TSF025

Effects of Inclined Baffles on Thermal Characteristics in a Square duct

Panuwat Hoonpong[1,*], Narongsak Arkamanont[2], Supattarachai Suwannapan[3], Sirisawat Juengjaroennirachon[1], and Pongjet Promvonge[3,*]

[1]Department of Mechanical Technology, Faculty of Industrial Technology, Thepsatri Rajabhat University, 321 Naraimaharat Road, Talaychubsorn, Lopburi, Thailand, 15000.

[2] Department of Mechanical and Automotive Engineering technology, Faculty of Engineering and Technology, King Mongkut's University of Technology North Bangkok (Rayong Campus), 19 Moo 11 Nonglalok, Bankhai, Rayong, Thailand, 21120.

[3] Department of Mechanical Engineering, Faculty of Engineering, King Mongkut's Institute of Technology Ladkrabang, Bangkok, Thailand, 10520.

*Corresponding Author: sompop2525@gmail.com, kppongje@kmitl.ac.th

Abstract

This research work presents a study of heat transfer and flow friction characteristics in a square duct heat exchanger with 30^o inclined baffles diagonally inserts. The tested duct has a square cross-section with a constant wall heat flux condition. The experiments are conducted by varying airflow rate for Reynolds number ranging from 4000 to 25,000. The inclined baffles with three blockage ratios (BR = e/H = 0.1, 0.15, 0.2), single baffle pitch ratio (PR = P/H = 0.5) and the attack angle, α = 30^o. The main aim of this work is to investigate the effects of the above parameters on the heat transfer and flow friction characteristics were shown in form of Nusselt number and friction factor respectively. The experimental data was presented for turbulent duct flows over the inclined baffled and expected to create vortex flows throughout the tested duct for the higher mixing flow between core and wall regimes leading to greater heat transfer rate and lower pressure drop. The experimental result shows that BR = 0.2 performs the highest heat transfer and friction factor. For all cases of the inclined baffles yield higher thermal performance than the smooth duct.

***Keywords**:* Inclined baffle, Nussult number, Reynolds number, Thermal performance

1. Introduction

Turbulators have been applied in high-performance thermal systems and they completely result in the change of the flow field and hence the variation of the local convective heat transfer coefficient. Several investigations have been conducted to study the effect of turbulators on heat transfer and pressure loss in the heat exchanger ducts such as ribs/fins/baffles [1-3], winglets [4], coiled wires [5] and helical/twisted tapes [6,7]. Most turbulators such as coil wire/helical tape/twisted tape are effectively applied to circular tubes while the ribs/baffles/fins and winglets are suitably employed for the channels or flat surface ducts. In the present work, the investigation on heat transfer and friction behaviors for turbulent flow over the inclined baffle turbulators placed on the central core has been conducted. The main aim of this work is to study the changes in the flow pattern and heat transfer performance in the duct for all cases of 30^o inclined baffles diagonally insert. The experimental results are performed for turbulent flow in the range of Renolds number from 4000 to 26000

2. Experimental Setup

A schematic diagram of the experimental apparatus is presented in Figure 1. In this figure, there is the square cross-section test duct made of 3 mm thick aluminum plates,each wall side of 45 mm (H) and a length of 1000 mm (L). The AC power supply was the source of power for the plate-type heater, used for heating all walls of the test section in order to maintain a uniform surface heat-flux. The external layer of the test duct was wrapped with high temperature insulation in order to reduce heat loss to environment. Air as the tested fluid was directed into the systems by a 1.45 kW high-pressure blower through the orifice meter to measure the flow rate and then the tested duct. The pressure drop across the orifice was measured using inclined manometer to indicate the airflow rate. In order to measure temperature distributions on the upper, lower and side walls, twenty-eight K-type thermocouples were fitted to walls, equally. To measure the inlet and outlet bulk temperatures, two thermocouples were positioned upstream and downstream of the test duct. The thermocouple voltage outputs were fed into a data acquisition system and then recorded via a personal computer. The pressure drop across the test section was measured by a digital manometer.

