

Thermal Performance Enhancement in a Heat Exchanger Square-Duct with V-Shaped Fins

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Abstract

The article presents an experimental study on thermal performance enhancement in a constant heat-fluxed square-duct inserted diagonally with V-shaped fins vortex generator. The experiments were carried out by varying the airflow rate through the tested square-duct with V-shaped fins inserts for Reynolds number from 4000 to 25,000. The effect of the V-shaped fins with V-tip pointing downstream at various relative fin pitches on heat transfer and pressure drop characteristics was experimentally investigated. Both the heat transfer and pressure drop were presented in terms of Nusselt number and friction factor, respectively. Several V-shaped fins characteristics were introduced such as fin pitch to duct height ratios or pitch ratios, ($P/H=PR=0.5, 1.0, 1.5, 2.0, 2.5$ and 3.0), fin to duct height ratio or blockage ratio, ($e/H=BR=0.2$) and fin attack angle, ($\alpha=45^\circ$). The experimental results reveal that the heat transfer and friction factor values for the V-shaped fins inserts increase with the decrement of PR. The inserted square-duct for $BR=0.2$ at $PR=0.5$ provides the highest heat transfer and friction factor but the one at $PR=1.5$ yields the maximum thermal performance is about 1.45.

Keywords: thermal performance, heat exchanger, V-shaped fins, vortex generator, heat transfer

1. Introduction

Originally, heat exchangers were introduced using plain or smooth surface. Then, the improvement has developed using several heat transfer enhancement devices. In the past, many researchers have studied the effects of turbulator or swirl/vortex generator devices on heat transfer, pressure drop and thermal performance behaviors in pipe/duct heat exchanger such as Liu et al. [1] experimentally studied on heat transfer characteristics in steam-cooled rectangular channels with two opposite rib-roughened walls. They reported that the average Nusselt number for the channel with $\alpha=45^\circ$ was higher than $\alpha=60^\circ$. Tang and Zhu [2] numerically investigated the turbulent flow and heat transfer behavior in rectangular channel with inclined broken ribs. They reported that the V-type ribbed array gave the best thermal performance factor. Shui et al. [3] conducted experimental and numerical investigation to study the effect of duct aspect ratio on heat transfer and friction characteristics. The optimum thermal enhancement factor was found at aspect ratio of 0.5. Xie et al. [4] presented numerical study of square cross-section ribbed channels with different arrangements of downstream half-size ribs. Using of downstream ribs was a suitable way to decrease the pressure loss and improve the flow structure while kept comparable enhancement in heat transfer as expected. Sriromreun et al.

[5] investigated the influence of baffle turbulators on heat transfer augmentation in a rectangular channel. In the experiment, the Nusselt number, friction factor and thermal performance enhancement factor for in-phase 45° Z-baffles were found to be considerably higher than the out-phase 45° Z-baffle. Promvonge et al. [6] numerically investigated on laminar flow and heat transfer characteristics in a three-dimensional isothermal wall square-channel fitted with inline 45° V-shaped baffles on two opposite walls. It was apparent that the longitudinal counter-rotating vortex flows created by the V-baffle can induce impingement/attachment flows over the walls resulting in greater increase in heat transfer over the test channel. Song et al. [7] numerically studied the flow and heat transfer in a finned flat tube heat exchanger with crossed discrete double inclined rib. The heat transfer is enhanced by secondary flows around the ribs and a series of vortices generated downstream of the ribs, including the main vortices, induced vortices and corner vortices. The Nusselt number for the attack angle of $\alpha=45^\circ$ was larger than for $\alpha=30^\circ$ and $\alpha=60^\circ$. The friction factor increased with the attack angle. In the present work, the study on thermal performance enhancement in a heat exchanger square-duct with V-shaped fins inserts has never been reported. These V-shaped fins acted as a vortex generator to create a vortex flow inside the duct. The result could lead to an increase in heat transfer rate. However, the pressure loss would be increased. Therefore, in order to yield the optimum thermal performance, the designed parameters which are shape, size, height, angle and pitch of the vortex generator are studied to investigate their effects on the heat transfer and flow friction.

