Heat Transfer Enhancement with Considering Pressure Loss Penalty of Airflow through Heated Plate Mounted by Perforated Concave Rectangular Winglet Vortex Generators

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Heat Transfer Enhancement with Considering Pressure Loss Penalty of Airflow through Heated Plate Mounted by Perforated Concave Rectangular Winglet Vortex Generators

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Abstract. The use of longitudinal vortex generator results in decreased hydrodynamic flow performance by increasing pressure drop values. The purpose of this research is to obtain the improvement of heat transfer with pressure drop as low as possible. In order to achieve this purpose, experiment was performed on perforated concave rectangular winglet pair vortex generators (CRWP VGs). The test results show that the CRWP VGs without hole increase thermal performance by 24.02% up to 129.75% against baseline (without using any vortex generators) and by 12,21% up to 19,41% against rectangular winglet pair vortex generator (RWP VG). The pressure drop value for the CRWP VGs case increased by more than 10 times against the baseline case and increased by almost 100% against RWP VGs case. By using the perforated CRWP VGs, pressure drop value can be reduced by 15.07% to 72.73% followed by a slight decrease in thermal performance of 5.50% to 5.73% against the CRWP VGs without hole. Based on the evaluation using London goodness factor criteria, perforated CRWP VGs is better than the CRWP VGs without hole.

Keyword : *pressure drop, concave rectangular winglet pair vortex generator, rectangular winglet pair vortex generator*

1. Introduction

Compact heat exchangers have been widely applied in areas such as radiators in the automotive industry, condensation and evaporation in refrigeration, power plant cooling systems, solar-powered air/water heaters, exhaust gas recovery, cogeneration, steam generators and the pharmaceutical industry [1]. The most common type of heat exchanger is the fin-and-tube heat exchanger where heat from the hot fluid on the tube side is transferred to the gas using the extended surface, commonly referred to as fin. The performance of this type of heat exchanger is often limited by the thermal resistance on the gas side because the heat transfer coefficient of the gas is lower than the liquid fluid [2]. It has been known that 90% of the total thermal resistance of the fin-and-tube heat exchanger found on the gas side [3]. The development of fin with high effectiveness needs to be done to improve convection heat transfer on the gas side.

Convective heat transfer enhancement can be done by active and passive methods [4-5]. The active method is complicated because this method requires some external source input to increase the rate of heat transfer [4]. Instead, passive methods do not require the supply of external resources. The passive method uses a modified heat transfer surface and/or uses a turbulence generator element. The passive method is very popular in the industry because it has a simple design, cheaper production cost, and high reliability.

The passive method for convection heat transfer cases in the fin-and-tube heat exchanger can be done with two general concepts. The first concept is to enlarge the area of surface heat transfer such as fin. The second concept is to increase the convection heat transfer coefficient using a vortex generator (VG). VG is a protrusion in the flow wall and serves to generate the vortex in the flow [1]. Vortex can be divided into two types: longitudinal vortex (LV) and transverse vortex (TV). LV is a vortex that has a rotation axis with the main flow, while the TV is a vortex that has the direction of the rotation axis normal to the main flow direction.LV is considered to be more effective because LV is capable of covering three basic mechanisms for increasing the convection heat transfer coefficient: reducing the boundary layer thickness, swirl flow, and enhancing turbulence intensity. While the TV only covers the mechanism of increasing turbulence intensity. Based on its shape, there are four basic types of VG namely rectangular wings, delta wings, rectangular winglets, and delta winglets [6]. Winglets are better than wings because of winglets able to enhance heat transfer by generating more intensive LV with lower pressure drop [7].

The study of VG has been widely investigated because VG has an excellent heat transfer performance. L. H. Tang et al. [8] have conducted numerical studies using 2 types of VG namely rectangular winglet pairs (RWP) and delta winglet pairs (DWP) with common flow up (CFU) and common flow down (CFD) configurations. Each of VG surface is combined with an elliptical pole. The test results show that the use of DWP with the elliptical pole can improve heat transfer with high effectiveness. This is indicated by the angle between the velocity and the temperature gradient smaller than the baseline case (without VGs). Based on j/f factor, DWP-CFU with the elliptical pole is the best VG followed by DWP-CFU, RWP-CFD, and DWP-CFD.

Cheng-Hung Huang and Po-Ching Chiang [9] carried out a numerical three-dimensional study using the CFD-ACE+ commercial package and Lavenberg-Marquardt Method (LMM) methods to estimate the optimal position and shape of the delta winglet vortex generators on the pin-fin heat exchanger. The purpose of their research was to minimize the average temperature of the heat exchanger base plate. The thermal performance of the heat sinks that they observe can be increased with the help of vortex generators. The numerical test results show that the pin-fin heat sink has the best thermal performance when using the optimal vortex generator shape and position.

Anupam Sinha et al.[10] has numerically tested the fin-and-tube heat exchangers using a rectangular winglet pair (RWP) with a common-flow-up (CFU) configuration. Tests were carried out to determine the effect of heat transfer using variations in tube arrangement (inline and staggered) and variations in VG attack angle on each variation of tube arrangement. From the test results, it was found that the thermal performance of the inline tube arrangement is better than the staggered arrangement. From the results of this study observed that the performance of heat exchanger increases with increasing angle of attack. U. Kashyap et al.[11] conducted a numerical study of the impact of modified cross-section surface shape on the RWP on the heat transfer rate. From the test results, it was found that the surface modification can help in determining the position of the main vortex behind the VG which resulted in an increase in the strength of the main vortex core behind the VG. The use of curved profiles on the VG front surface is judged to be appropriate to increase the average Nusselt number.

In addition to increasing the rate of heat transfer, the use of VG results in an increase in pressure drop in the flow [4]. Flow with a large pressure drop requires a large pumping power as well. Many researchers have developed the concept of VG that can increase the rate of heat transfer with a low-pressure drop. Anupam Sinha et al. [12] has numerically tested the effect of using two rows of delta winglet pairs (DWPs) using a combination of VG orientation (CFU-CFU, CFD-CFD, and CFD-CFU) as well as VG line arrangement configurations (inline and staggered)for thermal performance and quality factor. From the test results found that DWP with CFU-CFU configuration has the best thermal performance and quality factor.

P. Saha et al. [13] performed the numerical study to determine the effects of RWP and DWP prepared with CFD and CFU configurations for heat transfer and pressure drop. From the test results, it is observed that the pressure drop in RWP is greater than DWP. Performance analysis shows that RWP provides better thermal performance than DWP.CFD configuration is better than CFU configuration.

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Azita Abdollahidan M Shams [7] conducted a numerical study to determine the effect of VG (rectangular winglets, trapezoidal winglets, and delta winglets) and the angle of attack on thermal and pressure drop performance. Their study is carried out by using Pareto optimal strategy to get an increase in maximum heat transfer and a low-pressure drop. From the test results found that rectangular winglet with 45 ° attack angle has the best heat transfer performance and highest pressure drop.