The detail of inclined baffle inserted into the square-duct heat exchanger is depicted in Figure 2. While the inclined baffles have four blockage ratios (BR=e/H=0.1, 0.15, 0.2 and 0.3) with attack angles of 30° (α=30°) and PR=0.50.

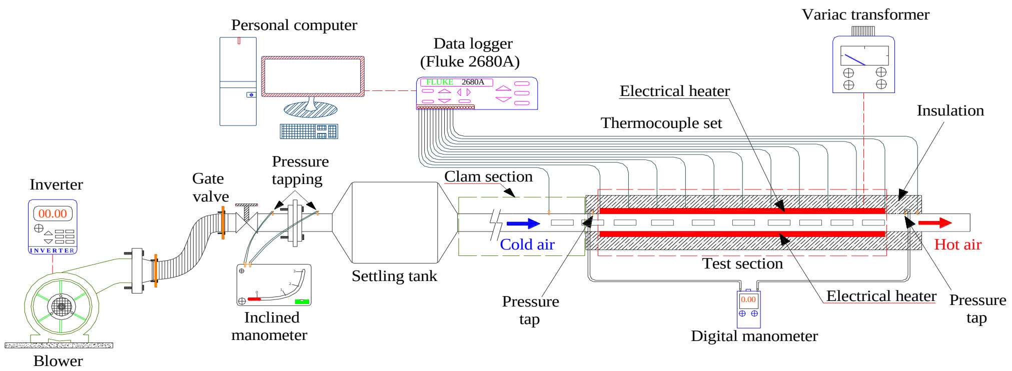

Fig. 1 Schematic diagram of experimental apparatus.

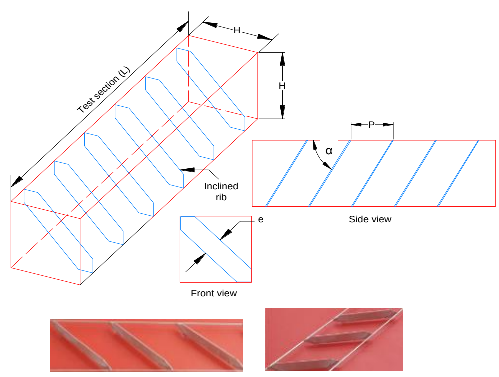

Fig. 2 The detail of inclined baffles

3. Data Reduction

The goal of this study Reynolds number based on the duct hydraulic diameter is given by

$$\mathrm{Re} = \mathrm{UD}_h / \nu \quad (1)$$

In experimental investigation, the heat transfer rate at steady state condition is supposed to be are evaluated from the measured temperatures and heat inputs, with added heat to fluid (Q_{air}) uniformly and the temperature difference between average surface temperature and fluid bulk temperature ($\tilde{T}_s - T_b$). The average heat transfer coefficient will be evaluated by using the following equations:

$$Q_{air} = Q_{conv} = \dot{m}C_p(T_o - T_i)$$

$$h = Q_{conv} / \left(A(\tilde{T}_s - T_b)\right) \quad (2)$$

where

$$T_b = (T_o + T_i)/2 \text{ and } \tilde{T}_s = \sum T_s / 28 \quad (3)$$

The term A is the convective heat transfer area of the heated wall whereas $\tilde{T}_s$ is the average surface temperature. Then, average Nusselt number based on the duct hydraulic diameter is written as:

$$\mathrm{Nu} = \mathrm{hD_h}/\mathrm{k} \quad (4)$$

The friction factor *f* is evaluated by:

$$f = 2\Delta P / \left((L/D_h)\rho U^2\right) \quad (5)$$

where ΔP is the pressure drop across the test section and U is the mean air velocity in the duct. The thermal enhancement factor (TEF) defined as the ratio of h of an augmented surface to that of a smooth surface, h_0, at a constant pumping power, Ref. [9], is written as

$$\mathrm{TEF} = h/h_0\big|_{pp} = \mathrm{Nu}/\mathrm{Nu}_0\big|_{pp} = (\mathrm{Nu}/\mathrm{Nu}_0)(f/f_0)^{-1/3} \quad (6)$$

4. Results and Discussion

4.1 Verification of smooth duct

The present experimental result shows the Nusselt number and friction factor obtained from the present smooth duct are compared with the correlations of Gnielinski and Petukhov [10].