2. Experimental Setup

A schematic diagram of the experimental setup is presented in Fig. 1. The system consists of a high-pressure blower, an orifice flow meter, a settling tank, and a square duct test section. The overall length of the duct was 3000 mm comprised an entrance section of 2000 mm and a test section of 1000 mm (L). The test section in square duct was made of aluminum plate having thickness of 3 mm. and a cross sectional area of $45 \times 45 \text{ mm}^2$ (H×H). Fig. 2 shows the V-shaped fins placed on a double-sided aluminum frame. The V-fin's geometries were fin-to duct-height ratio or blockage ratio, ($e/H=BR=0.2$), fin pitch to duct height ratio, ($P/H=PR=0.5, 1.0, 1.5, 2.0, 2.5$ and 3.0) and fin attack angle, $\alpha=45^\circ$ with V-tip pointing downstream (called "V-downstream")

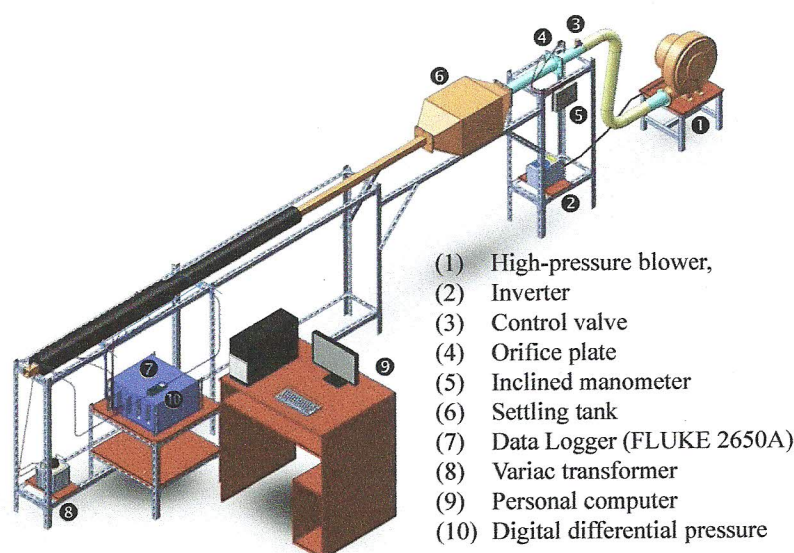


Fig. 1 Schematic diagram of experimental apparatus.

The AC power supply provided energy for heating four walls of the test section and maintaining a uniform surface heat flux condition. Air as the working fluid was forced through the system by a 1.45 kW high-pressure blower. An inverter was used to control the air flow rate. The flow rate was measured by using an orifice plate. The orifice plate was calibrated by hot wire and vane-type anemometers. The pressure drop across the orifice was measured using an inclined manometer. The

axial temperature distributions along the test section were measured by twenty-eight thermocouples. Two thermocouples were positioned at the entrance and the exit of the duct to measure the inlet and outlet temperatures. All measured temperature values were fed into a data logger (Fluke 2650A) and recorded via a personal computer. Two static pressure taps were located at the top walls to measure axial pressure drops across the test section. This pressure drop was determined to calculate the friction factor. The pressure drop was measured by a digital differential pressure transducer.

The uncertainty analysis in the data calculation was based on [8]. The maximum uncertainties of non-dimensional parameters were $\pm 5\%$ for Reynolds number, $\pm 7\%$ for Nusselt number and $\pm 9\%$ for friction factor.

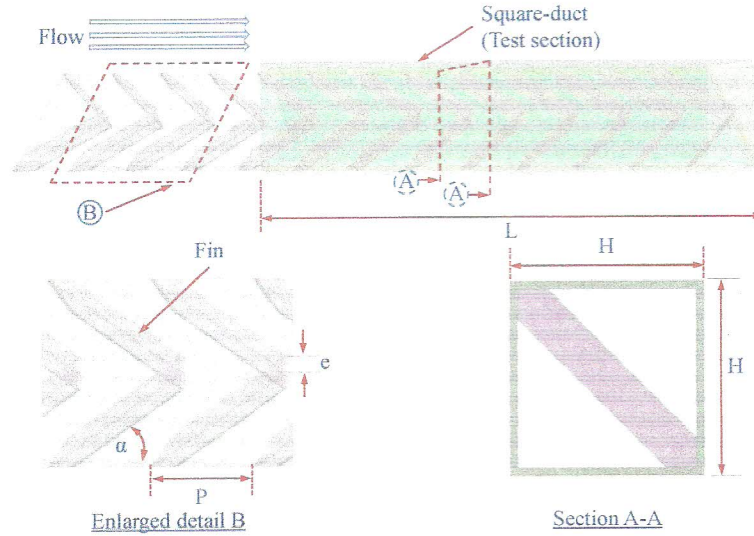


Fig. 2 Test section with V-shaped fins

3. Data processing

The present experiment was conducted to investigate the heat transfer, pressure drop and thermal performance in a square-duct inserted with the V-shaped fins. The results obtained are displayed in dimensionless terms of Nusselt number and friction factor. The average heat transfer coefficients are calculated by using the experimental data via the following equations:

