*Original Article*

# Thermal Performance Enhancement in a Heat Exchanger Square-Duct With V-Shaped Fins Vortex Generator

Supattarachai Suwannapan[1*], Suriya Chokphoemphun[2], Panuwat Hoonpong[3], Pongjet Promvonge[4]

Received: 12 June 2015; Accepted: 13 July 2015

## 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°). 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 highest thermal performance.

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

## Introduction

Originally, heat exchangers were introduced using a plain or smooth surface. Then, an improvement was developed using several heat transfer enhancement devices. In the past, many researchers studied the effects of turbulator or swirl/vortex generator devices on heat transfer, pressure drop and thermal performance behaviors in pipe/duct heat exchanger. 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° was higher than $\alpha$=60°. Promvonge et al.[2] studied the turbulent flow and heat transfer characteristics in a square duct fitted diagonally with 30° angle-finned tapes. Promvonge et al.[3] numerically investigated on laminar flow and heat transfer characteristicsin 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. Singh et al.[4] experimentally investigated the heat and fluid flow characteristics of rectangular duct having one broad wall heated and roughened with periodic discrete V-down rib. Tanda[5] experimentally investigated the forced convection

[1] *Department of Mechanical Engineering, Faculty of Engineering, Rajamangala University of Technology Isan, Khonkaen Campus, 150 Srichan Road, Muang, Khonkaen, 40000, Thailand*

[2] *Major of Mechanical Engineering, Faculty of Industry and Technology, Rajamangala University of Technology Isan, Sakonnakhon Campus, 193 Phangkhon-Waritchaphum Road, Phangkhon, Sakonnakhon, 47160, Thailand*

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

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

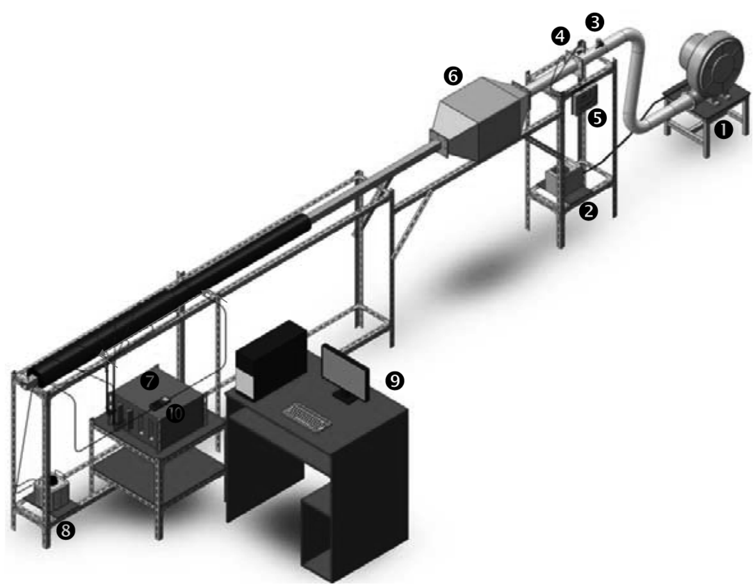
* *Corresponding author, E-mail address: oak_su@hotmail.com Tel.: +66 43-338-869-70; fax: +66 43-338-869-70*

heat transfer in a rectangular channel with inclined 45° angled rib turbulators. Hans et al.[6] experimentally investigated the effect of multiple V-rib roughness on heat transfer coefficient and friction factor in an artificially roughened solar air heater duct. Khan et al.[7] experimentally studied the heat transfer augmentation in developing turbulent flow through a ribbed square duct. Momin et al.[8] experimentally investigated the effect of geometrical parameters of V-shaped ribs on heat transfer and fluid flow characteristics of rectangular duct of solar air heater with absorber plate having V-shaped ribs on its underside.

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.

## Materials and Methods

A schematic diagram of the experimental setup is presented in (Figure 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 45×45 mm$^2$ (H×H). (Figure 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, a=45° with V-tip pointing downstream (called "V-downstream")




(1) blower, (2) inverter, (3) control valve, (4) orifice plate, (5) inclined manometer, (6) settling tank, (7) data logger, (8) variac transformer, (9) personal computer, (10) digital differential pressure.