Li Li et al. [14] performed numerical testing to determine the effect of RWP and DWP on thermal and pressure drop performance. Their test results show that RWP and DWP can improve thermal performance compared to baseline. Thermal performance enhancement on RWP is better than DWP.Y. He et al. [6] performed a numerical study to determine the effect of RWP on heat transfer and pressure drop. The test is performed using variations of the number of VG rows, angle of attack, Reynolds number, and configuration of the RWP arrangement. The test results show that RWP can increase the heat transfer coefficient. The heat transfer coefficient also increases as the number of RWP rows increases. Increased heat transfer coefficient in this case is accompanied by an increase in pressure drop value. From the results of their study, it was found that the pressure drop on the staggered configuration was lower than the inline configuration.

S. Skullong et al. [15] performed an experimental study to determine the effect of winglet perforated tape (WPT) and non-perforated winglet tape (WTT) on heat transfer and friction factor. The Reynolds numbers varied from 4,000 to 30,000. The test results show that WPT and WTT produce higher heat transfer performance than baseline. WTT provides a higher heat transfer rate than the WPT because the holes in the winglet cause weakening of the vortex in the flow. The weakness of vortex strength in the flow can reduce the temperature gradient. Friction factor for WPT case is lower than WTT.

Syaiful et al. [16] conducted experimental and numerical study using the concave delta winglet pair vortex generator (CDWP VG). The variations used in their study are the number of VG rows. From their study results, it was found that the thermal performance for the CDWP VG case was greater than the DWP VG due to the instability of the centrifugal force when the fluid passed through the concave surface. This causes the LV to become stronger so that mixing between hot fluid and cold fluid becomes better which ultimately results in improvement of heat transfer. The results of their analysis also showed a greater increase in pressure drop in the case of CDWP than DWP. Their test results observed that thermal and pressure drop performance increased with increasing number of VG rows. When compared with the results of experimental tests, the numerical test results have a relatively small difference of less than 6%. Hemant Naik et al. [17] performed numerical testing to determine the effect of convex and concave RWP VG on thermal and pressure drop performance. From the test results, it was found that the highest thermal performance and pressure drop were owned by concave RWP VG. S. K. Sarangi and D. P. Mishra [18] performed a numerical study to determine the effect of RWP rows, attack angles, and RWP positions on thermal and pressure drop performance in fin-and-tube heat exchangers. Their test results show that thermal and pressure drop performance increases with increasing number of RWP lines and increasing the angle of attack. Each number of RWP rows each has an optimal position on the tube.

Then, Syaiful et al. [19] performed numerical study using RWP VG and concave rectangular winglet pair vortex generator (CRWP VG).Variations used are the number of VG rows and Reynolds numbers. Their test results show that the thermal performance in the CRWP VG case is better than RWP. This is because CRWP VG produces a stronger LV due to centrifugal instability when the fluid flows through the concave wall. From the test results, it was found that thermal performance increased with increasing number of VG rows.

According to a study conducted by Syaiful et al., thermal performance in the CRWP VG case is better than RWP [19]. However, the use of CRWP VG results in a higher pressure drop than RWP VG. Therefore, the current study is focused on reducing the pressure drop resulting from the use of the CRWP VG by giving holes on VG. For each RWP VG and CRWP VG cases, the variations used are inlet air velocity, number of VG rows, and number of holes. The attack angle used in this study is set to 45 ° from the direction of the main flow.

2. Materials and Methods

2.1. Experimental apparatus

This study was carried out in a rectangular air duct made of glass with a thickness of 10 mm. This air duct has a length of 370 cm, width 8 cm, and height H = 8 cm. The air was sucked by a centrifugal blower and enters through the inlet side. Then, the air was flowed through a straightener consisting of pipes arrangement with a diameter of 5 mm and a screen so that the flow becomes uniform. The inverter is used to adjust the rotating speed of the electric motor in the blower so that the inlet air velocity can be adjusted. The inlet air velocity was varied from 0.4 m/s to 2.0 m/s with an interval of 0.2 m/s. Inlet air velocity was measured using a hot wire anemometer (Lutron type AM-4204 with accuracy of ± 0.1). To determine the effect of VG on the rate of heat transfer, air was passed through a flat plate with/without VG. This flat plate was heated at a constant rate, Q, of 35 Watt by a heater whose was regulated using a heater regulator. Wattmeter (Lutron DW-6060 with an accuracy of ± 1.0) was used to monitor the heat rate into the plate. Inlet, outlet, and plate surface temperatures were measured using K-type thermocouples. Thermocouples were connected with data acquisition (Advantech type USB-4718 which has ± 0.001 accuracy) and computer CPU. Then, in order to determine the effect of the vortex generators on the pressure drop value, the pitot tubes were installed on the inlet and outlet sides of the test section. The pitot tubes were connected with micromanometer (Fluke) type 922 which has an accuracy of ± 0.001. In order to observe the characteristics of the vortex due to the use of VGs in the flow, visualization of the flow was carried out using smoke as a medium. The smoke was produced from heated oil in the oil heater. A compressor was used to push smoke from inside the oil heater to the air duct through the capillary tube. The capillary tube was mounted on the inlet side of the test specimen. For the vortex to be seen clearly in the flow, three green lasers were used and arranged in the direction of the main flow. The laser beam is refracted so that the beam illuminates the flow cross-section. The schematic diagram of this testing apparatus can be seen in Figure 1.

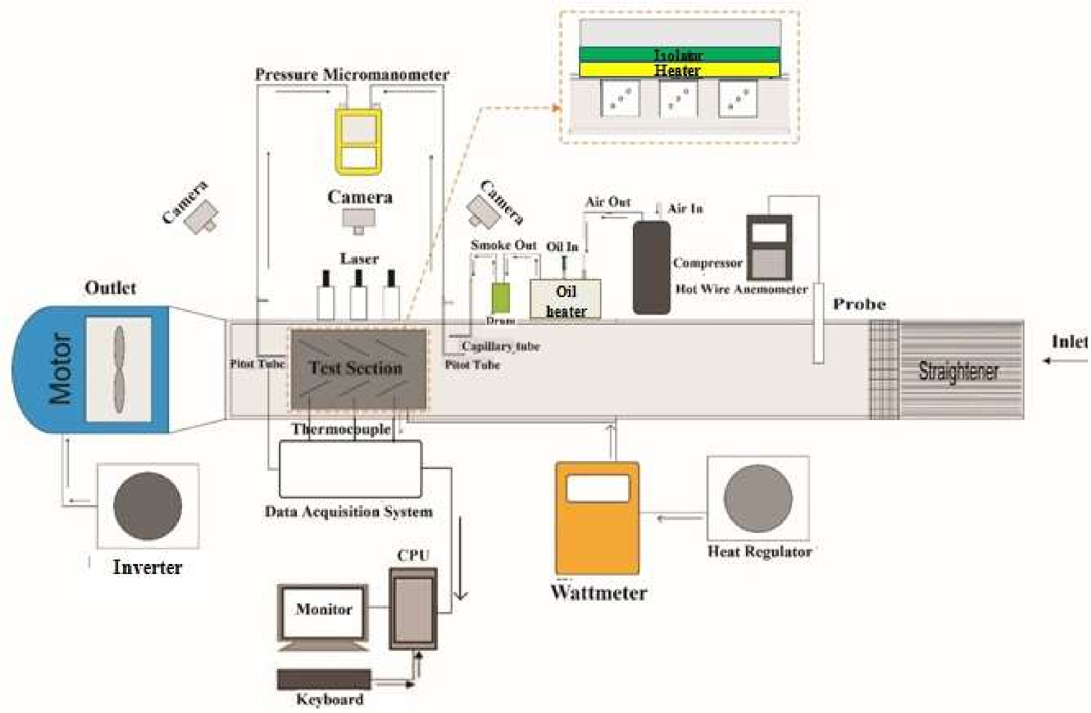
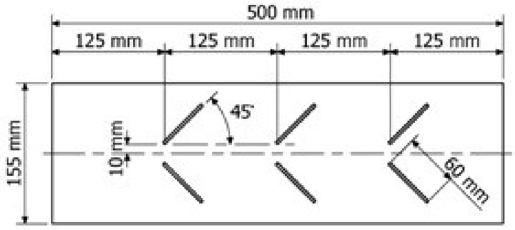


