfer increases with the increment in the number of ribbed walls and with reducing Reynolds number while the friction factor increases with both cases. Apart from experimental investigations, the numerical studies on heat transfer enhancement by means of the circular ring turbulators were also reported [18,19]. Ozceyhan et al. [18] numerically studied effect of space between the circular rings on heat transfer rate and friction factor. Similarly, Akansu [19] numerically investigated the

effect of pitch spacing of porous rings and demonstrated that the heat transfer rate and friction factor increased with decrement of the ring spacing. Kwankaomeng and Promvonge [20], and Promvonge et al. [21] studied numerically the laminar periodic flow over 30° and 45° angled baffles repeatedly mounted only on one wall of a square channel, respectively. They noted that the heat transfer enhancement for the 45° angled baffle with $BR = 0.4$ was about 2-3 fold higher than that for the 90° baffle while the friction loss was some 10-25% lower. In addition, they found that a single stream wise main vortex flow created by the angled baffles/fins can help to induce impingement flows on the upper, lower and baffle trailing end walls of the channel. The appearance of the vortex induced impingement (VI) led to drastic increase in the thermal performance of the channel. In comparison, the 30° baffle/fin performs better than the 45° one due to lower pressure losses. From the literature review cited above, the use of wire coil/twisted tape inserts in the tube may not be suitable due to lower strength of the vortex flow, leading to thermal performance in a range of 0.7 to 1.1. Promvonge et al. [22, 23] again investigated numerically the laminar flow structure and thermal behaviours in a square channel with 30° and 45° inline baffles on two opposite walls. Two stream wise counter rotating vortex flows were created along the channel and VI jets appeared on the upper, lower and baffle leading end walls while the maximum thermal performance was found for the 30° inline baffle case. The works in Refs. [20-25] triggered the present work to investigate the heat transfer enhancement in a tube inserted with 30° inclined VRs. In a square duct with wall roughened by repeated angled-baffles (or fins), the baffle-induced secondary flows (or vortex flows) accompanied by enhanced turbulence intensity provide a drastic increase in heat transfer due to the VI effect as reported in Refs. [21, 22, 24]. In the present work, another type of inserted devices has been developed by insertion of 30° inclined VRs into a circular tube. The present inserted device has been developed from a combination of the merits of baffle, winglet and twisted tape tabulators.

This means that the inserted device will provide a drastically high heat transfer rate like baffles, swirl/vortex flow as winglets and ease of practical use like twisted tapes. Therefore, a newly 30° inclined VR insert is proposed and expected to provide higher thermal performance than the wire coil/twisted tape inserts due to the VI effect. The experimental results using air as the test fluid for the VR inserted tube are presented for turbulent flows in a Reynolds number range of 12000 to 24,500 in the current study.

II. EXPERIMENTAL SETUP

The experimental work was conducted in an open-loop experimental facility as shown in Fig. 1. The loop consisted of a blower, orifice meter to measure the flow rate and the heat transfer test tube with VR insert. The mild steel test pipe has a length of $L = 600$ mm, with 50 mm inner diameter (D), 56 mm outer diameter (D_o), and 3 mm thickness (t) as depicted in Fig. 2. The inclined VRs made of a 3mm asbestos elliptical ring given in Fig. 2 were connected to each other by insertion of two small straight steel rods into the hole on both sides of the VRs. The VRs with ($BR = b/D = 0.2$) and pitch ratios ($PR = P/D = 0.5$) were inserted into the tube by wall-attached position. The tube was heated by nichrome wire, band type 400W electrical heater 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.