**Figure 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[9]. The maximum uncertainties of non-dimensional parameters were ±5% for Reynolds number, ±7% for Nusselt number and ±9% for friction factor.

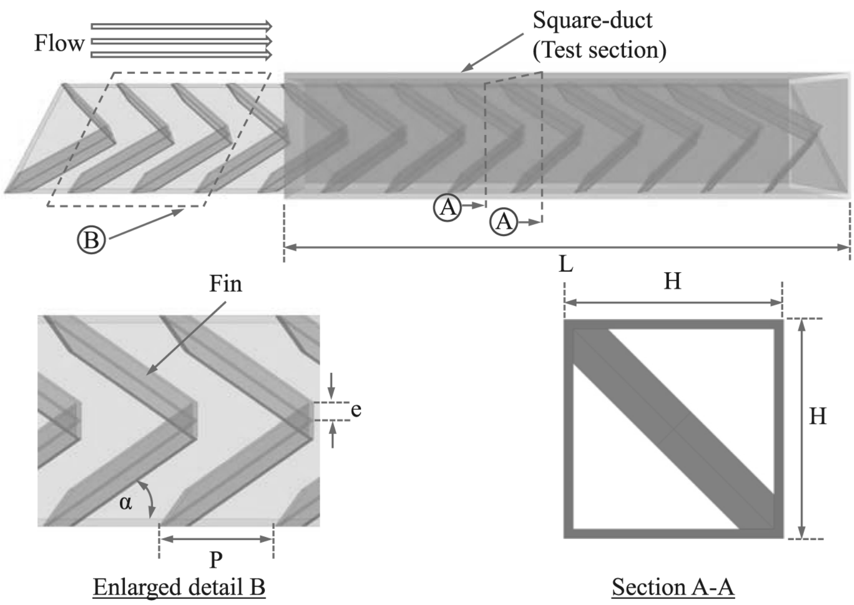



**Figure 2** Test section with V-shaped fins

## 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:

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

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

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

$$D_h = \frac{4A_c}{P_w} = H \tag{7}$$

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

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

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 (h) 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[10] is given by

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

## Results and Discussions

### 1. Validation of smooth square-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.[11] for turbulent flow in ducts.