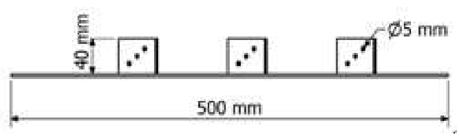
Figure 1. Schematic diagram of experimental apparatus

2.2. Test specimens

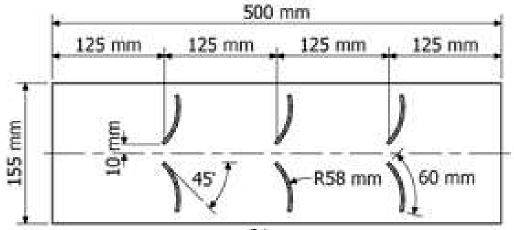
The test specimen used in this study was the winglet type. The type of winglet geometry used was rectangular winglet pair vortex generators (RWP VGs) and concave rectangular winglet pair vortex generators (CRWP VGs). Both test specimens were mounted on an aluminium plate that has a length of 500 mm, a width of 155 mm, and a thickness of 1 mm. The flow configuration used was common-flow-down (CFD) with the distance between the leading edge is 20 mm. The attack angle was set at 45° from the direction of the main flow. The specifications of the test specimen can be seen in Figure 2.



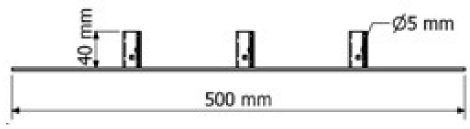
(a) Top view of three pairs DWP VGs



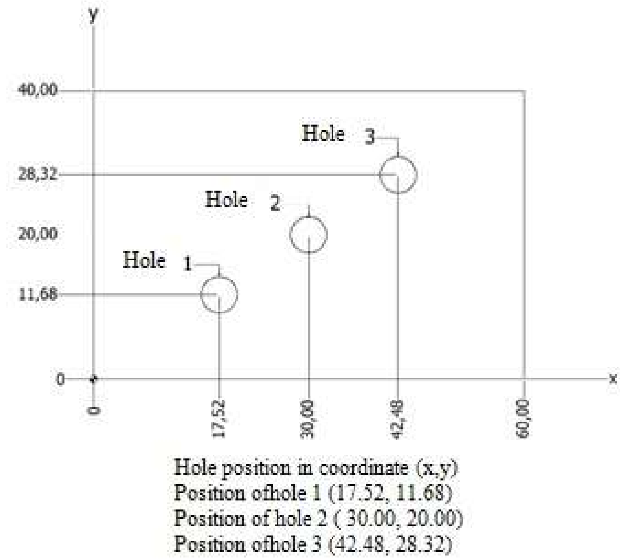
(b) Side view of three pairs DWP VGs



(c) Top view of three pairs CDWP VGs



(d) Side view of three pairs CDWP VGs



(e) Detail holes location in x-y coordinates

Figure 2. Geometry detail of test specimens

In the variation of the number of VG rows, the test was carried out using one, two, and three pairs of RWP VGs and CRWP VGs which are arranged inline. The distance of the leading edge between one pair of VGs and another VGs pair was 125 mm. The top view specifications of the test specimen can be seen in Figure 2 (a) for three pairs RWP VGs and Figure 2 (c) for the CRWP VGs. Figure 2 (b) and (d) show the side view of geometry detail of three pairs DW and CDW VGs, respectively.

In variations in the number of holes, the test was carried out using VG with one, two, three holes, and VGs without hole. Holes with a diameter of 5 mm were positioned as can be seen in Figure 2 (e). Position 2 is the hole location of the VG test with a variation of one hole. Positions 2 and 3 are the holes position of VG test with variation of two holes. Whereas, the three-hole vortex generator variation uses all hole positions.

Figure 3 shows an example of a test specimen of perforated RWP and CRWP VG that has been mounted on an aluminium plate with variations of one, two, and three rows of VGs. The test specimen of RWP VG case can be seen in Figure 3 (a). While the test specimen of CRWP VG case can be seen in Figure 3 (b).

(a) (b)

Figure 3. Test specimens for (a) RWP VGs and (b) CRWP VGs

2.3. Experimental method

For thermal performance experiment, the parameters measured were the temperature on the inlet, outlet, and flat surfaces of the plate. The temperature was measured using K type thermocouples which were connected with data acquisition. The data acquisition was connected to a computer device so that the temperature read by the thermocouples can be monitored directly and recapitulated into the table. The test specimen was heated at a constant heat rate, Q, of 35 W using a heater placed under the plate as shown in Figure 1. A heater regulator was used to adjust the rate of heating on the heater. The test specimen was heated to a steady temperature of 54°C-55°C. After steady was reached, the test can be started by turning on the blower and adjusting the frequency of the inverter until the reading of the air flow velocity on the hot wire anemometer reaches the desired velocity. The flow velocity range in this experiment was 0.4 m/s to 2.0 m/s with an interval of 0.2 m/s.

To calculate the convection heat transfer coefficient (h), the formula used refers to the Wu & Tao study [20]. The value of h can be calculated using the following equation:

$$h = \frac{\overline{Nu}k}{2H} \tag{1}$$

where $\overline{Nu}$ is the Nusselt number, k is the thermal conductivity, and H is the height of the duct. The Nusselt number value can be calculated using the following equation:

$$\overline{Nu} = \frac{Q}{A(T_w - T_{mb})}\frac{2H}{k} \tag{2}$$

where A is the heat transfer surface area, T_w is the average temperature of the heat transfer surface area and T_{mb} is the bulk temperature of the working fluid, $T_{mb} = 1/2\,(\overline{T}_{out} + \overline{T}_{in})$.

