Enhanced thermal performance in a square-duct heat exchanger with inclined square-rings

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Abstract
An experimental work has been conducted to explore thermal and turbulent flow friction characteristics in a uniform heat-fluxed square-duct contained with inclined square-rings (SR). Experiments were performed for different geometric parameters of the SR elements, focused on their detailed effects on the optimal thermal performance. Influences of five different ring-to-duct height or blockage ratios (BR=b/H= 0.1, 0.15, 0.2, 0.25 and 0.3) on friction factor (f), Nusselt number (Nu), and thermal enhancement factor (TEF) in the square duct are investigated for Reynolds number (Re) from 4210 to 25,800. The inclination angle (\( \alpha \)) and the ratio of axial pitch length of rings to the height of duct (called pitch ratio, PR=P/H) were kept constant at 60° and 3.0, respectively. The experimental result has shown that the SR insert gives greater heat transfer and friction loss than the plain duct acting alone. Increasing the blockage ratio (BR) results in the drastic enhancement in Nu and f values. The inserted duct with BR = 0.25 has the maximum TEF around 1.57. Comparing thermal performance at identical operating conditions, the SR insert at BR = 0.15 gives around 1.1%, 2.3%, 3.2 and 4.5% higher than the one at BR = 0.2, 0.25, 0.3 and 0.1, respectively.

Keywords: Square duct, Vortex generator, Heat exchanger, Square-ring, Thermal performance
1. Introduction
A heat exchanger is a crucial link in the management of the industrial system and energy conservation, which usually determines the heat transfer rate and thermal performance of such a system. Therefore, a heat transfer enhancement technique is a subject of considerable interests to researchers, especially for the passive heat transfer enhancement techniques [1]. One of such techniques is the use of vortex flow devices. The tube/duct is one of the most commonly used heat exchanger components, which are often found in energy-related systems such as thermoelectric power generation, thermal power plant, waste energy conservation and solar energy collection. Vortex flow devices (so-called “vortex generators”, VGs) such as twisted tape [2], coiled wire [3], conical/diamond ring [4,5], baffle [6], and winglet [7] which all belong to a vital group of the VGs by insertion into the heat exchanger ducts/tubes, have been extensively adopted in enhancing the thermal performance of those systems. There have been many attempts to enhance the convection heat transfer rate in the tubes/ducts. Recently, García et al. [8] investigated the heat transfer augmentation in a tube using three types: corrugated tubes, dimpled tubes and wire coils. A new design of a twisted-tape (double V-ribbed twisted-tape) inserted in a heat exchanger tube was reported by Tamna et al. [9] for turbulent flow regime. They showed that TEF for the V-ribbed twisted-tape rises around 22% above that for the typical twisted-tape. A comparison of the thermo-hydraulic performances between twisted tapes and wire coils was suggested by Wang and Sunden [10] for both turbulent and laminar flows. They reported that the wire coils give high performance in augmenting heat transfer in a higher turbulent flow regime while the twisted tapes provide a poorer overall performance. Promvonge et al. [11] experimentally examined the heat transfer enhancement in a round tube using the angled ring inserts and suggested the ring conditions giving the highest thermal-hydraulic performance. Influences of using a vortex-induced impinging-jet phenomenon on enhancing thermal-hydraulic performance in a duct heat exchanger with oblique horseshoe baffles (HBs) were numerically and experimentally explored by Skullong et al. [12]. They found that the increase of thermal-hydraulic performance for the HBs is considerably higher than that for the wire coils, due to the effect of vortex-induced impinging-flow (VI), resulting in the drastic rise in the convection heat transfer within the duct.

Most investigations stated earlier have emphasized on enhancing thermal-hydraulic performance in many heat exchangers by methods of vortex generator (VG) inserts with various pitches, heights and different arrangements: angled/inclined, transverse. The experimental study on turbulent flow through the tandem inclined square-rings (SR) contained in a square duct has rarely been reported so far. The current SR device has been developed by the combined advantages of winglet, twisted-tape and baffle turbulators. This implies that the inserted SR device is expected to have a vortex flow like winglets, ease of practical use like twisted-tapes and drastically greater heat transfer rate like baffles. Thus, a newly 60° SR element is introduced to enhance convection heat transfer better than the twisted tape/coiled wire as a result of the VI effect as mentioned earlier. Hence, this article aims to examine the effect of the SR insert at various BR values on the rate of heat transfer, pressure loss and thermo-hydraulic performance in the square-duct heat exchanger. Experimental setup and formulas for measuring the Nusselt number, friction factor and thermal enhancement factor are presented. The test range of the flow in the duct is for Reynolds number (Re) between 4210 and 25,800.

2. Experimental program and procedure
The square duct in the present work had a cross-section of 45-mm height (H) × 45-mm width (W) with a 3500-mm total length included the 1000-mm test section (L). The SR elements made of a 0.8-mm aluminium sheet to shape them as shown in Figure 1 were fastened each other by putting four small straight wires into small holes on the four corners of the SRs. The 60° inclined SRs at five different ring-to-duct height or blockage ratios (BR=b/H= 0.1, 0.15, 0.2, 0.25 and 0.3) were fitted in the square-duct by surface-attached position.