Correlation of Gnielinski,

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\mathrm{Pr}}{1+12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3}-1)} \quad (7)$$

Correlation of Petukhov

$$f = (0.79\ln \mathrm{Re} - 1.64)^{-2} \quad (8)$$

Figure 3 shows a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (7) and (8). In the figure, the present results agree very well within ±2.8% and ±3.2%.

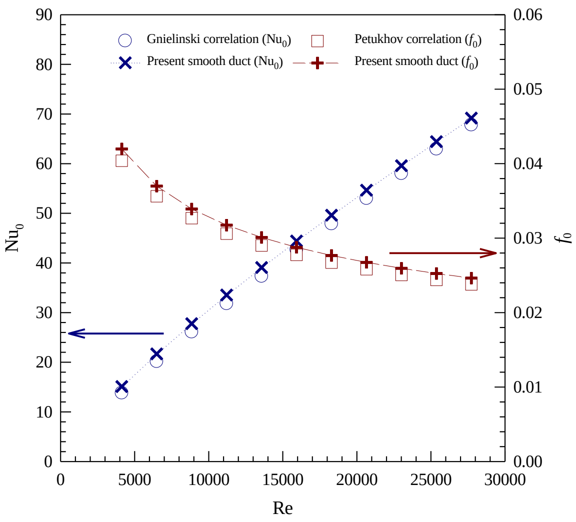

Figure 3. Validation of (a) *Nu* and (b) *f* for smooth duct.

4.2 Effect of BR on heat transfer rate

The experimental results on heat transfer and friction characteristics in a uniform heat-fluxed square duct with 30° inclined baffles diagonally insert are illustrated in the form of Nusselt number and friction factor. Nusselt numbers obtained under turbulent flow conditions and are presented in Fig. 4. In the figure it's shown that the Nu value increases with the rise of Re meanwhile with the reducing of BR. While the Nu of the baffle inserted duct is performed to be higher than of the smooth duct without baffles insert. This is because the inclined baffles interrupt the development of thermal boundary layer thickness of the flow and help to increase the turbulence degree of flow. It is worth noting that the Nusselt number for the inclined baffles with BR = 0.2 is higher than those with BR = 0.15, 0.1, and smooth duct, respectively.

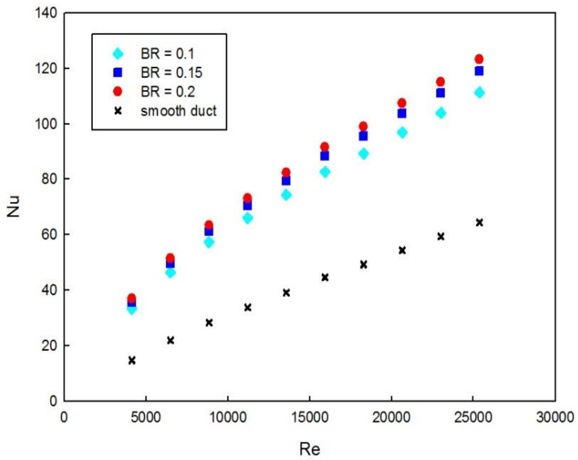

Figure 4. Variation of Nu with Re

The effects of various baffle parameters on heat transfer is presented in the form of Nusselt number ratio, Nu/Nu_0, defined as a ratio of the augmented Nu to Nu of smooth duct (Nu_0). Figure 5 displays the variation of Nu/Nu_0 with Re. In the figure, it can be observed that the baffles provide a considerable Nu/Nu_0 increase with reducing BR and the figure shows that Nu values considerably decrease with the rise of Re. The baffles with BR=0.2 gives the maximum heat transfer rate because of highly interrupting the flow and promoting higher level of vortex strength. The baffles with BR= 0.2, 0.15 and 0.1 yields the Nu/Nu0 in the range of 1.91-2.51, 1.85-2.42, 1.73-2.27 times, respectively. A close examination reveals that the inclined baffle BR=0.2, produces the highest heat transfer rate than the others.

