Thermal Characteristics in Square Duct Fitted Diagonally with Angle-ribbed Tapes

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Abstract

The paper presents an experimental study on turbulent flow and heat transfer characteristics in a square duct fitted diagonally with 45° angle-ribbed tapes. The tested duct has a square section and uniform heat-fluxed walls and the flow rate of air used as the test fluid is presented in terms of Reynolds number from 4000 to 25,000. The angle-ribbed straight tape in the present work is newly invented without previous investigations available. The insertion of the ribbed tape is performed with a rib-pitch to duct-height ratio, (P/H=3) at the rib attack angle of 45° with respect to the main flow direction. The ribbed tape inserted diagonally in the duct is expected to generate a longitudinal vortex flow pair through the heated duct. Influences of five rib-to-duct height ratios (e/H=0.1-0.3) on the heat transfer in terms of Nusselt number and the pressure loss in the form of friction factor are experimentally investigated. The experimental result indicates that the rib with e/H=0.3 provides the highest heat transfer and friction factor values but the one with e/H=0.2 provides the highest thermal enhancement factor at lower Reynolds number.

***Keywords**:* thermal characteristics; angle-ribbed tape; square duct; inserted duct; turbulator

1. Introduction

Vortex/swirl flows have been commonly used for increasing convection heat transfer coefficients in several engineering applications such as heat exchangers, drying processes and vortex combustors. There are many types of vortex generators employed in the heat exchanger ducts such as helical/twisted tapes [1], coiled wires [2], ribs/fins/baffles [3]. Most vortex flow devices mentioned above are effectively applied to circular tubes while the rib/baffle/fin and winglets are suitably employed for the channels or flat surface ducts. In high performance heat exchanger duct systems, periodic ribs/baffles/fins have been widely applied in many industrial applications. The baffled/finned duct successfully prevents the development of thermal boundary layer, and therefore augments the heat transfer performance and results in much better heat transfer efficiency than that in smooth duct with no baffle/fin. Because of the practical importance, the heat transfer and flow characteristic in ducts with rib/baffle/fin turbulators have attracted many investigators.

Han et al. [4,5] investigated experimentally the heat transfer in a square channel with ribs on two walls for nine different rib configurations. Average heat transfer and friction factor were reported for the P/e=10 and e/H=0.0625 rib by heating either only one of the ribbed walls or both of them, or all four channel walls. The heat transfer augmentations and the friction factor were highest for the 60° orientation amongst the angled ribs. Lopez et al. [6] studied numerically of laminar forced convection in a baffled channel with a uniform heat flux on the top and bottom walls for periodically fully developed flow was conducted. Promvonge et al. [7,8] investigated numerically the laminar flow structure and thermal behaviors in a square channel with 30° and 45° inline baffles on two opposite walls. Two streamwise counter-rotating vortex flows were created along the channel and VI jets appeared on the upper, lower and baffle leading end side walls while the maximum thermal performance factor was found for the 30° inline baffle case but the 45° inline baffle provides higher heat transfer rate. Promvonge et al. [9,10] again studied experimentally and numerically the turbulent square-duct flow over the 30° angle-finned tapes inserted diagonally. They found that at smaller fin pitch spacing, the finned tape with e/H=0.2 and P/H=1.0 gives the best thermal performance. The thermal performance of the newly invented finned tape was reported to be much higher than that of the wire coil/twisted tape turbulators.

In the present work, the inserted device has been applied by insertion of a 45° angle-ribbed tape into a square duct. The diagonally inserted tape in the square duct is expected to induce two counter-rotating vortices throughout the duct. 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 will provide a drastically high heat transfer rate like baffles, low pressure drop like angled ribs, swirl/vortex flow as winglets and ease in practical use like twisted tapes. Therefore, the insertion of a newly invented 45°-ribbed tape is proposed in the present study. The experimental results using air as the test fluid for the 45° angle-ribbed tape inserted diagonally in the square duct are presented for turbulent region in a *Re* range of 4000 to 25,000.