$$Q_{\text{air}} = Q_{\text{conv}} = \dot{m}C_p(T_o - T_i) \quad (1)$$

$$h = \frac{Q_{\text{conv}}}{A(\tilde{T}_s - T_b)} \quad (2)$$

in which,

$$T_b = (T_o + T_i)/2 \quad (3)$$

and

$$\tilde{T}_s = \sum T_s/28 \quad (4)$$

where A is the heat transfer surface area of duct, T_s is the local surface temperature along the duct length, and \tilde{T}_s is the average surface temperature. Thus, the average Nusselt number is written as

$$Nu = \frac{hD_h}{k} \quad (5)$$

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

$$Re = UD_h/\nu \quad (6)$$

The duct hydraulic diameter is defined by

$$D_h = \frac{4A_c}{P_w} = H \quad (7)$$

in which, A_c is the cross-sectional area and P_w is the wetted perimeter of the cross section. The friction factor is evaluated by

$$f = \frac{2}{(L/D_h)} \frac{\Delta P}{\rho U^2} \quad (8)$$

where ΔP is the pressure drop across the test duct and U is the mean air velocity in the duct. All properties of air are evaluated at the overall bulk air temperature from Eq. (3).

The thermal enhancement factor (η) defined as the ratio of the heat transfer coefficient of an inserted duct, h to that of smooth duct, h_0 , at a constant blowing power [9] is given by

$$\eta = \frac{h}{h_0} \bigg|_{bp} = \frac{Nu}{Nu_0} \bigg|_{bp} = \left(\frac{Nu}{Nu_0} \right) \left(\frac{f}{f_0} \right)^{-1/3} \quad (9)$$

4. Results and Discussions

4.1 Validation of smooth duct

The experimental results of Nusselt number and friction factor obtained from the present smooth duct are compared with those from correlations of Gnielinski and Petukhov found in Ref. [10] for turbulent flow in ducts.

Gnielinski's correlation

$$Nu = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad (10)$$

Petukhov's correlation

$$f = (0.79 \ln Re - 1.64)^{-2} \quad (11)$$

The comparison of Nu and f obtained from the present smooth duct with those from correlations of Eqs. (10, 11) is depicted in Fig. 3. It is seen in the figure that the present results are in good agreement within $\pm 5\%$ with the correlation data.

4.2 Effect of PR on heat transfer

The Nu and Nu/Nu_0 plotted against Re values is displayed in Fig. 4a and b, respectively. The Nusselt number ratio, Nu/Nu_0 , is defined as a ratio of the enhanced Nusselt number to Nusselt number of smooth duct. In Fig. 4a, the inserted duct yields the considerable heat transfer enhancement with similar trend pattern in comparison with the smooth duct and thus, the Nu increases with the rise of Re .

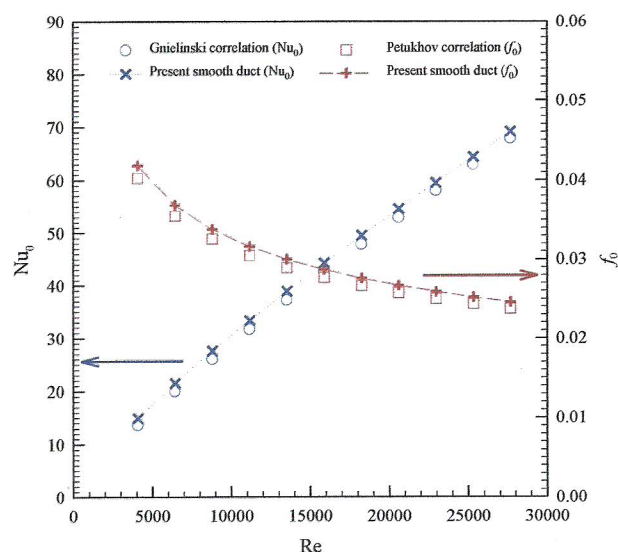


Fig. 3 Validation of Nu_0 and f_0 for smooth duct.