Gnielinski's correlation

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

Petukhov's correlation

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

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

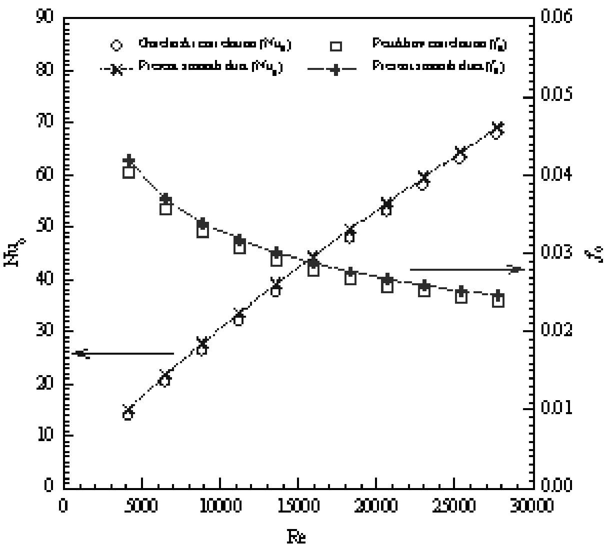


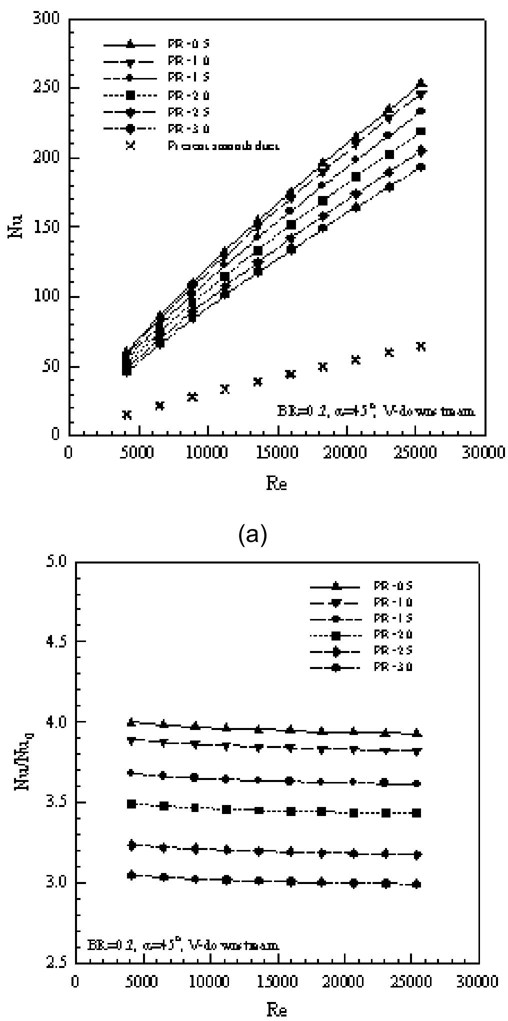

**Figure 3** Validation of $Nu_0$ and $f_0$ for smooth duct.

### 2. Effect of PR on heat transfer

The Nu and $Nu/Nu_0$ plotted against Re values is displayed in (Figure 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 (Figure 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. It is seen in the (Figure 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.

### 3. Effect of PR on friction loss

The plot of *f* and $f/f_0$ against Re is, respectively, exhibited in (Figure 5a) and b for various PR values. It is visible in (Figure 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 (Figure 5b) that the $f/f_0$ 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 $f/f_0$ 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 $f/f_0$ values for the V-shaped fins of the BR=0.2 is 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.



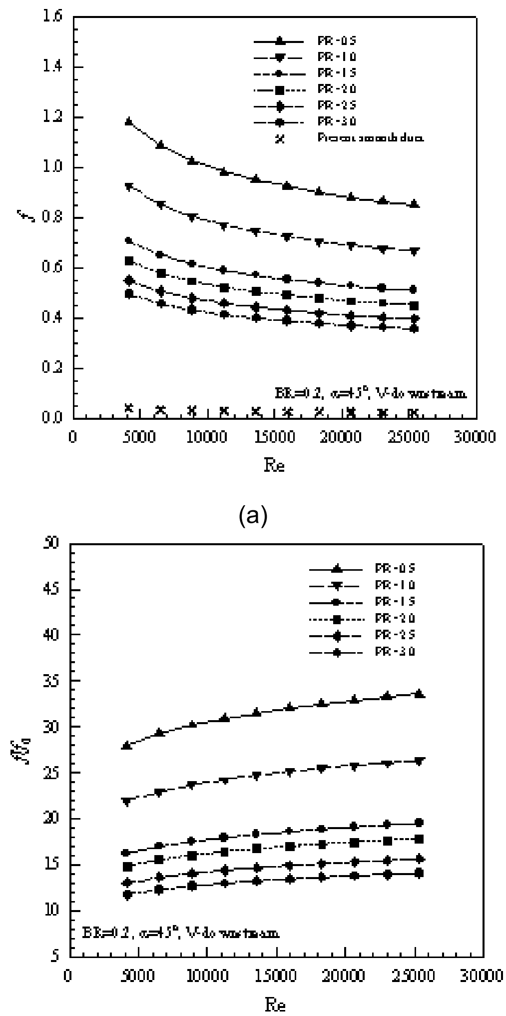

(a)

(b)

**Figure 4** Variation of (a) Nu and (b) $Nu/Nu_0$ with Re.

## 4. Thermal performance evaluation

(Figure 6) presents the influence of PR values on thermal enhancement factor (h) for using the V-shaped fins vortex generators. The h value was obtained from comparison of the $Nu/Nu_0$ and $f/f_0$ ratios at similar blowing power as defined in Eq. (9). In the figure it is observed that the h shows the downtrend pattern with the increment in Re for all cases investigated. The use of the V-shaped fins leads to much higher h than that of the smooth duct for all cases. It is visible that the maximum h 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 h is about 1.45 at the lowest Re.