2. Experimental Setup

A schematic diagram of the experimental apparatus is presented in Fig. 1 while the detail of a tape with double-sided angled-ribs, inserted diagonally into the square duct is shown in Fig. 2. In Fig. 1, a circular tube used for connecting a high-pressure blower to a settling tank was attached by an orifice flow meter to measure the flow rate while a square duct including a calm section (2000 mm) and a test section (1000 mm) was employed after the settling tank. The duct configuration was characterized by the duct height, H of 45 mm while a single rib axial pitch equal to three times of duct height (pitch ratio, P/H=3) and the rib attack angle of 45° were introduced. The tested square duct made of 3 mm thick aluminum sheets has a cross section of 45×45 mm^2 and 1000 mm length (L). The diagonal straight tape was made of aluminum sheet with its dimension of 63×1000×0.5 mm^3. The five rib strip sizes were 4.5, 6.75, 9, 11.25 and 13.5 mm high (b) with 0.3 mm thickness (t). The angled ribs made of a 0.3 mm aluminum strip were attached on the two sides of the aluminum tape with hot superglue. The test section consisted of the four heating walls. 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.

Air as the test fluid in both heat transfer and pressure drop experiments, was directed into the systems by a 1.45 kW high-pressure blower. The operating speed of the blower was varied by using an inverter to provide desired airflow rates. The flow rate of air in the systems was measured by using an orifice plate system pre-calibrated by using hot wire and vane-type anemometers. The pressure drop across the orifice was measured using an inclined manometer. In order to measure temperature distributions on the upper and lower wall and a sidewall, thirty thermocouples were fitted into the outer duct walls. The thermocouples were installed in holes drilled from the rear face and centered of the duct walls with the respective junctions positioned within 1.5 mm of the inside wall and axial separation was 100 mm apart. To measure the inlet and outlet bulk temperatures, two sets of two thermocouples were positioned upstream and downstream of the test duct. All thermocouples were type-K, 1.5 mm diameter wire. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2650A) and then recorded via a computer.

The uncertainty in the data calculation was based on Ref. [11]. The maximum uncertainties of non-dimensional parameters were ±5% for Reynolds number, ±6% for Nusselt number and ±8% for friction. The uncertainty in the axial velocity measurement was estimated to be less than ±5%, and pressure has a corresponding estimated uncertainty of ±5%, whereas the uncertainty in temperature measurement at the duct wall was about ±0.5%.

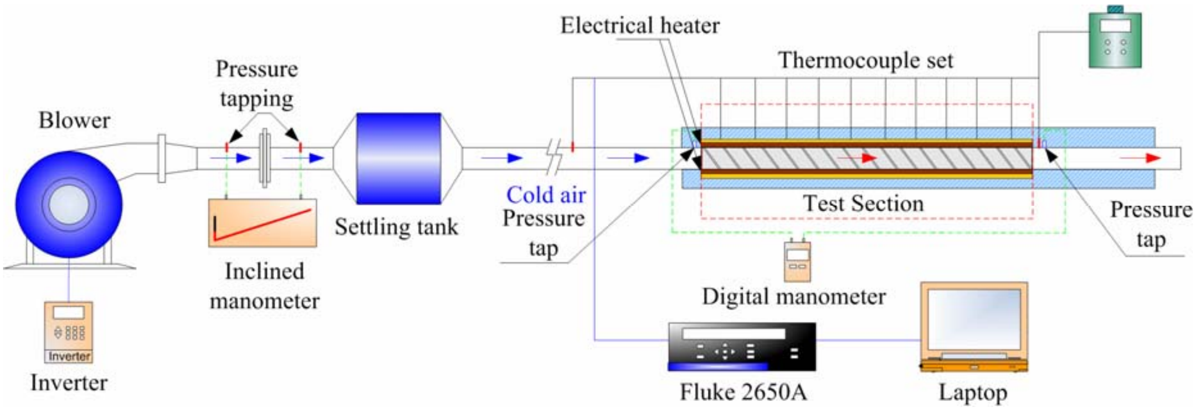

Fig. 1 Schematic diagram of experimental apparatus.