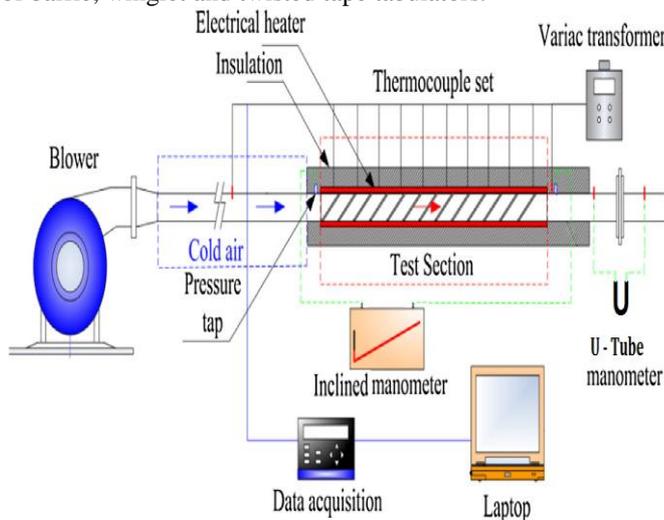


Fig.1 Experimental setup block diagram

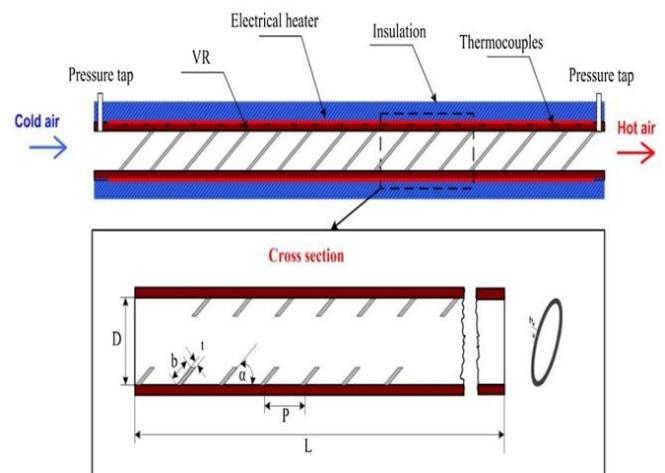


Fig.2: Test section with inclined VRs

The outer surface of the test tube was well insulated to minimize convective 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 were tapped equally along the local top and side walls of the outer tube surface. The inlet air from a high pressure blower was directed through the orifice meter and passed to the tested section fitted repeatedly with 25° , 30° and 35° inclined VRs. The air flow rate was measured by an orifice meter, built according to ASME standard [26]. Manometric fluid was used in a U-tube manometer with specific gravity (SG) of 0.61 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 flow control valve. During the experiments, the bulk air was heated by an adjustable electrical band heater wrapping along the test section.



Fig.3: Actual view of VRs

III. DATA COLLECTION AND ANALYSIS

A. Heat Transfer Calculations:

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

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

The heat supplied by electrical heater in the test tube is found to be 3-5% 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

$$Q_{conv} = hA (T_s - T_b) \quad (3)$$

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

$$T_s = \sum T_s / 5 \quad (5)$$

Where for a constant heat flux, the average surface temperature T_s can be calculated from 5 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:

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

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

The Reynolds number is given by

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

Friction factor, f , can be written as

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

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:

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

IV. RESULT AND DISCUSSION

A. Verification for 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 and friction factor. The Nusselt number and friction factor obtained from the present smooth tube are, respectively, compared with the correlations of Dittus Boelter for Nusselt number, and of Petukhov for friction factor found in the open literature [28] for turbulent flow in ducts.

Correlation of Dittus Boelter:

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

Correlation of Petukhov,

$$f = (0.79 \ln Re - 1.64)^{-2} \quad (12)$$

Fig. 4(a) & (b) shows a comparison of the Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (11) and (12). In the figure, the present results agree very well within $\pm 10\%$ with the published correlations.

