Oral Presentation

The 7th TSME International Conference on Mechanical Engineering
13-16 December 2016

ETM0008

Turbulent heat transfer and pressure loss in a square-duct heat exchanger with inclined-baffle inserts

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Abstract

The paper deals with an experimental investigation on heat transfer and friction loss characteristics in a square-duct heat exchanger with inclined-baffle inserts. Air as the working fluid enters the test duct having a uniform surface heat-flux. The inclined baffles were placed on both sides of a rectangular centre-cleared tape/frame before diagonally inserting the baffled frame into the test duct to produce longitudinal vortex flows through the test section. Effects of five different baffle-to-duct height or blockage ratios (b/H=BR=0.1, 0.2, 0.3, 0.4 and 0.5) on heat transfer, pressure loss and thermal performance in the square duct are examined for Reynolds number ranging from 4100 to 25,600. The baffle-pitch to duct-height or pitch ratio (P/H=PR) and an attack angle (α) of the baffle were fixed at 3.0 and 30°, respectively. The experimental results reveal that the heat transfer and pressure loss in the form of respective Nusselt number (Nu) and friction factor (f) from the baffle tend to increase with the rise of Reynolds number (Re) and BR. The maximum enhancement in Nu and f has been found to be 4.61 and 63.67 times above the smooth duct, respectively. The thermal enhancement factor (η) is maximum at BR=0.3.

***Keywords**:* Inclined baffle, Square duct, Thermal performance, Reynolds number, Vortex generator.

1. Introduction

Heat exchangers are widely used in industrial applications such as, steam generation, chemical processing, electric power generation, etc. Rib/baffle-roughened surface are extensively applied for heat transfer augmentation (HTA) in heating or cooling passages. The HTA by baffles relies on induced vortex (or high swirl) in the flow of fluid. The vortex generators (VGs) are inserted into the flow field to provide an interruption of boundary layer development and to cause enhancement of the rate of heat transfer by increasing turbulence intensity. Several researchers have reported the effects of tube insert with swirl or vortex flow devices on the HTA, pressure loss and thermal performance. Sivashanmugam and Suresh [1] examined the turbulent heat transfer and pressure loss characteristics in a circular tube fitted with full-length helical screw elements having different twist ratio, and increasing and decreasing order of twist ratio. Kreith and Margolis [2] proposed that the heat transfer rate can be enhanced by introducing swirl flow in the heat exchanger with tangential injection of the fluid at various locations along the tube. Chang et al. [3] investigated the heat transfer and pressure drop characteristics in a tube fitted with serrated twisted tape. In their work, the serrations on two edges of the twisted tape with twist ratios of 1.56, 1.88, 2.81 and ∞ were the square-sectioned ribs with identical rib pitch and height. Eiamsa-ard et al. [4] studied the thermal and friction loss in a circular tube with two-typed regularly spaced twisted tape elements: (1) full-length typical twisted tape at different twist ratios, and (2) twisted tape with various free space ratios (S=1.0, 2.0, and 3.0). For compound swirl or vortex generators, Promvonge and Eiamsa-ard [5–7] investigated the effects of the conical-nozzle, conical-ring or V-nozzle together with a swirl generator (decaying swirl) on thermal and flow friction characteristics in a uniform heat-fluxed tube and found that for using both enhancement devices, the increase in heat transfer rate is about 20–50% higher than the use of a single enhancement device but also substantial rise in friction loss.

The use of ribs/baffles creating the high strength of vortex flow leads to a considerable increase in heat transfer rate in channels/ducts. Ko and Anand [8] examined experimentally the turbulent channel flow through porous baffles and found that the porous baffles provided flow and thermal behaviors as good as the solid baffles. Laminar flow behaviors in a channel fitted with 90° transverse baffles mounted on two opposite walls with staggered array were studied by Berner et al. [9] who found that the flow is free of vortex shedding at a Reynolds number below 600. Gupta et al. [10] reported the local heat transfer distributions in a square channel with continuous, profiled and broken ribs. They indicated that the heat transfer from the V-broken ribs was higher than that of the continuous and profiled ribs. Promvonge et al. [11, 12] experimentally and numerically studied thermal behaviors in turbulent square-duct flow through 30° angle-finned tapes and found that the finned tape with smallest fin pitch ratio (PR=1) yielded the highest Nusselt number, friction factor and the thermal enhancement factor.