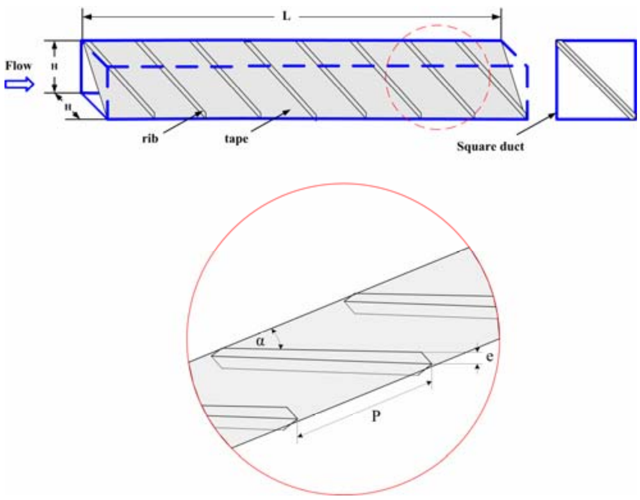

Fig. 2 Test section with 45° angle-ribbed tape inserts.

Table 1. Details on rib tape geometry and tested conditions

Working fluid	Air
Reynolds number	4000 to 25,000
e/H	0.1, 0.15, 0.2, 0.25, 0.3
PR	3
Tape thickness	0.5 mm
Tape length	1000 mm

3. Data Processing

The Reynolds number based on the duct hydraulic diameter is given by:

$$Re = UD_h / \nu \quad (1)$$

The main purpose of the experiment is to investigate the heat transfer and flow friction behaviors in a square duct inserted diagonally with an angle-finned tape. The average heat transfer coefficients are evaluated from the local measured temperatures and heat inputs. With heat added uniformly to fluid (Q_{air}) and the temperature difference of wall and fluid ($T_w - T_b$), average heat transfer coefficient will be calculated from the measured data via the following equations:

$$Q_{air} = Q_{conv} = \dot{m}C_p\left(T_o - T_i\right) \quad (2)$$

$$h = \frac{Q_{conv}}{A\left(\tilde{T}_s - T_b\right)} \quad (3)$$

in which,

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

and

$$\tilde{T}_s = \sum T_s / 30 \quad (5)$$

The term A is the inner surface area of the four heated duct walls whereas $\tilde{T}_s$ is the average surface temperature obtained from local surface temperatures on the upper, lower and side walls along the axial length of the heated duct. Then, average Nusselt number is written as:

$$Nu = \frac{hD_h}{k} \quad (6)$$

The friction factor is evaluated by:

$$f = \frac{2}{(L/D_h)}\frac{\Delta P}{\rho U^2} \quad (7)$$

The thermal enhancement factor, η, defined as the ratio of the heat transfer coefficient of an inserted duct, h to that of a smooth duct, h_0, at an equal pumping power is given by:

$$\eta = \left.\frac{h}{h_0}\right|_{pp} = \left.\frac{Nu}{Nu_0}\right|_{pp} = \left(\frac{Nu}{Nu_0}\right)\left(\frac{f}{f_0}\right)^{-1/3} \quad (8)$$

4. Results and discussion

4.1 Validation of smooth duct

The present experimental results on heat transfer and friction characteristics in a smooth wall square duct are first validated in terms of Nusselt number and friction factor. The Nusselt number and friction factor obtained from the present smooth square duct are compared with the correlations of Gnielinski and Petukhov found in the literature [12] for turbulent flow in ducts.

Correlation of Gnielinski,

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

Correlation of Petukhov,

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

Figure 3 shows a comparison of Nusselt number and friction factor obtained from the present work with those from correlations of Eqs. (9) and (10). In the figure, the present smooth duct results are in excellent agreement within ±3% with the correlation data.

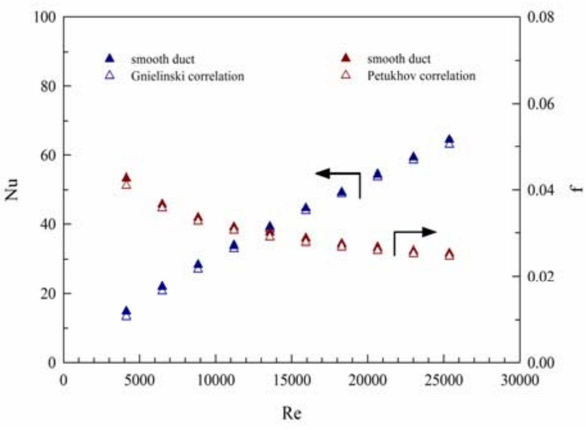

Fig. 3 Verification of Nusselt number and friction factor for smooth square duct.