For hydrodynamic performance experiment, the pressure drop, ΔP measurement was carried out by measuring the pressure difference on the inlet side (P_{inlet}) and outlet side (P_{outlet}) of the test specimen in the test section so that:

$$\Delta P = P_{inlet} - P_{outlet} \tag{3}$$

Two pitot tubes were placed on the inlet and outlet sides of the test section. The two pitot tubes were connected with a micromanometer to monitor the pressure difference. The pressure drop data retrieval was carried out at 5-second intervals as much as 30 times at each inlet velocity variation. Similar with thermal performance experiment, the inlet velocity was varied in the range of 0.4 m/s to 2.0 m/s with an interval of 0.2 m/s.

Analysis of thermal and hydrodynamic performances was carried out using the London goodness factor criteria (*j/f*). Each thermal and hydrodynamic parameter was evaluated using the Colburn factor (*j*) for thermal parameter and friction factor (*f*) for hydrodynamic parameter. Colburn factor is calculated using the equation:

$$j = \frac{\overline{Nu}}{Re.Pr^{1/2}} \tag{4}$$

where *Re* is the Reynolds number, *Pr* is the Prandtl number. While friction factor is calculated using the equation:

$$f = \frac{2H}{L}\frac{\Delta P}{\frac{1}{2}V_{in}^2} \tag{5}$$

where *L* is the length of the test specimen plate and *ρ* is the density of the working fluid.

For the flow visualization, oil was put into the oil heater and heated until it evaporates. Then the smoke was pushed out of the oil heater using a compressor. The smoke entered the rectangular channel through the capillary tube placed in the inlet of the test section. To observe the vortex structures in the flow, three lasers were arranged in the direction of the main flow. The laser beam was refracted by using a transparent cylinder so that the area of cross-section laser beam was formed to capture the LV structures. A digital camera was used to capture and record the LV structures.

The thermal and hydrodynamic performance experiments were carried out on each type of VG (RWP VGs and CRWP VGs), the number of rows (one, two, and three rows), as well as the number of holes (one, two and three holes). As a comparison, a plate without VG was also tested for its thermal and hydrodynamic performance which is then referred to as the "baseline" case. For flow visualization, the test specimen used was CRWP VG with a variation of one row and three holes.

2.4. Data validation

Data validation was done to know that the data taken in the experiment are correct. The data in the present experiment were compared with the experiment data from Wu & Tao study [21] because of the present experiment is referred to their study. The data that is compared was found from testing with an identical VG configuration. In this case, an identical configuration is a baseline case (without VG). Comparison of data from the baseline case between the present and Wu & Tao studies can be seen in Figure 4.

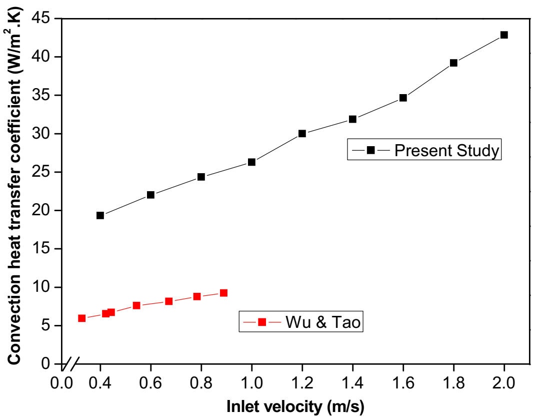


Figure 4. Comparison of convection heat transfer coefficient between present study and Wu and Tao study [20] for baseline case

From Figure 4, it is known that the two data have the same tendency, the convection heat transfer coefficient increases with increasing inlet air velocity. However, the results of the baseline data in the present study are greater than the results of the baseline on Wu & Tao's study due to smaller heat transfer to the test plate. In the Wu & Tao study, VG was mounted in the middle of the air duct. Whereas the present study, VG was placed on the channel wall. In other words, the hydraulic diameter in the present study is greater so that the convection heat transfer coefficient value becomes larger.

3. Results and Discussion

3.1. Effect of vortex generators on convection heat transfer coefficient

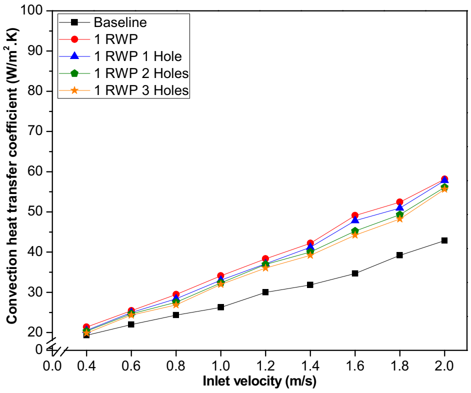
The effect of inlet air velocity, number of VG rows, and number of holes on convection heat transfer coefficients have been experimentally conducted. The convection heat transfer coefficient for each variation of the inlet velocity, number of VG rows, and number of holes for the RWP and CRWP VGs can be seen in Figures 5 and 6, respectively. The convection heat transfer coefficient was calculated by referring to the Wu and Tao study [20]. From Figures 5 and 6 it is observed that the use of RWP VG and CRWP VG can increase the convection heat transfer coefficient to the baseline case [19].

Generally, the convection heat transfer coefficient increases with increasing air inlet velocity because the LV produced by VG becomes stronger so that the intensity of mixing between hot fluid and cold fluid in the flow becomes better [16, 21, 22]. The convection heat transfer coefficient also increases with increasing number of VG rows [16, 18] because the resulting LV will continue to be strengthened until the last VG row. However, the convection heat transfer coefficient decreases slightly with increasing number of holes of the VG surface. Holes on the surface of the VG cause weakening of the vortex strength in the flow which results the reduction of the intensity of mixing between hot fluid and cold fluid [15]. The convection heat transfer coefficient for CRWP VGs case is greater than that of RWP VGs at the same of inlet velocity and the number of rows because the fluid experiences centrifugal force instability when the fluid passes through the concave wall producing stronger vortices [16, 19]. The LV produced by CRWP VGs becomes stronger and the radius becomes larger than RWP VGs causing better mixing between hot fluid and cold fluid.

