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Research Article

Heat Transfer Enhancement in Rectangular Channels with Punched V-Ribs: An Experimental Investigation

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ABSTRACT

In recent decades, researchers have focused on passive techniques that use vortex generators (VGs) to enhance the transport of heat. By trying to decrease the flow's pressure drop, this study aims to examine the strength of the vortex produced by VGs. The result show that the winglet vortex generators' increased attack angles produce stronger secondary flows, which improve fluid mixing and the heat transfer coefficient. Experiments were carried out in the Reynolds number range of 4096 to 20,480 in the current study to assess pressure drop and heat transfer enhancement. The study's findings show that, with comparatively modest errors, the measured coefficients of heat transport from the test data have the same inclination as per the trend. Compared to CRWVGs, RWVGs exhibit a lower pressure drop; however, there is a marginal reduction in heat transfer enhancements. Heat transfer performance was only decreased (by about 2.1%) when holes were added to RWVGs. But because of increased flow resistance, these improvements came with a higher pressure drop, for which RWVGs reached 35.37% and for CRWVGs even higher at 60°C.

1. Introduction

Improved heat transfer has a major positive impact on energy efficiency in contemporary heat exchangers. Vortex generators (VGs), one of the most popular passive methods, create longitudinal vortices that enhance heat transfer by promoting fluid mixing and upsetting the

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thermal barrier layer [1–4]. Geometric factors that affect vortex strength and pressure drop characteristics, such as aspect ratio, location, and attack angle, affect how well VGs perform [2–5].

Rectangular winglet vortex generators (RWVGs) and their variations have drawn the most interest among the many designs. Research indicates that while RWVGs can improve heat transfer, they frequently produce weaker corner vortices than better designs [6,7]. Stronger vortex production and improved fluid mixing are two reasons why concave rectangular winglet vortex generators (CRWVGs) exhibit higher thermo-hydraulic performance, especially at optimal attack angles [6]. Similarly, by enhancing turbulence intensity while maintaining pressure loss, wavy or curved RWVGs function better than traditional flat RWVGs [7, 8]. These results imply that significant increases in heat transfer efficiency can be obtained by altering the RWVG geometry, whether by concavity, curvature, or surface combinations.

Overall, studies show that optimizing VG design is crucial for avoiding pressure drop penalties and attaining substantial thermal enhancement, particularly through sophisticated RWVG and CRWVG combinations [9–15].

The thermal enhancement factor (TEF) is significantly influenced by geometry, as demonstrated by comparisons between V- and delta-shaped configurations, with V-shaped delta winglets typically outperforming their rectangular counterparts [15]. However, a number of factors, such as fluid flow rate, tube design, and fin shape, can affect VG efficiency [16–18]. Though frequently at the expense of greater pressure drop, research indicates that optimizing VG design can greatly boost the thermo-hydraulic efficiency of heat exchangers. Zhang et al. [19] study offers important new information about how RWVGs might be used to improve convective heat transfer in circular tubes. The results provide direction for designing more effective heat exchangers by highlighting the significance of geometric optimization in striking a compromise between enhanced heat transfer and tolerable pressure penalties.

The effect of vortex generators on the enhancement of heat transfer in a microchannel heat sink is examined in the work by Lee et al. [20]. According to the results, adding vortex generators considerably increases heat transfer rates while preserving a controllable rise in pressure drop.

2. Material and Methods

2.1. Experimental Set-Up

A rectangular channel of 5 m in length, 0.12 m in width, and 0.24 m in height served as the test portion. A pipe that joins the blower output to the heat exchange module in the air stream bench carries the forced air from the blower to the heat exchange module. A flow regulating valve is fastened to the pipe adjacent to the blower outlet in order to regulate the air flow. Before reaching the heated test section, air at barometric temperature is sent via a pipe that is about two meters long in order to create a fully formed flow. An orifice meter is placed adjacent to the regulating valve to track the pipe's regulated air flow rate. A water manometer is connected to the orifice meter. The water manometer shows the pressure differential across the orifice meter in centimeteres of water column difference. The differential manometer before the test section measures the pressure difference between the settling chamber and the inlet, which is used to determine the air velocity and corresponding Reynolds number. It ensures steady and uniform airflow conditions entering the test section. The manufactured heat exchange module is connected to an airflow bench in order to push the air parallel to the ribbed test surface. Attached to the bottom test plates, the strip plate heater is connected to a power outlet via a wattmeter. Wattmeter values are modified to supply the required heat input to the test plate. Temperature readings were taken using previously calibrated Copper-Constantan thermocouple wires. Because the thermocouples' accuracy was within ±0.5 °C, the temperature changes during the experiments were reliably measured. The schematic of the experimental setup is shown in Fig. 1.