(a)

(b)

**Figure 5** Variation of (a) *f* and (b) $f/f_0$ with Re.

## 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 maximum 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 V-shaped fins with PR=1.5 yields the highest thermal enhancement factor around 1.45 at the lowest Re value.

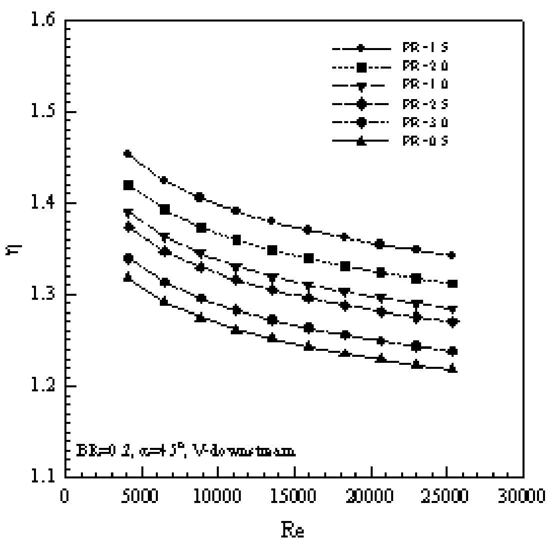



**Figure 6** Variation of $\eta$ with Re

## References

[1] Liu, J., Gao J., Gao T. and Shi X., Heat Transfer Characteristics in Steam-Cooled Rectangular Channels with Two Opposite Rib-Roughened Walls, Appl. Therm. Eng., 2013; 50, 104-111.

[2] Promvonge, P., Skullong, S., Kwankaomeng, S. and Thianpong, C., Heat Transfer in Square Duct Fitted Diagonally with Angle-Finned Tape-Part 1: Experimental Study, Int. Commun. Heat Mass Transfer, 2012; 39: 617-624.

[3] Promvonge, P., Jedsadaratanachai, W., Kwankaomeng, S. and Thianpong, C., 3D Simulation of Laminar Flow and Heat Transfer in V-Baffled Square Channel, Int. Commun. Heat Mass Transfer, 2012; 39: 85-93.

[4] Singh, S., Chander, S. and Saini, J.S., Heat Transfer and Friction Factor Correlations of Solar Air Heater Ducts Artificially Roughened with Discrete V-Down Ribs, Energy, 2011; 36: 5053-5064.

[5] Tanda, G., Effect of Rib Spacing on Heat Transfer and Friction in a Rectangular Channel with 45° Angled Rib Turbulators on One/Two Walls, Int. J. Heat Mass Transfer, 2011; 54: 1081-1090.

[6] Hans, V.S., Saini, R.P. and Saini, J.S. Heat Transfer and Friction Factor Correlations for a Solar Air Heater Duct Roughened Artificially with Multiple V-Ribs, Solar Energy, 2010; 84: 898-911.

[7] Khan, R.K., Ali, M.A.T. and Akhanda, M.A.R., Heat Transfer Augmentation in Developing Flow Through a Ribbed Square Duct, J. of Therm. Sci., 2006;15: 251-256.

[8] Momin, A.M.E., Saini, J.S. and Solanki, S.C. Heat Transfer and Friction in Solar Air Heater Duct with V-Shaped Rib Roughness on Absorber Plate, Int. J. Heat Mass Transfer, 2002; 45: 3383–3396.

[9] [ANSI/ASME, Measurement Uncertainty, PTC 19, 1–1985. Part I, 1986.

[10] Webb, R. L., Principles of Enhanced Heat Transfer, New York: John-Wiley & Sons, 1994.

[11] Incropera, F.P., Dewitt, D.P., Bergman, T.L. and Lavine, A.S., Introduction to Heat Transfer, 5th ED, John Wiley & Sons, 2006.