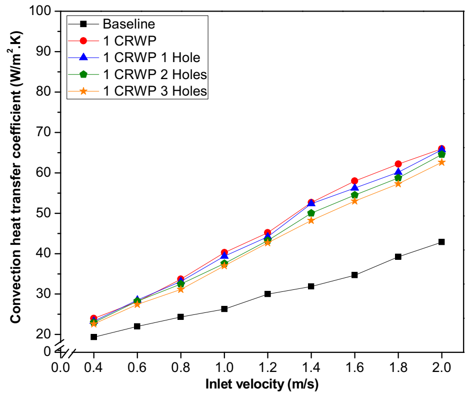
As seen in Figure 5, convection heat transfer coefficient for RWP VGs without hole cases with variations of one, two, and three VG rows are increased by 35.73%, 65.55%, and 92.40%, respectively, against the baseline at the inlet velocity of 2.0 m/s. For perforated RWP VGs cases, the convection heat transfer coefficient decreases slightly with increasing number of holes. As shown in Figure 5 (a), the convection heat transfer coefficient for one pair RW VGs case with variation of one, two, and three holes decreases 0.58%, 3.48%, and 4.35%, respectively, against one pair RW VGs without hole. In

Figure 5 (b), a decrease in convection heat transfer coefficient on two pairs RW VGs with variation of one, two, and three holes are found by 2.95%, 4.76%, and 6.67% against two pairs RW VGs without holes case. Whereas in Figure 5 (c) it is observed that convection heat transfer coefficient for the three pairs RW VGs case with variations of one, two, and three holes decrease 0.65%, 3.33%, and 7.47%, respectively, against three pairs RW VGs without hole.

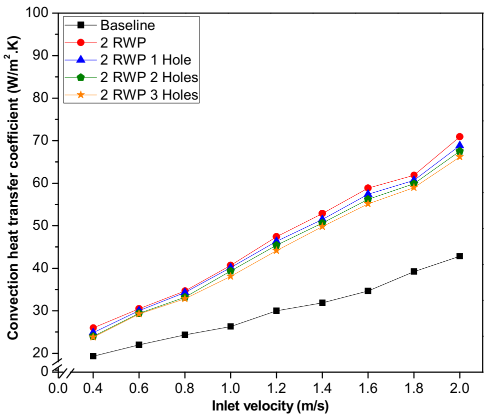
Figure 6 shows a comparison of the convection heat transfer coefficient for the CRWP VG case. For the case of CRWP VGs without hole, convection heat transfer coefficients with variations of one, two, and three VG pairs increase 54.01%, 89.53%, and 129.75%, respectively, against the baseline or 13.47%, 14, 48%, and 19.41% of RWP VGs without hole with the same number of pairs at the inlet velocity of 2 m/s. As in the case of RWP VGs, the convection heat transfer coefficient on the use of perforated CRWP VGs decreases slightly compared to those of case without holes. In the case of a single pair of CRW VGs with one, two, and three-holes as shown in Figure 6 (a), it is observed that the convection heat transfer coefficient decreased by 0.38%, 2.18%, and 5.12%, respectively, against CRWP VGs without hole. In the case of two pairs of CRW VGs as seen in Figure 6 (b), the convection heat transfer coefficients with one, two, and three holes decreases 2.53%, 3.28%, and 4.06%, respectively, against the CRWP VGs without hole. Whereas in the variation of the three-pairs of CRW VGs, the decrease in convection heat transfer coefficient is 2.15%, 3.95%, and 5.49% for the variations of one, two and three holes, respectively, against CRWP VG without hole as observed in Figure 6 (c).



(a)



(a)



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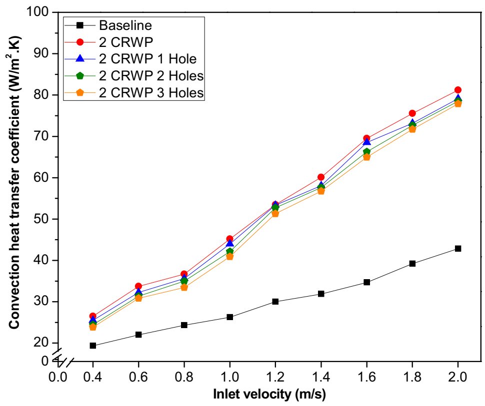

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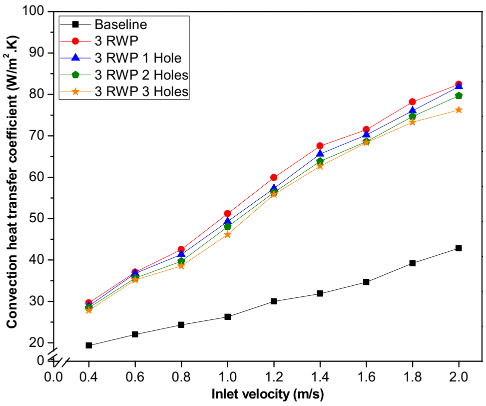
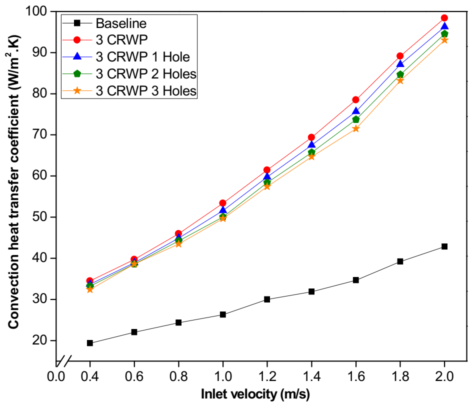


(c)

Figure 5. Comparison of convection heat transfer coefficients with variations of inlet velocity and number of holes for different number of pairs on (a) one pair, (b) two pairs, and (c) three pairs of RW VGs



(c)

Figure 6. Comparison of convection heat transfer coefficients with variations of inlet velocity and number of holes for different number of pairs on (a) one pair, (b) two pairs, and (c) three pairs of CRW VGs

3.2. Effect of perforated vortex generator on pressure drop

The effect of VG on the pressure drop can be seen in Figure 7 for the RWP VGs case and Figure 8 for the CRWP VGs case. In general, the pressure drops increase with increasing inlet velocity because the formed drag increases with increasing turbulent intensity. As shown in Figures 7 and 8 it is observed that the baseline case has the lowest pressure drop in each variation of the inlet velocity and the variation in the number of pairs because the baseline does not have VG that inhibits the flow. With the same number of pairs and number of holes, the pressure drop for the CRWP VGs case is greater than that of RWP VGs due to larger frontal area of CRWP VGs producing higher formed drag [16, 19].

As can be expressed in Figure 7, it is observed that the pressure drops for RWP VGs without hole case at the inlet velocity of 2 m/s increases to 158.67%, 355.33%, and 460% of the baseline case for RWP VGs with one, two and three pairs, respectively. The pressure drop can be reduced by using the perforated VGs because the longitudinal vortices (LVs) that are generated are weakened due to leakage of flow by the existence of holes [15]. At one pair RW VGs as described in Figure 7 (a), pressure drop decreases 7.22%, 14.95%, and 21.13% against RWP VGs without hole for one, two and three holes, respectively. The similar tendency is observed in Figure 7 (b), the pressure drops for the case of two pairs RW VGs with one, two and three holes is decreased by 8.20%, 11.86%, and 25.04% against RWP VG without hole, respectively. Then Figure 7 (c) expresses that the pressure drops for three pairs RW VGs with one, two, and three holes decreases 6.31%, 10.36%, and 13.93% against RWP VG without hole, respectively.