4.2 Effect of blockage ratio (*e/H*)

The present experimental results on heat and flow friction characteristics in a uniform heat flux square duct diagonally inserted by a straight tape with 45° angled ribs attached repeatedly on the two tape sides are presented in terms of Nusselt number and friction factor. The Nusselt numbers for all cases investigated are presented in Fig. 4. In the figure, the angle-ribbed tape inserts yield considerable heat transfer enhancements with similar trends in comparison with the smooth duct and the Nusselt number increases with the increment of Reynolds number. This is because the angle-rib turbulators interrupt the development of thermal boundary layer of the fluid flow and increase the turbulence intensity of the flow. It is worth nothing that the heat transfer (*Nu*) for the ribs with *e/H*=0.3 is considerably higher than those with *e/H*=0.25, 0.2, 0.15 and 0.1. This is caused by higher blockage of using *e/H*=0.3 interrupting the flow and diverting its direction thus promoting high levels of mixing over others.

The effect of the ribbed tape insert on the isothermal pressure drop across the tested duct is depicted in Fig. 5. The variation of the pressure drop is shown in terms of friction factor (*f*) with Reynolds number. In the figure, it is apparent that the use of the rib turbulators leads to a substantial increase in *f* over the smooth duct. This can be attributed to flow blockage, higher surface area and the act caused by the vortex flow. As expected, the *f* of the ribs with *e/H*=0.3 is considerably higher than that of the ones with *e/H*=0.25, 0.2, 0.15 and 0.1. The losses mainly come from the dissipation of the dynamical pressure of the air due to high viscous losses near the wall, to higher friction of increasing surface area and the flow blockage because of the presence of the ribs.

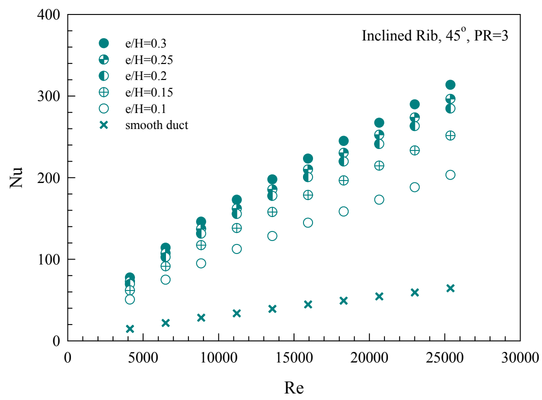

Fig. 4 Variation of Nusselt number with Reynolds number for various rib heights.

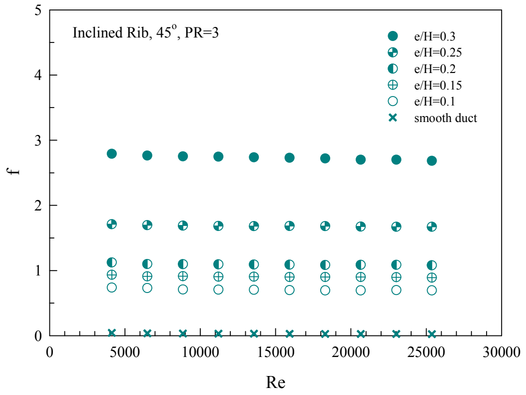

Fig. 5 Variation of friction factor with Reynolds number for various rib heights.

4.3 Performance evaluation

The Nusselt number ratio, Nu/Nu_0, defined as a ratio of augmented Nusselt number to Nusselt number of smooth duct, plotted against the Reynolds number value is displayed in Fig. 6. In the figure, the Nu/Nu_0 tends to slightly decrease with the rise of Reynolds number for all cases of the angle-ribbed tape insert. The mean Nu/Nu_0 values are found to be about 5.04, 4.74, 4.54, 4.04 and 3.27 times above the smooth duct for e/H=0.3, 0.25 0.2, 0.15 and 0.1, respectively.