It is seen in the Fig. 4b that the decrement of PR results in the increase in Nu/Nu_0 . This is because the effect of the vortex flows from the V-shaped fins can help to increase the turbulence intensity and to transport the fluid from the central core to the near-wall regions. Also, the vortex flows can wash up the flow trapped in the duct corner regions normally act as ineffective heat transfer areas, leading to higher heat transfer rate in the duct. At a given BR, the V-shaped fins provides the Nu/Nu_0 in the range of 3.93-4.00, 3.82-3.89, 3.62-3.68, 3.43-3.50, 3.18-3.24 and 3.00-3.05 for PR=0.5, 1.0, 1.5, 2.0, 2.5 and 3.0 respectively. It can be observed that the duct with PR=0.5 provides heat transfer rate higher than others. This can be attributed to the use of V-shaped fins with the smallest pitch leading to stronger vortex strength of the flow and thus promoting high levels of mixing over the others.

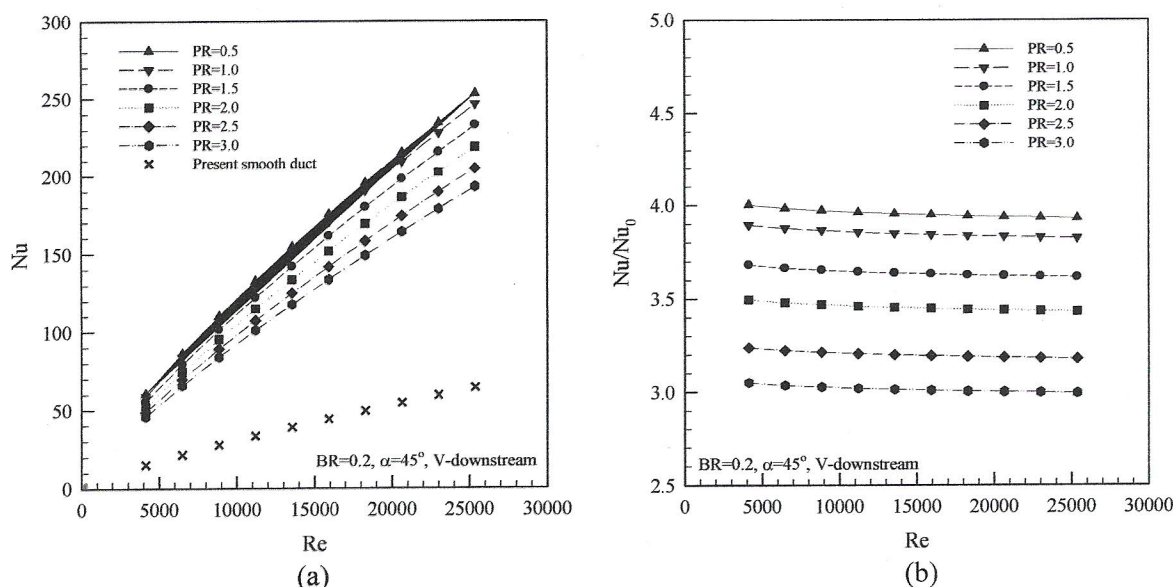


Fig. 4 Variation of (a) Nu and (b) Nu/Nu_0 with Re .

4.3 Effect of PR on friction loss

The plot of f and f/f_0 against Re is, respectively, exhibited in Fig. 5a and b for various PR values. It is visible in Fig. 5a that the use of the V-shaped fins vortex generators leads to substantial increase in f above the smooth duct and the f shows the decreasing tendency with the increment of Re . It can be

observed in the Fig. 5b that the ff_0 for the inserted duct tends to increase considerably with rising Re but with reducing PR. The inserted duct at PR=0.5 provides the highest ff_0 value. This is because the use of PR=0.5, caused higher flow resistance, larger surface area and the reverse flow, leading to the substantial increase in pressure drop. The ff_0 values for the V-shaped fins at BR=0.2 are in the range of 28.04–33.62, 22.00–26.38, 16.30–19.54, 14.94–17.91, 13.10–15.70 and 11.82–14.17 for PR=0.5, 1.0, 1.5, 2.0, 2.5 and 3.0 respectively.