For the case of CRWP VGs without hole as shown in Figure 8, pressure drop with one, two, and three pairs increases 372.67%, 660%, and 1014.67%, respectively, against the baseline or increases 82.73%, 66.91 %, and 99.05% of RWP VGs, respectively, at 2 m/s inlet velocity. As in the case of RWP VGs, the pressure drop for the perforated CRWP VG case decreases with increasing number of holes. The longitudinal vortices produced by VG are weakened with the presence of hole on VGs due to flow leakage [15]. As seen in Figure 8, the pressure drops decrease with increasing number of holes for each variation in the number of pairs. In the one pair CRW VGs as expressed in Figure 8 (a), the pressure drops for one, two, and three holes decreases 7.89%, 12.27%, and 15.09%, respectively, against the CRWP VGs without hole. For the case of one, two and three holes of two pairs CRW VGs, the pressure drop decreases 10.44%, 14.65%, and 20.96%, respectively, against CRWP VG without hole as can be seen in Figure 8 (b). Whereas for the three pairs RW VGs case as stated in Figure 8 (c), the pressure

IOP Conf. Series: Materials Science and Engineering **506** (2019) 012062 doi:10.1088/1757-899X/506/1/012062

drops for variations of one, two, and three holes is decreased by 4.19%, 10.29%, and 15.07%, respectively, against CRWP VGs without hole.

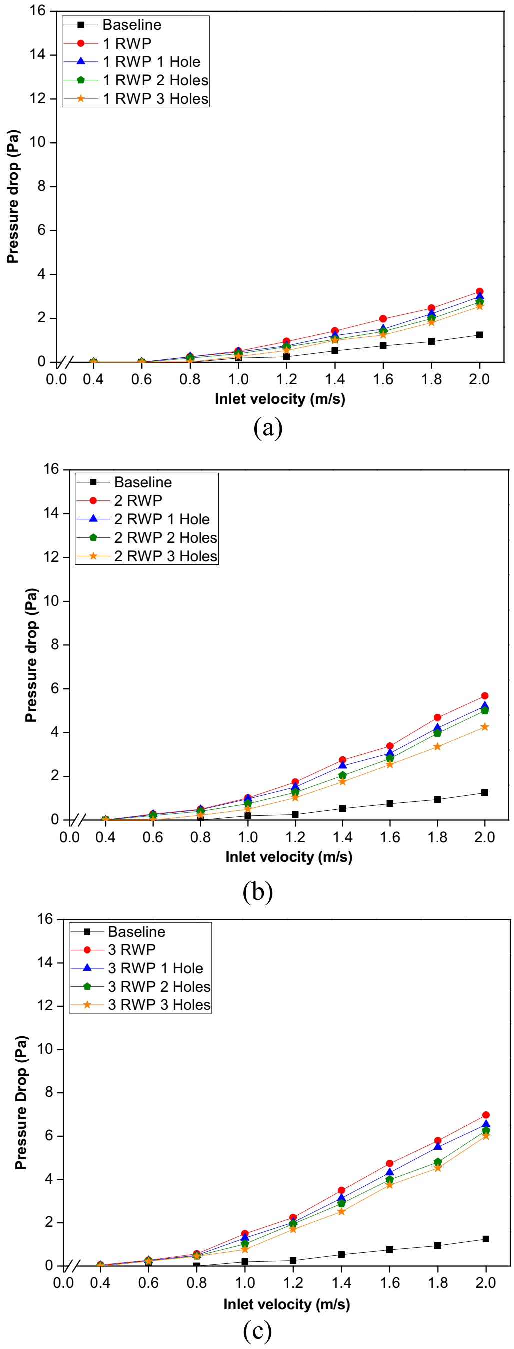


Figure 7. Comparison of flow pressure drop with variations of inlet velocity and number of holes for different number of pairs on (a) one pair, (b) two pairs, and (c) three pairs of RW VGs

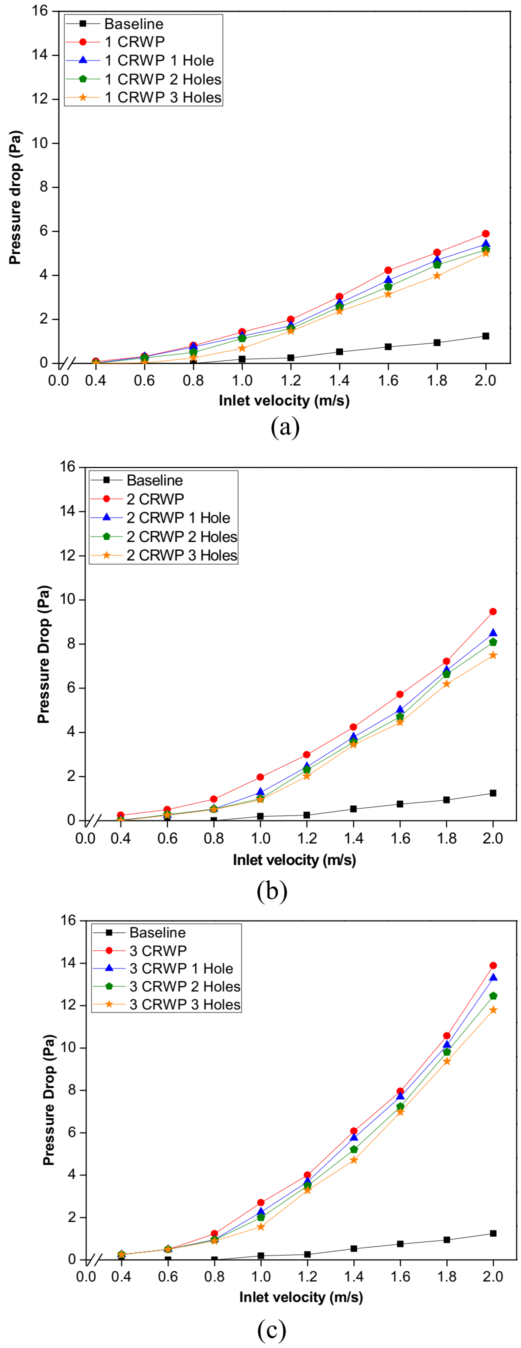


Figure 8. Comparison of flow pressure drop with variations of inlet velocity and number of holes for different number of pairs on (a) one pair, (b) two pairs, and (c) three pairs of CRW VGs

3.3. Thermal-hydrodynamic performances with presence of perforated vortex generators

Thermal and hydrodynamic performances are the main parameter components used to analyse heat exchanger performance. In order to determine the highest heat transfer and the lowest pressure drop, the criteria for London goodness factor (j/f) was used in this study. In general, the value of j/f will decrease with increasing inlet velocity [23]. At a certain inlet velocity, the difference in the value of j/f for each variation in the number of holes in the same number of VG rows tends to be insignificant [23, 24].