The schematic of the experimental setup in this study is depicted in Figure 2. The setup mainly consists of a calm section, test section and exit section. The test duct was made of aluminium plates in which air flowed through it. To heat the duct uniformly, an electrical heating wire was continually
winding around the duct to give a constant wall heat flux. The outermost duct-wall surface was covered entirely by a 50-mm thick ceramic insulation to avert the convective heat loss to the surrounding. The 2-kW blower directed the air with $T_{in} = 25^\circ C$ through an orifice flow meter. An inverter was utilized to adjust the airflow rate by controlling the motor speed of the blower to achieve the desired Reynolds number between 4210 and 25,800. For measuring temperatures of the air at the outlet and inlet ($T_{out}$ and $T_{in}$), RTD-typed temperature sensors were employed at certain positions upstream and downstream of the test duct using a multi-channel temperature measurement unit whereas 28 T-type thermocouples for measuring the wall temperatures ($T_w$) were placed equally on each of the side and upper walls along the test duct. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2680A) and then recorded via a personal computer. A digital manometer (Dwyer Mark III) was employed to obtain the pressure drop across the air duct. More details on the experimental set-up, method and uncertainty analysis were already mentioned in another paper of the authors [12].

![Figure 1. Test square-duct with inclined 60° SRs.](image1)

![Figure 2. Schematic sketch of the experimental set-up.](image2)

3. Data Processing
The current experimental work aims to examine the pressure loss (friction factor, $f$), heat transfer (Nusselt number, Nu) and thermal-hydraulic performance in a square duct fitted with SRs. The data processing is as follows:

The parameters involved include the flow blockage ratio (BR) and Reynolds number (Re). The Reynolds number is written as
\[ \text{Re} = \frac{\rho \times U \times D}{\mu} \]  

(1)

The \( f \) evaluated using the pressure drop of airflow along the test duct is defined as:

\[ f = \frac{2}{(L/D)} \frac{\Delta P}{\rho \times U^2} \]  

(2)

where \( U \) is the average velocity of the fluid in the duct.

In this work, air entered the test duct having a constant wall heat-flux. At a steady-state, the rate of heat transfer to the air can be presumed to equate the convection heat loss of the tested duct expressed as:

\[ Q_{\text{air}} = Q_{\text{conv}} \]  

(3)

where

\[ Q_{\text{air}} = \dot{m} \times C_{p,\text{air}} \times (T_{\text{out}} - T_{\text{in}}) \]  

(4)

The heat transfer within the test duct can be estimated by

\[ Q_{\text{conv}} = h \times A \times (\bar{w} - T_b) \]  

(5)

in which

\[ \bar{w} = \frac{\sum T_w}{28} \]  

(6)

and

\[ T_b = \frac{T_{\text{out}} + T_w}{2} \]  

(7)

where \( T_w \) is the local wall-temperature positioned equally along the test duct. The average wall temperature, \( \bar{w} \), was estimated from 28 points of the local surface temperatures. Thus, the evaluations of the mean heat transfer coefficient \( (h) \) and the average Nu are as follows:

\[ h = \dot{m} \times C_{p,\text{air}} \times \frac{(T_{\text{out}} - T_{\text{in}})}{A \times (\bar{w} - T_b)} \]  

(8)

Then, the Nu is achieved by

\[ \text{Nu} = \frac{h \times D}{k} \]  

(9)

All properties of the air are determined at the bulk mean air temperature \( (T_b) \) by Eq. (7).

A novel thermal enhancement factor (TEF) is employed as a thermal-hydraulic performance indicator of the inserted duct which its \( h \) is relative to the smooth duct at the same pumping power as suggested by Refs. [13,14] and expressed by

\[ \text{TEF} = \frac{h}{h_0} = \frac{\text{Nu}}{\text{Nu}_0} \left( \frac{f}{f_0} \right)^{0.8} \left( \frac{\text{Pr}}{\text{Pr}_0} \right)^{0.25} \]  

(10)

where subscript “0” stands for the smooth duct only.

4. Results and Discussion

4.1. Validation test

The smooth square-duct was tested to verify the experimental setup. The comparison of the \( f \) and Nu results of the current smooth square-duct with the results of the published correlations [15] of Blasius and Dittus-Boelter, as prescribed by Eqs. (11) to (12), is shown in Figure 3 and 4, respectively.

Correlation of Blasius,

\[ f = 0.316 \text{Re}^{-0.25} \]  

(11)

Correlation of Dittus-Boelter:

\[ \text{Nu} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \]  

(12)
Figure 3. Validation test of (a) $f$ and (b) $Nu$ for smooth square-duct.

Figure 3a and 3b show the comparison between the present experimental work and the standard correlations of Blasius (Eq. 11) and Dittus–Boelter (Eq. 12). In the figures, the results of the present work agree reasonably well within ±7.6% for $f$ of Blasius and ±6.8% for $Nu$ of Dittus–Boelter correlations. Hence, the experimental setup is reliable.