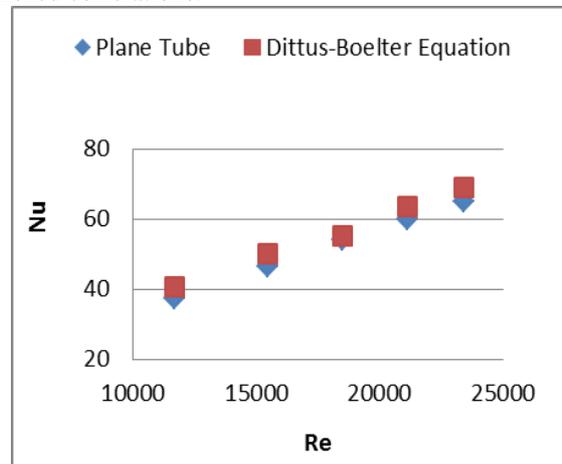


Fig.4 (a): Validation results for Nusselt number

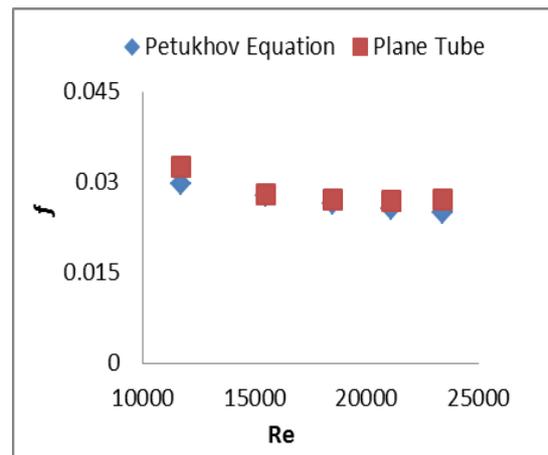


Fig.4 (b): Validation results for friction factor

B. Effect of angle of inclination

The experimental results on heat transfer and friction characteristics in a uniform heat-fluxed tube fitted with VRs at three relative angle 25° , 30° and 35° and relative ring width or blockage ratios ($BR = 0.2, 0.15, 0.1$) are presented

in the form of Nusselt number and friction factor as displayed in Fig. 4a and b, respectively. In Fig. 4a, it is found that the Nusselt number is higher at an angle 35° , apart from the Re values.

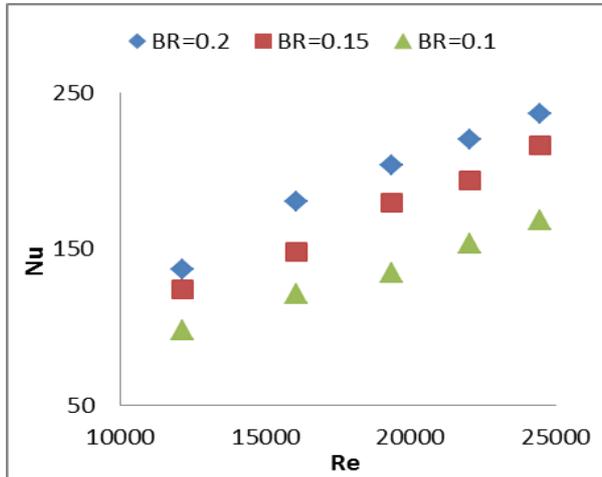


Fig.5 (a): Variation of Nusselt Number for 300 VRs at different BR.

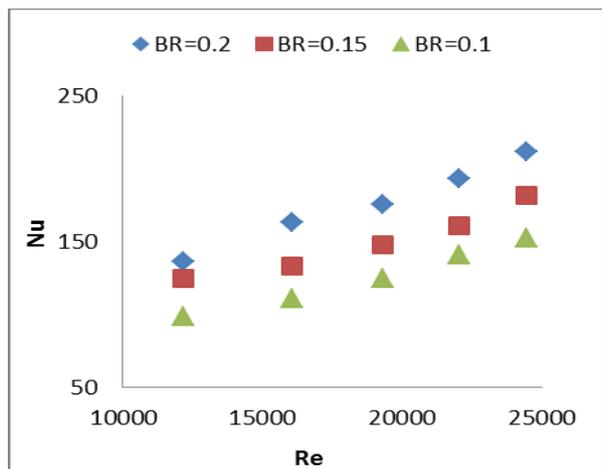


Fig.5 (b): Variation of Nusselt Number for 250 VRs at different BR.