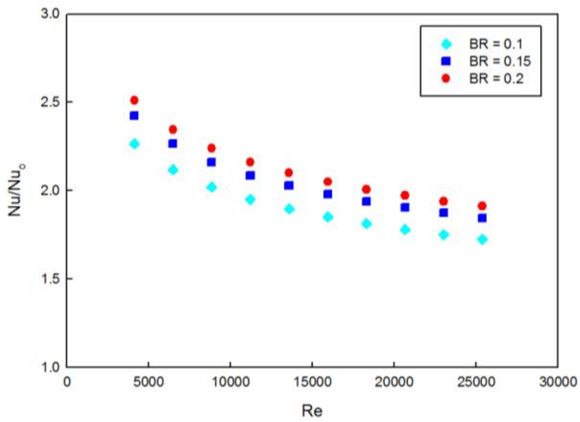

Figure 5. Variation of Nu/Nu_0 with Re

4.3 Effect of BR on flow friction

The effect of using 30° inclined baffles on the isothermal pressure loss across the tested section is shown in fig. 6. The variation of the pressure loss is shown in the form of friction factor with Reynolds number. The figure performed that the use of inclined baffles directs to considerable increase in friction factor over the smooth duct. The increasing in Re caused the declining of f value. As forecasted, friction factor values of 30° inclined baffle with BR = 0.2 are significantly higher than those with BR = 0.15 0.2 and smooth duct. For the baffle with BR = 0.2, the increase of friction factor is in the range of 120-150% above one with BR = 0.15 and 0.2 and of 450% over smooth duct.

The variation of the friction factor ratio, f/f_0, plotted against the Re is depicted in Figure 7. In the figure, it is visible that the f/f_0 tends to increase with the increment of Re. The baffles at BR=0.2 provides the highest f/f_0 because of higher flow blockage and larger surface area. The ranges of f/f_0 values are around 4.31-5.40, 3.54-4.42, and 2.81-3.52 – folds, for the inclined baffles with BR=0.2, 0.15 and 0.1, respectively.