Figure 9 shows a comparison of the j/f criteria for the RWP VGs with and without cases with variations in the number of pairs. The variations of the j/f value for the RWP VG case become insignificant when the inlet velocity is more than 1.4 m/s. This is because the increase in pressure drop is greater than the increase in convection heat transfer performance at the highest velocity resulting low value of the j/f. At certain inlet velocity, the j/f value is unreadable in the image because the pressure drop value is outside of the micromanometer accuracy limit. Thus, the value of the criteria to be discussed in the RWP VGs case is the value found at the inlet velocity of 1.4 m/s. For RWP VG without hole cases, the j/f values for variations of one, two, and, three pairs with an inlet velocity of 1.4 m/s are 0.12, 0.08, and 0.08, respectively.

Figure 9 (a) shows a comparison of the j/f value for the RWP VGs one pair case with variations in the number of holes. By comparing London goodness between RWP VGs with and without holes, RWP VG with one, two, and three holes increases the j/f values of 14.15%, 29.46%, and 33.16%, respectively. The value of j/f in the case of inlet velocity less than 0.8 m/s is unreadable (for RWP VGs cases of one pair with one, two and RWP VGs without hole) and below 1 m/s (for RWP VGs one pair with three holes). This is caused by the pressure drop at the incoming velocity which is very low because the value is outside the micromanometer accuracy limit.

Furthermore, for two pairs RWP VGs case as shown in Figure 9 (b), an increase of 7.77%, 29.05%, and 47.31% is observed in a pair with the use of RWP VG with one, two, and three holes, respectively, against RWP VGs without hole. The value of j/f for this case is unreadable at inlet velocities less than 0.6 m/s (for two pairs RW VGs cases with one, two and without hole) and less than 0.8 m/s (for two pairs RW VGs case with three holes) due to limitation of micromanometer accuracy. Whereas for the case of three pairs RW VGs with one, two and three holes, the j/f value increases 6.34%, 8.55%, and 23.46%, respectively, against RWP VGs without hole as seen in Figure 9 (c). In this case, the j/f value is unreadable at the inlet velocity less than 0.6 m/s for all variations in the number of holes. This is because the pressure drop at velocity less than 0.6 m/s is very low.

From the evaluation results using the London goodness factor criteria in the RWP VGs case, the best configuration is found for the case of RWP VGs with a variation of three holes. This is expressed by the j/f value for the RWP VGs case with three holes in each variation in the number of VG pairs is the highest for almost all variations of the inlet velocity. The second bests configuration was found for RWP VGs with a variation of two holes. Then RWP VG with three holes and without holes is in the third and fourth position, respectively.

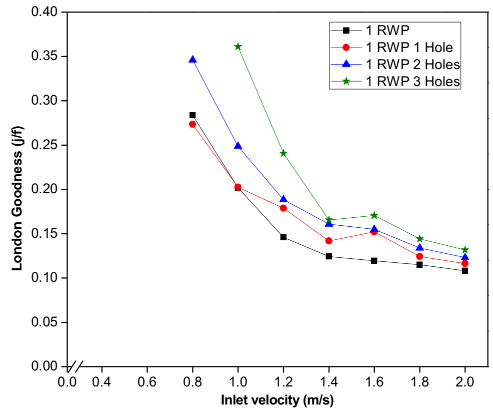
For the CRWP VGs case, the comparison of the j/f value between with and without holes can be seen in Figure 10. In this case, the change in the j/f value becomes insignificant when the inlet velocity is more than 1.2 m/s. This is because the increase in pressure drop is greater than the increase in convection heat transfer performance at the highest inlet velocity yielding the small j/f value. At certain inlet velocity, the j/f value is unreadable because the pressure drop value is outside of the micromanometer accuracy limit. In the case of CRWP VGs without hole at 1.0 m/s inlet velocity, the j/f values for one, two, and three pairs of VGs are 0.07, 0.06, and 0.05, respectively.

As observed in Figure 10 (a), the value of j/f for the case of one, two, and three holes of one pair CRW VGs are increased by 10.04%, 12.19%, and 17.20% against CRWP VG without hole. The value of j/f at this case is unreadable at the inlet velocity below 0.8 m/s (for one pair RW VGs case with one, two and RWP VGs without hole) and less than 1.0 m/s (for one pair RW VGs with three holes case). Then for the case of two pairs CRW VGs with one, two, and three holes, the value of j/f increases 8.23%, 14.30%, and 16.37%, respectively, against CRWP VG without hole as shown in Figure 10 (b). In the case of a two pairs CRW VGs, the j/f value is unreadable at inlet velocity less than 0.6 m/s (for two pairs CRW VGs case with one, two and CRWP VGs without hole) and less than 0.8 m/s (for the two pairs CRW VGs with three holes). Whereas for the three pairs CRW VGs case as shown in Figure 10 (c), the

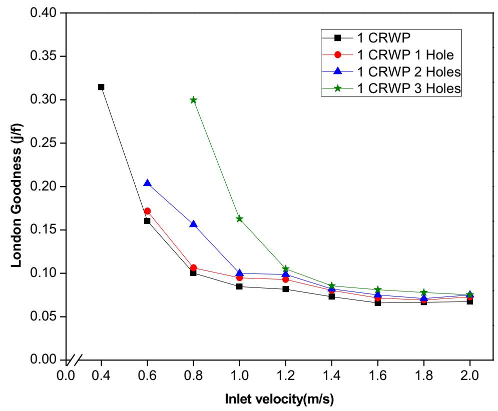
IOP Conf. Series: Materials Science and Engineering **506** (2019) 012062 doi:10.1088/1757-899X/506/1/012062

j/f value for one, two and three holes increases 2.75%, 10.61%, and 20.36% against CRWP VGs without hole. As it is observed in the case of CRWP VGs with one and two pairs, the j/f value is not readable at the inlet velocity less than 0.6 m/s for all variations of the number of holes.

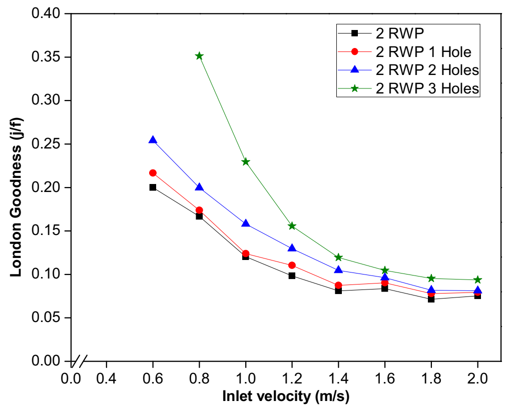
The best thermal and hydrodynamic performances which are evaluated are found in the CRWP VGs case with three holes. This is because the value of j/f for the case of the CRWP VGs with the three holes is the highest than the other variation of holes number at the same VG pairs and the same inlet velocity. Then, the second bests configuration was found by CRWP VGs with two holes and followed by CRWP with three holes and no holes.