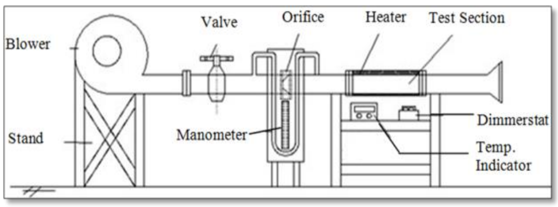


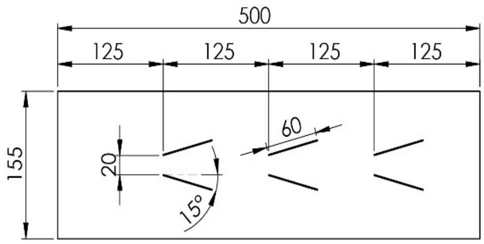
Fig. 1. Schematic of the experimental set-up

The test surface has provisions for fixing the thermocouple connection. Additionally, the air temperatures at the heat exchange module's input and output are recorded. The temperature readings (°C) obtained by the thermocouple wires are shown on a digital temperature indicator with twelve terminals. The image of the experimental setup is displayed in Fig. 2.

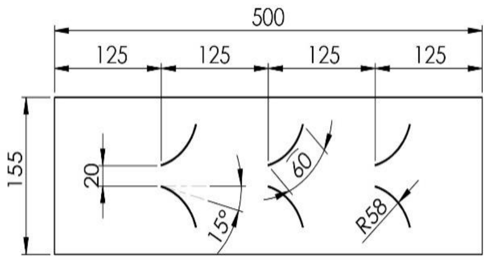
Fig. 2. Experimental set-up

2.2. Physical Model

Concave rectangular winglets (CRW) and rectangular winglets (RW) were the VGs utilized in this investigation. They were placed in a similar flow down with a 150 attack angle. Vortex generator was constructed from one millimeter thick aluminum plate with or without five millimeter diameter holes. Figure 3 displays the CRW and RWVGs' measurements and geometry. A pair of VGs have a transverse pitch distance of 20 mm and a pitch distance of 125 mm longitudinally.



(a)



(b)

Fig. 3. Test plate top view showing (a) RWVGs; (b) CRWVGs

An aerodynamic device called a vortex generator (VG) is made out of a tiny vane that is typically fixed to a wind turbine's rotor blade or a lifting surface (or airfoil, like an airplane wing). VGs can also be fastened to a component of an aerodynamic vehicle, like the fuselage of an airplane or a car. Using vortex generators to create longitudinal vortices that create a lot of swirling motion is one promising method that can enhance heat transfer at a minor cost of increased pressure loss.

Figures 4 and 5 give a detailed idea of the test plate with RWVGs and CRWVGs at different angles of attack.

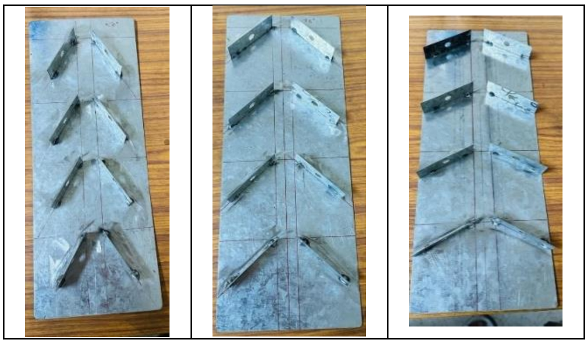


Fig. 4. RWVGs test plate with 30^0, 45^0, and 60^0 angle of attack

Fig. 5. CRWVGs test plate with 30^0, 45^0, and 60^0 angle of attack

2.3. Experimentation

The experimentation process is described in detail here. A plate heater is installed on the top surface of test plates, and tests for measuring forced convection heat transfer are chosen to be conducted on plain and ribbed surfaces.

The thermocouples are attached to the appropriate temperature indication channels. To show the temperatures at the appropriate surface and air temperatures, various temperature indicator channels are designed. The following are the temperatures that the temperature indicator recorded:

Temperatures at various locations on the test surface are denoted by T_1 to T_8.

T_{ai} is the temperature of the air at the input of the heat exchanger module.

T_{ao} is the air temperature at the heat exchanger module's exit.

A step-by-step explanation of the experimentation process used is as follows:

1. Turn on the heaters' power supplies at the necessary wattmeter value.
2. Verify that the temperatures recorded at each test surface point are the same once the test surfaces have reached an appropriate temperature.
3. To push air over the test surface, turn on the blower.

4. Manually adjust the regulating valve to change the air mass flow rate and show the required pressure differential measurement across the water manometer's orifice.
5. Record the temperatures at various test surface locations as well as the intake and outflow air temperatures once the steady state condition has been reached. The reproducibility of the readings is examined.
6. For the same input to the heaters, repeat the procedure for different flow rates of air.