Most of the investigations, cited above, have focused on thermal performance characteristics for thin fins/ribs (called ‘‘baffle’’) placed on the duct wall and the baffles mounted on the typical (solid) tape with diagonal insert. The study on turbulent flows through inclined baffles mounted on a rectangular centre-cleared tape/frame (punched hole of solid-tape) has rarely been reported. This type of the baffles is very fruitful for reducing friction loss of heat exchanger systems. Thus, the main aim of the present work is to extend the experimental data available on diagonally inclined-baffle inserts. Experimental results using air as the working fluid for five different baffle heights are presented in turbulent flows ranging from Re = 4100 to 25,600.

2. Experimental setup and test procedure

2.1. Details of inclined baffle

The schematic view of inclined baffles chosen for study is shown in Fig. 1. The baffle was made of aluminum strip with 0.5 mm thickness (t) while the centre-cleared tape used for supporting the baffles was made of aluminum sheet with 0.5 mm thickness (δ), 1000 mm length (L) and 45 mm width (H), the straight tape was centrally cut to become the rectangular centre-cleared tape with 0.2H width (w) of each tape edge. The five baffle sizes were 4.5, 9.0, 13.5, 18.0 and 22.5 mm high (b), equivalent to five baffle-to-duct height or blockage ratios, b/H=BR= 0.1, 0.2, 0.3, 0.4 and 0.5, respectively, and were placed on both tape edges at three times of duct height, P/H=PR= 3 and at attack angle (α) of 30°. The range of baffle parameters and test conditions is given in Table 1.

2.2. Experimental procedure

The schematic representation of experimental facility and test section of square-duct heat exchanger is shown in Fig. 2. The main components of the experimental setup were a high-pressure blower, aluminum square-duct, electric heater, orifice plate, circular pipe, settling tank, AC power supply, inclined manometer, ammeter, voltmeter, thermocouples. The test duct made of 3 mm thick aluminum sheets having a cross section of 45 mm × 45 mm (hydraulic diameter, D_h = 45 mm) was 1000 mm long (L). The test duct used in the present experiment was composed of the four heating walls (top, bottom and sides). A uniform heat flux of 3000 W/m^2 (maximum) was maintained using AC power supply (variable transformer). Air at room temperature (25 °C) entered the system by a high-pressure blower. A circular pipe was connected between a high-pressure blower and a settling tank while an orifice plate was placed in this pipeline. Ten values of volumetric flow rates were maintained to cover the entire range of Reynolds number (4100–25,600). The temperature distributions along the test duct were measured by 28 K-type thermocouples fitted into the outer duct walls. The inlet and outlet bulk temperatures were measured using two K-type thermocouples located at the entry and the exit of the test duct. All thermocouples were read by a data acquisition system (Fluke 2680A) and recorded by a laptop. More details of the method and uncertainty analysis are similar as reported in earlier paper [13].

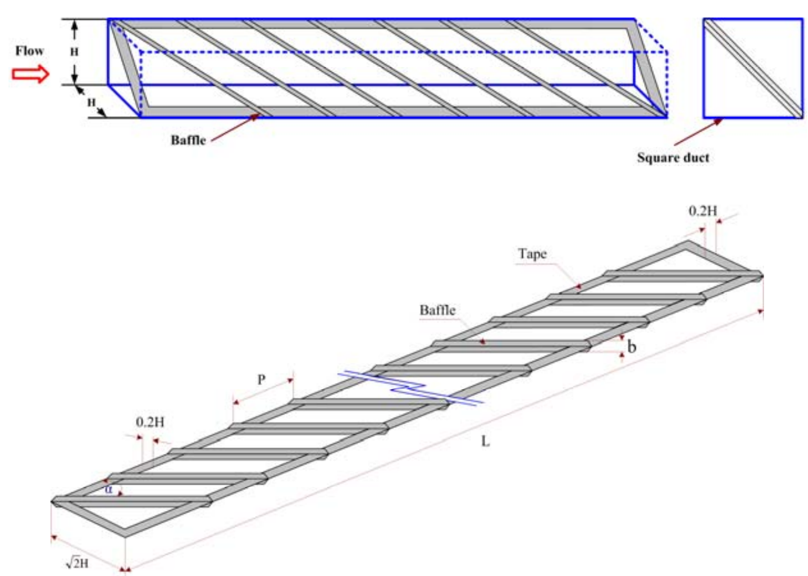

Fig. 1. Test section with 30° inclined baffle inserts.