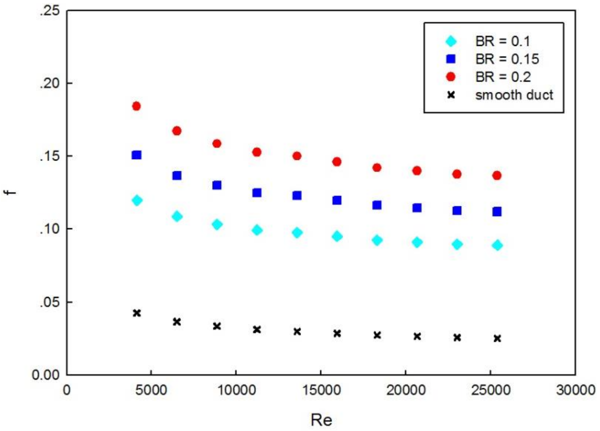

Figure 6. Variation of *f* with Re

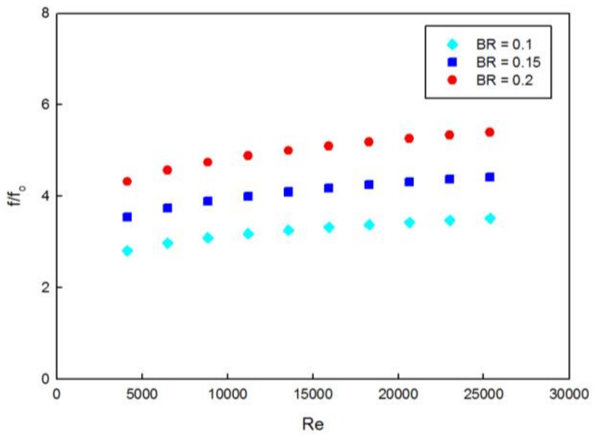

Figure 7. Variation of f/f_0 with Re

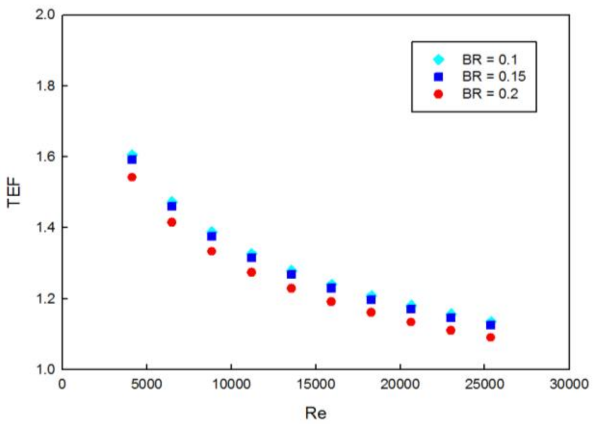

Figure 8. Variation of TEF with Re.

4.4 Effect of BR on thermal enhancement factor

Figure 8 shows the variation of the thermal enhancement factor (TEF) with Re for various BR. All of the data of Nu/Nu_0 and f/f_0 are compared at a similar pumping power. In the figure, the TEF tends to decline with the rise of Re. It is seen that the inclined baffles with BR=0.10 gives the highest TEF for all cases at a similar attack angle and Reynolds number. The maximum TEF of the present work is found at BR=0.10 and lower Re because at this point yields high value of Nu/Nu_0 while shows the lowest of f/f_0 value thus, provides the highest TEF of about 1.61 at the lowest Re.

5. Conclusions

An experimental investigation has been carried out to examine airflow friction and heat transfer characteristics in a uniform heat-fluxed square-duct fitted with baffle turbulators for the turbulent regime, Re ranging from 4000-25,000. The use of 30° inclined baffles can induce vortex flows through out the tested duct and provides better flow mixing than the smooth duct. The inclined baffle turbulators yield a considerable increase in pressure drop and heat transfer over the smooth duct. The heat transfer rate and the flow friction increase with the increment of BR. The baffle with BR= 0.2 gives the highest heat transfer rate and pressure loss, respectively. In comparison, uses of inclined baffle leads to the higher heat transfer rate but the baffle with BR=0.10 performed the highest thermal enhancement factor due to the lower pressure loss.

6. References

[1] Promvonge, P., Skullong, S., Kwankaomeng, S. and Thiangpong, C. (2012). Heat transfer in square duct fitted diagonally with angle-finned tape–Part 1: Experimental study, Int. Commun. Heat and Mass Transfer. pp. 617-624.

[2] Chandra, P.R., Alexander, C.R. and Han, J.C. (2003). Heat transfer and friction behaviour in rectangular channels with varying number of ribbed walls, Int. J. Heat Mass Transfer. vol.46, pp. 481–495.

[3] Han, J.C., Zhang, Y.M. and Lee, C.P. (1991). Augmented heat transfer in square channels with parallel, crossed and V-shaped angled ribs, ASME. Heat Transfer. vol, 113. pp. 590–596.

[4] Chompookham, T., Thianpong, C., Kwankaomeng, S. and Promvonge, P. (2010). Heat transfer augmentation in a wedge-ribbed channel using winglet vortex generators, Int. Commun. in Heat and Mass Transfer. vol, 37 pp. 163–169.

[5] Promvonge, P. (2008). Thermal performance in circular tube fitted with coiled square wires, Energy Convers. Manage. vol. 49, pp. 980–987.

[6] Eiamsa-ard, S., Thianponge, C., Eiamsa-ard, P. and Promvonge, P. (2009). Convective heat transfer in a circular tube with short-length twisted tape insert, Int. Commun. in Heat and Mass Transfer. vol. 36, pp. 365–371.

[7] Eiamsa-ard, S. and Promvonge, P. (2007). Heat transfer characteristics in a tube fitted with helical screw-tape with/without core-rod inserts, Int. Commun. in Heat and Mass Transfer. vol. 34, pp. 176–185.

[8] ANSI/ASME, Measurement Uncertainty, PTC 19. 1–1985, Part I, USA (1986).

[9] Webb, R.L. (1992). Principles of Enhanced Heat Transfer, John Wiley & Sons, New York.

[10] Incropera, F.P., Dewitt, P.D., Bergman, T.L. and Lavine, A.S., (2006). Introduction to heat transfer, fifth ed., John Wiley & Sons.