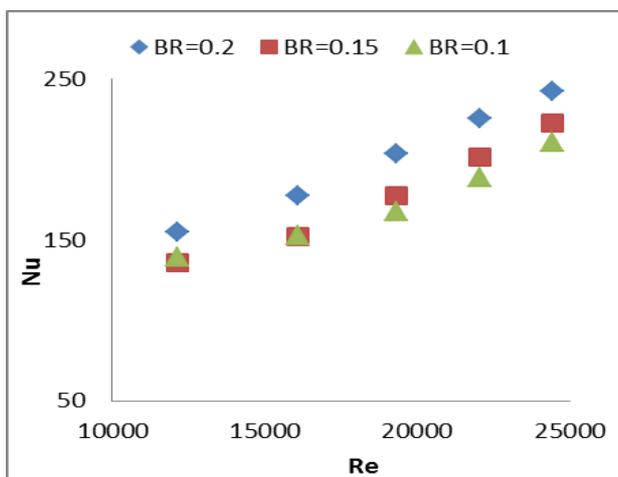


Fig.5 (c): Variation of Nusselt Number for 350 VRs at different BR.

The variation of friction factor with Reynolds number for different angle is presented in Fig.6(a),(b) and (c). It is observed that friction factor gradually reduced with rise in Reynolds number. It is observed to be maximum for VR angle 35° and minimum for 25° VRs.

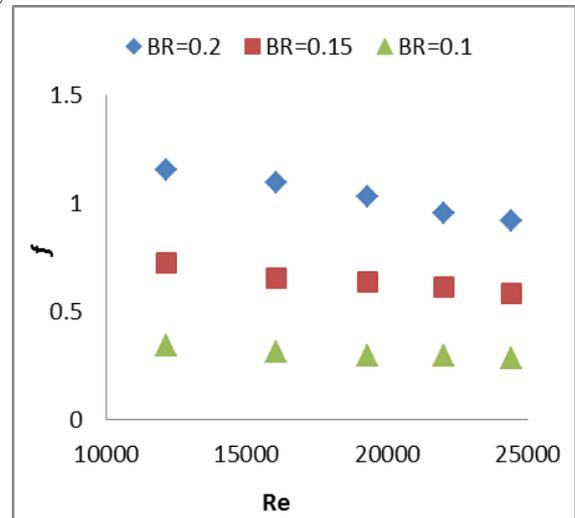


Fig.6 (a): Variation friction factor for 300 VRs at different BR

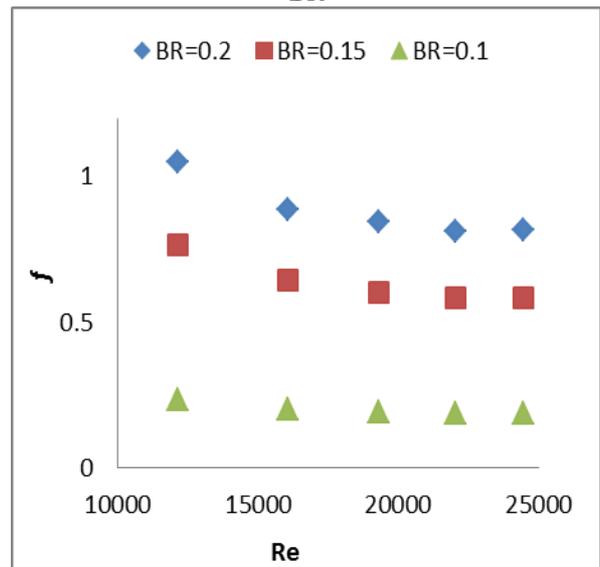


Fig.6 (b): Variation of friction factor for 250 VRs at different BR

C. Effect of Blockage ratio (BR)

The variation in Nusselt number & friction factor is shown in figure 5 (a), (b), (c) and 6 (a), (b), (c). There is increase in Nusselt number and friction factor with increase in blockage ratio for all inclination. Nu and f are maximum for 35° VRs at BR=0.2. This is because the large BR causing flow circulation and separation provides high turbulence intensity of the flow than smaller one. The Nusselt number values for 35° vortex rings angle at PR = 0.5 and for the BR=0.2, 0.15 and 0.1 are in the range of 155-242, 136-222 and 139-210. Similarly for 30° and 25° Nu values are in the range of 137-236, 124-216, 97-168 and 136-211, 124-181, 98-152 respectively. The effect of VR insert on the pressure drop