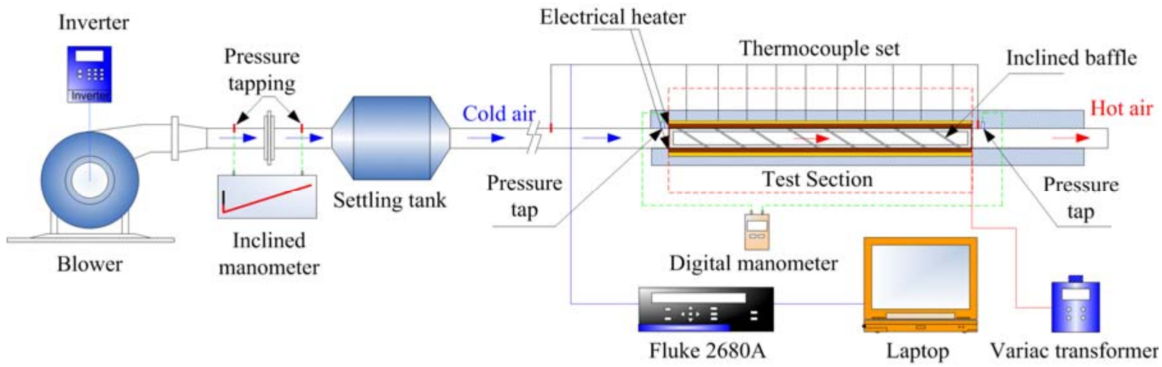

Fig. 2. Schematic diagram of experimental setup.

Table. 1 Details on baffle geometry and test conditions

Working fluid	air
Reynolds number, Re	4100 to 25,600
Duct high = Duct wide, H	45 mm
Hydraulic diameter, D_h	45 mm
Test section length, L	1000 mm
Baffle attack angle (α)	30° (fixed)
Blockage ratio, BR	0.1, 0.2, 0.3, 0.4, 0.5
Pitch ratio, PR	3 (fixed)
Baffle thickness, t	0.5 mm
Tape length	1000 mm
Tape thickness, δ	0.5 mm

3. Data analysis

In the present study, air used as the test fluid is flowed through a uniform heat-fluxed square-duct heat exchanger. The steady state of the heat transfer rate is assumed to be equal to the heat loss in the test section. The average heat transfer coefficients are evaluated from the measured temperatures and heat inputs. With heat added uniformly to fluid (Q_{air}) and the temperature difference of wall and fluid (T_w-T_b), the average heat transfer coefficient is evaluated from experimental data via the following equations:

$$Q_{air} = Q_{conv} = \dot{m} \cdot C_p \cdot (T_o - T_i) = V \cdot I \tag{1}$$

$$h = Q_{conv}/(A \cdot (\tilde{T}_w - T_b)) \tag{2}$$

in which,

$$T_b = (T_o + T_i)/2 \tag{3}$$

and

$$\tilde{T}_w = \sum T_w/28 \tag{4}$$

The term A is the convective heat transfer area of the heat transfer surface, T_b is bulk temperature of the fluid in the test duct and $\tilde{T}_w$ is the average surface (wall) temperature along the axial length of the heated duct. Then, average Nusselt number (Nu) is written as:

$$\mathrm{Nu} = (h \cdot D)/k \tag{5}$$

The Reynolds number (Re) based on the duct hydraulic diameter (D) is given by

$$\mathrm{Re} = U \cdot D/\nu \tag{6}$$

The friction factor (f) is evaluated by:

$$f = \frac{2}{(L/D)}\frac{\Delta p}{\rho U^2} \tag{7}$$

The thermal enhancement factor, η, defined as the ratio of the heat transfer coefficient of vortex generator insert, h to that of a smooth duct, h_0, at an equal pumping power [14] is given by

$$\eta = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right|_{pp} = \left(\frac{\mathrm{Nu}}{\mathrm{Nu}_0}\right)\left(\frac{f_0}{f}\right)^{1/3} \tag{8}$$

4. Results and discussion

4.1 Verification of smooth square-duct

The present experimental results on fluid flow and heat transfer characteristics in a smooth wall square duct are first validated in terms of Nusselt number (Nu) and friction factor (f). The result received from smooth duct is compared with results obtained from standard equation of Nu and f. The Nu and f obtained from the present smooth square-duct are compared with the correlations of Gnielinski and Petukhov found in the literature [15] for turbulent flow in ducts.