4.4 Thermal performance evaluation

Fig. 6 presents the influence of PR values on thermal enhancement factor (η) for using the V-shaped fins vortex generators. The η value was obtained from comparison of the Nu/Nu_0 and ff_0 ratios at similar blowing power as defined in Eq. (9). In the figure it is observed that the η shows the downtrend pattern with the increment in Re for all cases investigated. The use of the V-shaped fins leads to much higher η than that of the smooth duct for all cases. It is visible that the maximum η is achieved for using the V-shaped fins with PR=1.5 and this case of PR values is considered to be the best operating condition in this investigation. The highest η is about 1.45 at the lowest Re value.

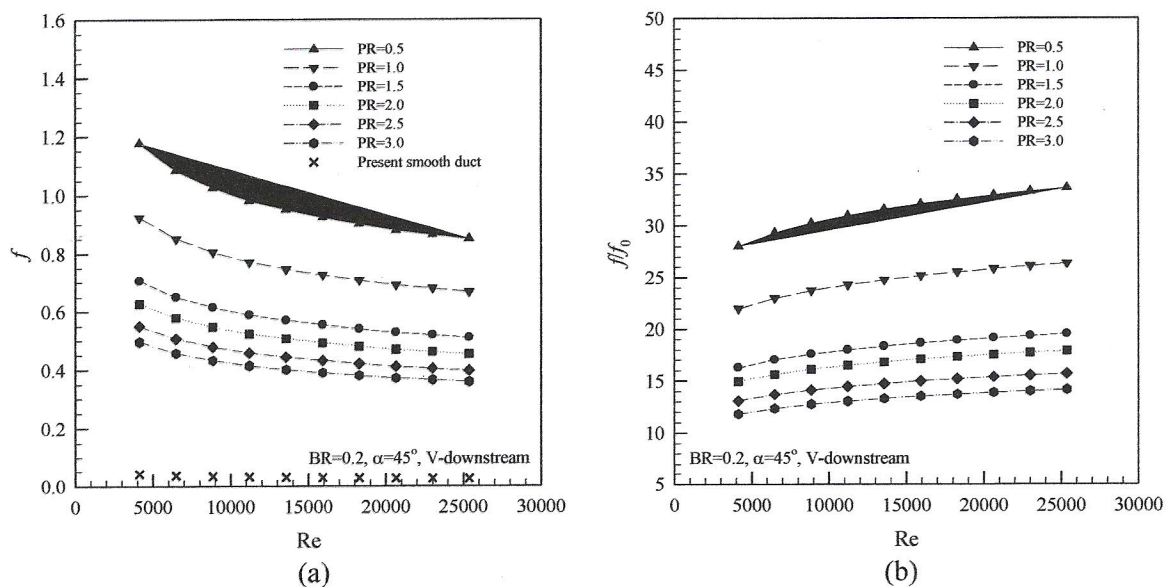


Fig. 5 Variation of (a) f and (b) ff_0 with Re.

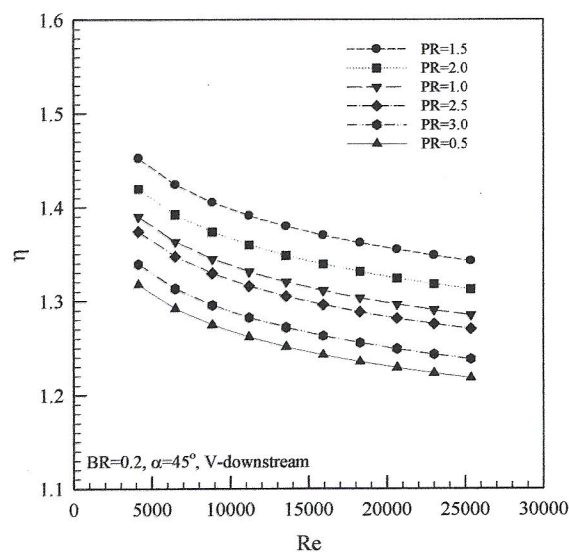


Fig. 6 Variation of η with Re

5. Conclusion

An experimental investigation on heat transfer, flow friction and thermal enhancement factor characteristics in a uniform heat-fluxed square-duct with the V-shaped fins vortex generator inserts at different PR for turbulent air flow, Reynolds number from 4000 to 25,000 has been conducted. The V-shaped fins insert provides a significant effect on the change of flow direction in the duct leading to the considerable increase in both heat transfer and pressure drop. It is seen that the highest heat transfer rate and pressure drop from the vortex generator devices is found at the smallest of PR. At the given BR, the V-shaped fins with PR=0.5 provides much higher Nusselt number and friction factor than the others while, the square-duct with PR=1.5 yields the maximum thermal enhancement factor is around 1.45.

6. References

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