across the tested tube is depicted in fig. 6(a), (b) & (c). From the figure it is apparent that the use of the VR inserts leads to a substantial increase in friction factor over smooth tube. As expected friction factor for BR=0.2 gives higher values in all angles of inclination. Friction factor ranges from 0.425-0.481, 0.847-1.04 and 1.26-1.52 for 35°, 0.281-0.345, 0.584-0.723 and 0.927-1.157 for 30° and 0.185-0.235, 0.583-0.765 and 0.819-1.049 for 25° VRs inclination with increasing Reynolds numbers.

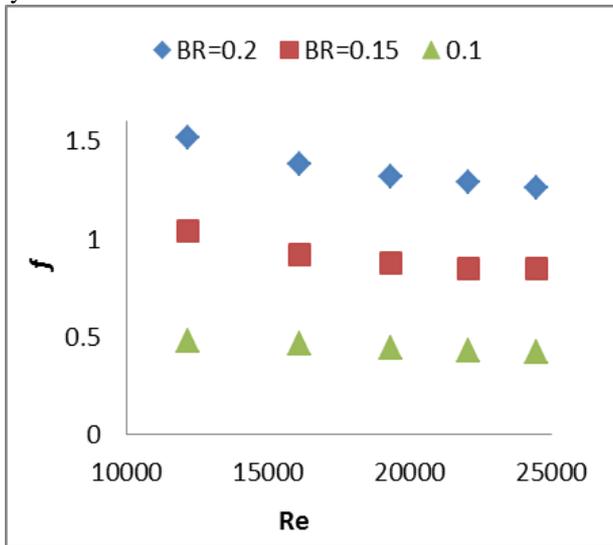


Fig.6 (c): Variation of friction factor for 35° VRs at different BR

D. Performance evaluation:

Fig 7(a), (b) and (c) shows the variation of the thermal performance enhancement factor (TEF) with Reynolds number. For all, the data of Nusselt number and friction factor ratios obtained from the correlations are compared at an identical pumping power condition. In the figure, the TEF found to be above unity tends to reduce with the increase in Re for all the cases. Nusselt number for 35° VRs is highest but it also increases the friction factor proportionally and hence decreasing TEF.