Correlation of Gnielinski,

$$\mathrm{Nu} = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{0.5}(\mathrm{Pr}^{2/3}-1)} \tag{9}$$

Correlation of Petukhov,

$$f = (0.79\ln\mathrm{Re} - 1.64)^{-2} \tag{10}$$

The experimental and correlation values were compared and shown in Fig. 3a and b, respectively. Fig. 3a shows a comparison of Nu and f obtained from the present work with those from correlations of Eqs. (9) and (10). The average deviation was observed for Nu is around 5.1% and for f is 5.6% which shows an excellent agreement.

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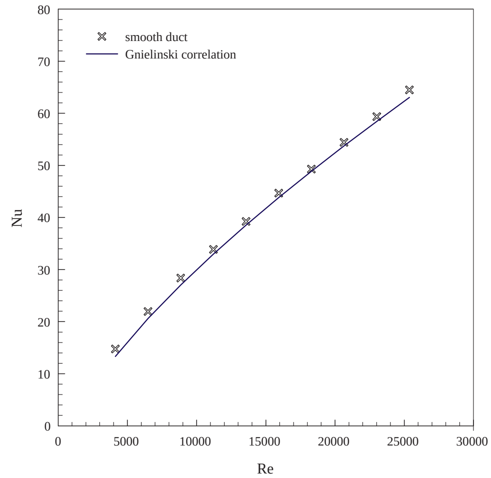
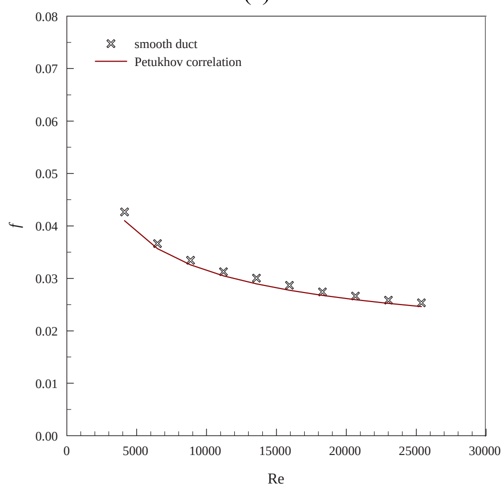


(a)



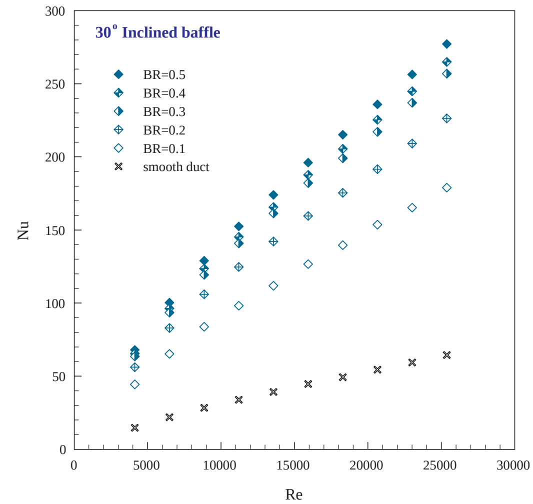
(b)

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

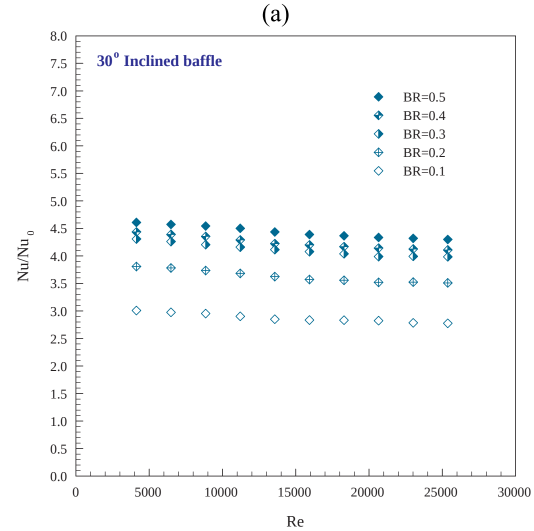
4.2 Heat transfer results

An experimental study has been carried out to study the augmentation of heat transfer by 30° diagonally inclined baffles inserted in a square-duct heat exchanger. The relationship between Nu and Re of the baffled duct is demonstrated in Fig. 4a. In the figure, the values of Nu are found to increase with the rise of Re and BR. The heat transfer for the baffles is found to be considerably higher than that for the smooth duct with no insert. It is worth noting that the baffle with largest blockage ratio (BR=0.5) yields the highest Nu. In the present work, the Nu values for BR=0.1, 0.2, 0.3, 0.4 and 0.5 are, respectively, up to 64.2%, 72.4%, 75.7%, 76.4% and 77.5% above the smooth duct.