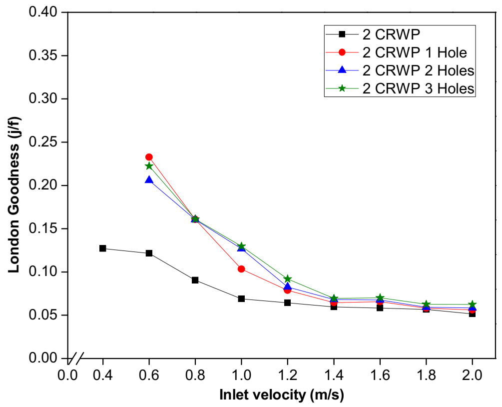
(a)



(a)



(b)



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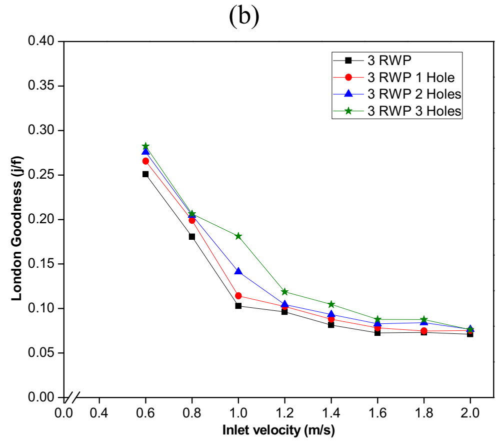
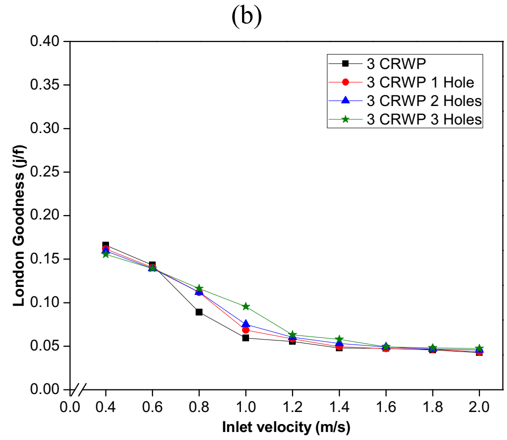


(c)



(c)

Figure 9. Comparison of flow London goodness with variations of inlet velocity and number of holes for different number of pairs on (a) one pair, (b) two pairs, and (c) three pairs of RW VGs

Figure 10. Comparison of flow London goodness with variations of inlet velocity and number of holes for different number of pairs on (a) one pair, (b) two pairs, and (c) three pairs of CRW VGs

3.4. Flow visualization
Flow visualization was performed to determine the LV structure produced by VG in the flow. The test was carried out using smoke from heated oil. The smoke was projected on cross-section planes in the flow obtained from laser beam induced to transparent cylinder. This visualization test used one pair CRW VGs with one pair and three holes at the inlet velocity of 0.6 m/s. The result of the visualization can be seen in Figure 11. From the Figure 11, it is observed that the pair winglet produces two LVs in the flow. Both LVs have opposite rotation directions. When it has been viewed from the inlet side, the LV to the left of the flow rotates clockwise while the LV to the right of the flow rotates counter-clockwise.

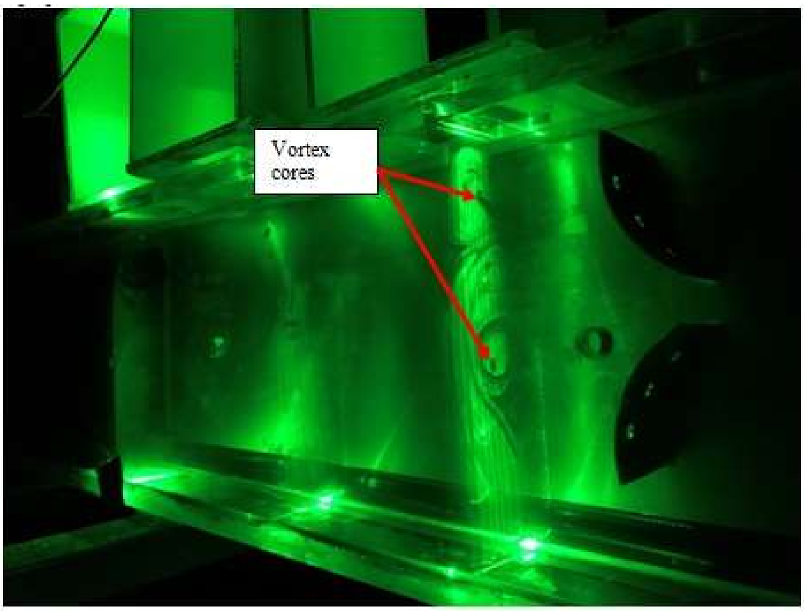


Figure 11 Visualization of longitudinal vortices generated by CRWP VG with various of one pair CRW VGs with three holes in the inlet velocity of 0.6 m/s.

3.5. Error calculation in experiment results
When the physical quantity measurement is performed using a measuring instrument and obtains numeric values, an evaluation of how close the measurement results to the true value needs to be performed. This is because it is very impossible to get physical data measurement results that are very precise in the measurement process. A good measurement process is carried out in accordance with the instructions for making measurements (measurement methods) that are truly implemented as best as possible. When this process is carried out repeatedly with the assumption of identical conditions, large amounts of measurement data has been found. Usually, the measurement data in each measurement process is not the same. By taking large amounts of physical data, therefore, requires scientific data processing to find out the errors in the data obtained from the actual value [25].

The calculation method used in processing data to determine deviations that occur is the Mean, Standard Deviation, Standard Deviation of the Mean, and Overall Error. Calculation of the error used in this test refers to Ernest O. Doebelin [25]. From the calculation, it is found that the average deviation of the calculation of convection heat transfer and pressure drop coefficients are 1.55% and 3.73%, respectively.

4. Conclusion

In the present study, the effect of using perforated RWP and CRWP VGs in thermal and hydrodynamic performances has been experimentally tested. The test results can be summarized as follows:

i. Thermal performance for the CRWP VGs case is better than RWP VGs. This was indicated by the convection heat transfer coefficients in the case of the CRWP VGs increased up to 19.41% against the RWP VGs. When using perforated VGs, thermal performance experienced a slight decrease of 7.47% against the VGs without hole.
ii. By judging from the hydrodynamic performance, the pressure drops in the case of the CRWP VGs increased to 99.05% against RWP VGs. However, the use of perforated VGs can reduce pressure drop up to 25.04% against VGs without hole.
iii. To determine the variation of VG with the best thermal and hydrodynamic performance, evaluation using the London goodness factor criteria (j/f) was required. The evaluation results showed that RWP VG and CRWP VG with 3 holes have the best thermal and hydrodynamic performances for each variation in the number of VG pair. This was indicated by the value of j/f for VG with three holes increased up to 47.31% against VG without hole.
iv. From the results of flow visualization, it was observed that the winglet pair VG produced two LVs with opposite rotation direction.

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