4.2. Influence of SR on heat transfer

Figures 4a, b and c display the plots of the heat transfer coefficient ($h$), Nusselt number ($Nu$) and Nusselt number ratio ($Nu/Nu_0$) against $Re$ for various BR values of the 60º SR. The results in the figure show that $h$ of the SR inserts is increased considerably with the rise of $Re$ and BR (see Figure 4a). The $Nu$ for the inclined SR insert is also seen to be higher than that of smooth duct around 262–436%, and the greatest $Nu$ is found at the highest BR (BR=0.3) as shown in Figure 4b. This is because the presence of the SRs can help induce faster flow mixing, destruct the boundary layer and create vortex-induced impingement (VI) apart from prolonging the residence time in the duct. The variation of a ratio of enhanced $Nu$ to smooth duct $Nu$, ($Nu/Nu_0$) with $Re$ is exhibited in Figure 4c. As observed in the figure, $Nu/Nu_0$ has a slightly reducing tendency with rising $Re$. It is obvious that the largest $Nu/Nu_0$ can be found at BR = 0.3. This is because using the highest blockage ratio, BR = 0.3 leads to stronger vortex strength behind the SR and then, higher the convection heat transfer. In the current experimental data, the values of $Nu/Nu_0$ from the 60º SR insert are about 5.13–5.36, 4.93–5.17, 4.66–4.92, 4.35–4.63 and 3.61–3.94 times at BR = 0.3, 0.25, 0.2, 0.15 and 0.1, respectively. The BR = 0.3 yields the average $Nu/Nu_0$ around 4%, 9.8%, 17.3% and 40.5% higher than the BR = 0.25, 0.2, 0.15 and 0.1, respectively.
4.3. Influence on friction loss
Effects of the SR insert on pressure drop ($\Delta P$), friction factor ($f$) and friction factor ratio ($f/f_0$) are examined by plotting against $Re$ as displayed in Figures 5a, b and c, respectively. In Figure 5a, the variations of $\Delta P$ with $Re$ have the same trends for all cases, $\Delta P$ gradually increases with increasing $Re$ until $Re \approx 10,000$ and then, rises rapidly for $Re > 10,000$. $\Delta P$ of the inserted duct is found to be considerably higher than the smooth duct alone. It can be seen in Figure 5b that the SR insert has an extreme rise in $f$ above the smooth square-duct owing to higher flow blockage of larger BR value aside from the pressure dissipation due to increasing surface area of the SR. The $f$ of the SR inserts increases around $2361-6645\%$ above that of the smooth duct. Figure 5c shows the distributions of $f/f_0$ with $Re$ for different BRs. As seen in the figure, $f/f_0$ has the increasing tendency with $Re$ for all insert cases. The mean values of $f/f_0$ at BR = 0.3, 0.25, 0.2, 0.15 and 0.1 are, respectively, about 57.8, 49.6, 41.1 and 32.6 times. The maximum $f/f_0$ of about 67.45 times is found at the largest blockage ratio (BR = 0.3). The
average increase of $f/f_0$ for $BR = 0.3$ is, respectively, about 14.2%, 40.6%, 77.2% and 168% higher than that for $BR = 0.25, 0.2, 0.15$ and 0.1.

![Figure 5](image)

**Figure 5.** Variations of (a) $\Delta P$, (b) $f$ and (c) $f/f_0$ with Re for SR inserts.

### 4.4. Effect on thermal performance

The potential assessment from using the SR insert for a real practice is performed in the form of TEF. In general, TEF can be achieved by simultaneously evaluating the $f$ and Nu of the duct with SR inserts and of the smooth duct at identical pumping power conditions as denoted by Equation (10). The distribution of TEF with Re is portrayed in Figure 6. It is visible from the figure that TEF has a declining trend with rising Re for all insert cases. The values of TEF for $BR = 0.3, 0.25, 0.2, 0.15$ and 0.1 are, respectively, around 1.26–1.53, 1.27–1.55, 1.28–1.56, 1.23–1.57 and 1.24–1.51, depending on Re. The optimal TEF of about 1.57 is seen for $BR = 0.15$ at the lowest Re. Hence, if the choice of a vortex flow device is the SR, the $BR = 0.15$ condition should be used to achieve superior thermal performance.
5. Conclusions
An experimental procedure has been conducted to investigate a single-phase turbulent flow and heat transfer behaviors in a square-duct heat exchanger equipped with inclined square-ring vortex generators. Air is employed as the working fluid. The test range of the rate of fluid flow is Reynolds number between 4210–25,800 at a uniform wall heat flux. The experimental results show that the $f$ and Nu increase considerably with rising Re and BR, especially for $BR=0.3$. The 60° SR at $BR = 0.3$ gives the maximum $\frac{Nu}{Nu_0}$ and $\frac{f}{f_0}$ of about 5.36 and 67.45, respectively. The SR insert with $BR = 0.15$ has the maximum TEF around 1.57 and shows the declining trend with increasing Re.

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