A ratio of the augmented Nu to the Nu of smooth duct in terms of Nusselt number ratio (Nu/Nu_0) plotted against BR is displayed in Fig. 4b. It is seen that Nu increases with the increase in BR and is a maximum at BR=0.5. This is caused by stronger vortex-flow strength of BR=0.5 interrupting the thermal boundary layer, resulting in heat transfer enhancement by increasing turbulence intensity or fast fluid mixing.



(a)



(b)

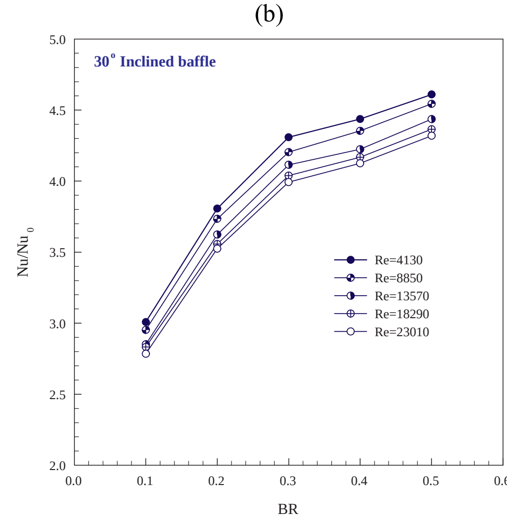


Fig. 4. (a) Variation of Nu with Re; (b) Variation of Nu/Nu_0 with Re; (c) Variation of Nu/Nu_0 with BR.

It is found in Fig. 4c that the Nu/Nu_0 shows a steeper increase at the beginning with the rise of BR. The Nu/Nu_0 tends to upward with the increment of BR from 4130 to 23,010 and its increasing trend for BR≤0.3 is steeper than that for BR>0.3. The Nu/Nu_0 values for the baffles at BR=0.5, 0.4, 0.3 0.2 and 0.1 are, respectively, about 4.3–4.6, 4.1–4.4, 3.9–4.3, 3.5–3.8 and 2.8–3.0. The average Nu/Nu_0 for BR=0.5 are about 4.4%, 7.3%, 18.2% and 35.2% higher than that for BR=0.4, 0.3, 0.2 and 0.1, respectively.

4.3 Friction factor results

The effects of Re on friction factor (f) and friction factor ratio (f/f_0) for different BR values are depicted in Fig. 5a and b, respectively, while the variation of f/f_0 with BR is shown in Fig. 5c. In Fig. 5a, it is observed that use of the baffles leads to a substantial increase in f over the smooth duct. The mean increases in f for BR=0.5 are about 41.1%, 57.6%, 64.4% and 74.6% above those for BR=0.4, 0.3, 0.2 and 0.1, respectively. The insertion of the baffles creates more obstruction in the flow. This can be attributed to blocking flow of the baffles, larger surface area and the act caused by the reverse flow. As expected, the f/f_0 of the largest BR (BR=0.5) is the maximum as shown in Fig. 5b.

Fig. 5c shows the variation of f/f_0 with BR for different Re values. In the figure, the f/f_0 is found to increase with the increment in Re and BR. Among five BRs, the BR= 0.5 gives the highest f/f_0. The average f/f_0 values for BR=0.5, 0.4, 0.3, 0.2 and 0.1 are about 54.0, 31.7, 22.8, 19.1 and 13.7 times, respectively.

4.4 Thermal performance results

The relationship between thermal enhancement factor (η) at the same pumping power and Re is presented in Fig. 6. In the figure, η tends to reduce with the rise of Re for all BRs. Determining from Eq. (8), the η of the inclined baffle at BR=0.3 is highest around 1.68 at the lowest Re. The maximum TEFs are, respectively, about 1.40, 1.57, 1.68, 1.55 and 1.36 for BR=0.1, 0.2, 0.3, 0.4 and 0.5. The BR=0.3 yields the TEF around 6–7%, 8–9%, 16–17% and 18–19% higher than the BR=0.2, 0.4, 0.1 and 0.5, respectively. This suggests that the 30° inclined baffle with BR=0.3 should be used to obtain higher thermal performance.