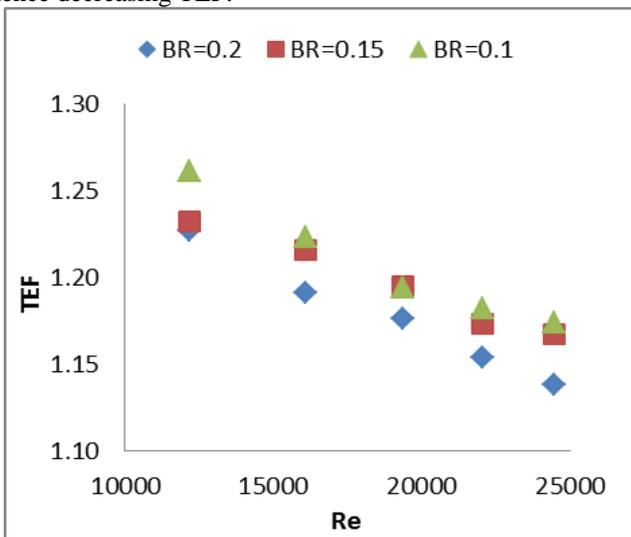


Fig.7 (a): Variation TEF for 30° VRs at different BR

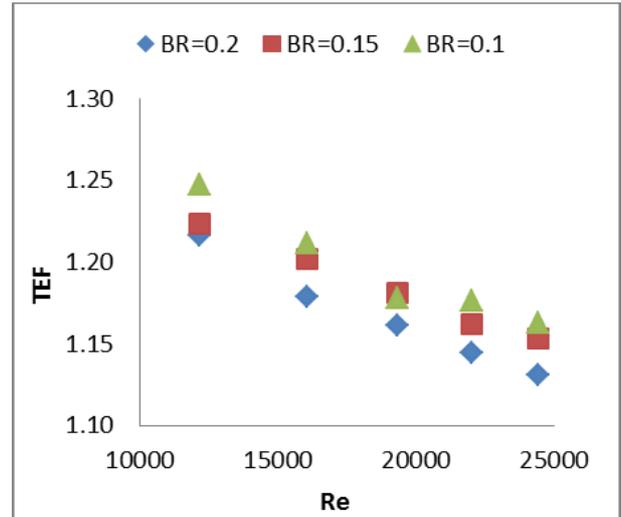


Fig.7 (b): Variation of TEF for 25° VRs at different BR

The maximum TEF of the VRs for 30°, 35° and 25° are 1.261, 1.247 and 1.184 for BR=0.1 at constant PR=0.5 with low Re values.

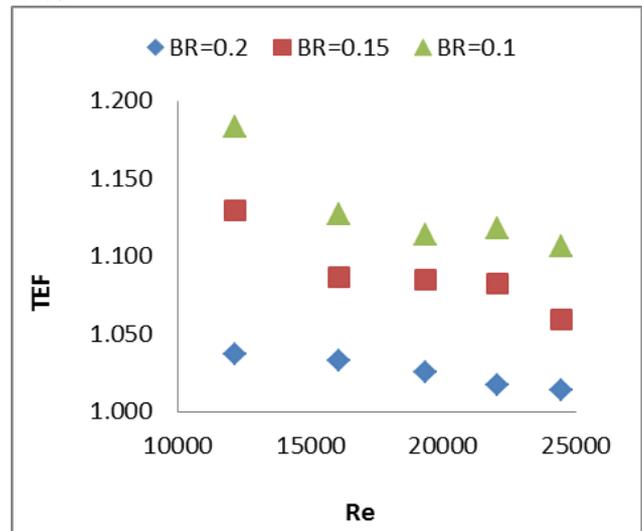


Fig.7(c): Variation of TEF for 35° VRs at different BR

It is seen that the VR 30° inserted tube at BR 0.1 and PR = 0.5 gives the highest TEF at lower Re.

V. CONCLUSIONS

An experimental study has been conducted to examine the heat transfer and flow friction characteristics in a round tube inserted with the inclined VRs at different angles (25°, 30° and 35°) different BRs with constant PR (0.5) for the turbulent flow, Reynolds number from 12000 to 24,500. The presence of the VRs at BR = 0.2, PR = 0.5 causes a much high pressure drop increase, but also provides a considerable heat transfer augmentation in the tube. The Nusselt number of the VR inserted tube shows an increasing trend with the rise in BR and Re values. The TEFs of the inserted tube are all above unity and tend to decrease with the increment in Re and BR values. The 30° VR at BR= 0.1 yields the highest TEF of about 1.26 at lower Reynolds number.

VI. NOMENCLATURE

A	heat transfer surface area, m ²
b	ring width, m
BR	relative ring width or blockage ratio = b/D, e/D
C _{p,air}	specific heat capacity of air, kJ /kg K
D	diameter, m
e	rib or baffle height, m
f	friction factor
h	mean heat transfer coefficient, W/ m ² /K
I	current, A
k	thermal conductivity of air, W/m K
L	length of test tube, m
m	mass flow rate, kg/s
Nu	Nusselt number
Q	heat transfer rate, W
P	pitch length of VR, m
DP	pressure drop, Pa
PR	relative ring pitch or pitch ratio =P/D
Pr	Prandtl number
Re	Reynolds number
T	temperature, K
TEF	thermal performance enhancement factor
t	thickness of test tube, m
U	mean axial velocity, m/ s
V	voltage, V
VI	vortex-induced impingement
VR	vortex ring

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