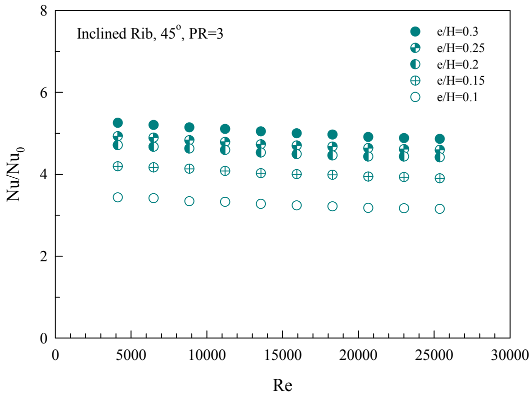

Fig. 6 Variation of Nusselt number ratio with Reynolds number.

The variation of isothermal friction factor ratio, f/f_0, with Reynolds number for five rib strip sizes is depicted in Fig. 7. In the figure, the f/f_0 value is found to be increased with the rise of Reynolds number. The mean f/f_0 values for using the ribs with e/H=0.3, 0.25, 0.2, 0.15 and 0.1 are around 90.82, 55.98, 36.34, 30.10 and 23.47 times above the smooth duct, respectively. This indicates that the use of lower e/H can help to reduce the pressure loss considerably.

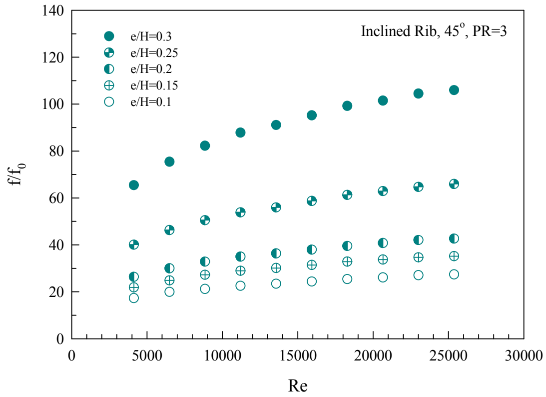

Fig. 7 Variation of friction factor ratio with Reynolds number.

Figure 8 shows the variation of the thermal enhancement factor, η with Reynolds number for all cases. For all, the data obtained by Nusselt number and friction factor values are compared at a similar pumping power condition. In the figure, the η tends to decrease with the increase of Reynolds number for all cases. It is seen that the ribs with e/H=0.2 provides the highest value of the mean thermal enhancement factor. The mean η values are around 1.38, 1.31, 1.25, 1.15 and 1.13 for the ribbed tapes with e/H=0.2, 0.15, 0.25, 0.1 and 0.3, respectively. The maximum thermal enhancement factor is found for using the ribbed tape with e/H=0.2.

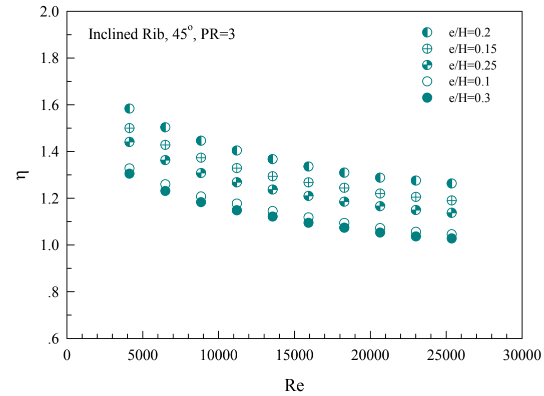

Fig. 8 Variation of thermal enhancement factor with Reynolds number.

5. Conclusions

An experimental work has been conducted to investigate airflow friction and heat transfer characteristics in a square duct fitted diagonally with 45°-ribbed tapes at different rib heights for the turbulent flow, Reynolds number of 4000 to 25,000. The summary results can be written as follows:

1. The presence of the angle-ribbed tape at *e*/*H*=0.3 causes a much high pressure drop increase, f/f_0=65-106 but also provides a considerable heat transfer augmentation in the duct, Nu/Nu_0=4.8-5.2.

2. The Nusselt number of the ribbed tape insert shows increasing trend with the rise in *e*/*H* and Reynolds number values.

3. The 45°-ribbed tape insert of *e*/*H*=0.2 yields the highest thermal enhancement factor of about 1.59 at lower Reynolds number.

6. Acknowledgment

The funding of this work obtained from the Thailand Research Fund (TRF) is gratefully acknowledged.

7. References

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