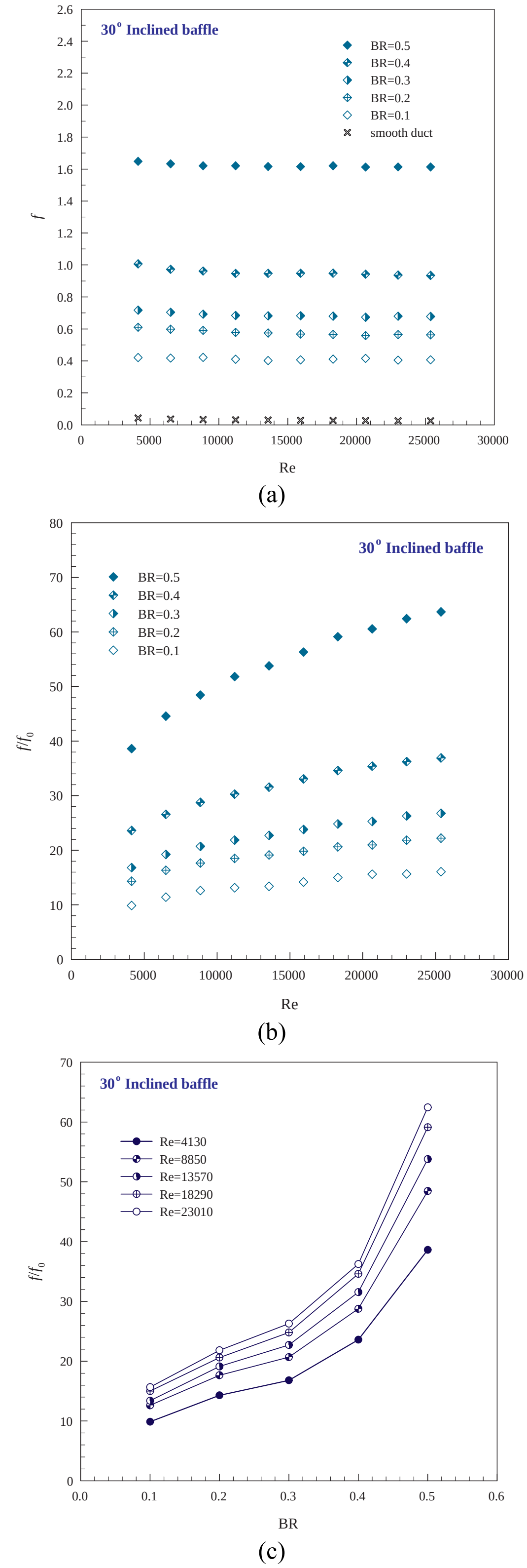


Fig. 5. (a) Variation of f with Re; (b) Variation of f/f_0 with Re; (c) Variation of f/f_0 with BR.

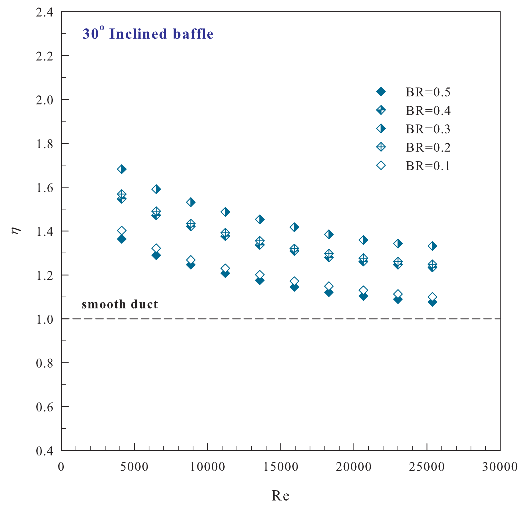


Fig. 6. Variation of η with Re for various BRs.

5. Conclusions

The effects of the inclined-baffles with different blockage ratios (b/H=BR=0.1, 0.2, 0.3, 0.4 and 0.5) on the heat transfer, friction factor and thermal performance behaviors have been investigated experimentally for Re from 4100 to 25,600. The findings from the results can be concluded as:

1. The Nu/Nu_0 tends to be nearly independent with the rise of Re.
2. The increase in BR and Re provides higher the rate of heat transfer and friction loss due to fast fluid mixing.
3. The presence of the baffle at BR=0.5 offers the maximum Nu of around 77.5% and f of 63.67 times over the smooth duct.
4. The η shows the downtrend with increasing Re. The highest η is around 1.40, 1.57, 1.68, 1.55 and 1.36 for BR=0.1, 0.2, 0.3, 0.4 and 0.5, respectively.
5. The BR=0.3 is more attractive in heat transfer application due to higher η than the others.

6. References

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