The current experimental investigation simulated realistic thermal loading circumstances by uniformly applying a steady heat flow of 800 W/m^2 to the heated surface of the rectangular channel. Fully turbulent flow conditions were ensured by the inlet velocity varying between 0.4 and 2.0 m/s, which translates to a Reynolds number range of roughly 4000 to 20,000. To make sure the results could be replicated, each experimental condition was carried out three times. Analysis was conducted using the average of the repeated measurements, and it was determined that the fluctuation between repeats was within acceptable limits. In the current study, the influence of vortex intensity was inferred indirectly by observing changes in the Nusselt number and thermal performance factor with varying attack angles of the vortex generators, rather than directly measuring it (for example, using vorticity contours or field synergy angle analysis). A detailed stepwise calculation is given below:

- Step 1: Mean bulk temperature:

$$T_b = \frac{T_{ai} + T_{ao}}{2} \quad (K) \tag{1}$$

- Step 2: Orifice differential pressure:

$$\Delta P = \rho_w . g . h_w \quad (Pa) \tag{2}$$

- Step 3: Orifice area and volumetric flow:

$$A_o = \frac{\pi}{4} * d^2 \quad (m^2) \tag{3}$$

$$Q = C_d . A_o . \sqrt{\frac{2\,\Delta P}{\rho}} \quad (m^3 . s^{-1}) \tag{4}$$

- Step 4: Mass flow rate:

$$\dot{m} = \rho . Q \quad (kg . s^{-1}) \tag{5}$$

- Step 5: Hydraulic Diameter:

$$D_h = \frac{4.(cross - sectional\ area)}{wetted\ perimeter} = \frac{2 * W * H}{W + H} \quad (m) \tag{6}$$

- Step 6: Reynolds number:

$$Re = \frac{\rho . V_{avg} . D_h}{\mu} \tag{7}$$

- Step 7: Heat transfer rate and convective heat transfer coefficient:

$$q_{\text{conv.}} = \dot{m} . C_{\text{p}} . (T_{\text{ao}} - T_{\text{ai}}) \ (W) \tag{8}$$

$$h = \frac{q_{conv.}}{A_{plate} . (T_s - T_b)} \quad (W . m^{-2} . K^{-1}) \tag{9}$$

- Step 8: The total pressure drop across the test section was taken as the sum of distributed (friction) and local losses,

$$\Delta P = f \frac{L}{D_h} \frac{\rho . V_{avg}{}^2}{2} + (\sum_i K_i) \frac{\rho . V_{avg}{}^2}{2} \tag{10}$$

The friction factor f was calculated from the Swamee–Jain explicit formula for turbulent flow (and $f = {}^{64}/_{Re}$ for laminar flow). Local loss coefficients K_i for the inlet, fittings, and vortex generators were taken from published tables for similar geometries; when unavailable, representative values were used, and a sensitivity range is reported.

3. Results and Discussions

According to the current findings, perforated VGs at optimal angles of attack considerably improve heat transfer while reducing the pressure loss that follows. This pattern is in line with the results of Arvind et al. [13], who found that punched RWVGs partially mitigated the pressure drop penalty while improving heat transfer more successfully than non-punched designs. Our findings demonstrate the efficacy of perforations in enhancing thermo-hydraulic efficiency, as in contrast to their study, perforated VGs not only preserve higher heat transfer performance but also offer a more advantageous balance between enhancement and pressure loss. Vortex intensity is not assessed directly in this study. The observed enhanced trends in heat transport are used to infer its influence. It is well known that stronger secondary flows produced by vortex generators with higher attack angles amplify vortex strength, which is reflected in larger Nusselt numbers and thermal performance factors.

3.1. Impact of VG's Angle of Attack on the Coefficient of Convection Heat Transfer

The plane channel without any vortex generator is considered as a baseline case. Addressing RWVGs (rectangular winglet vortex generators) with an attack angle (α) of 30°, 45°, and 60°, Fig. 6 compares the convection heat transfer coefficients (h). As air velocity rises, the convection heat transfer coefficient increases. This is because when air velocity increases, the swirl flow becomes wider and stronger. In all situations, a velocity of 2 m/s results in the largest rise in the convection heat transfer coefficient. The convection heat transfer coefficient increases by 16.33%, 23.67%, and 40.56% at this velocity in the order of attack angle of 30°, 45°, and 60° of RWVGs respectively, compared to the baseline case. This is because as the VGs row increases, more longitudinal vortices (LV) are created. By passing the fluid's bulk from the heated wall to the main flow, the produced LV increases the pace at which heat is transferred. The pace at which heat moves from the hot wall to the fluid is accelerated by this mass transfer and subsequent heat transfer.

The coefficient of convection heat transfer (h) for the CRWVGs with angles of attack of 30°, 45°, and 60° is compared in Fig. 7. Since the centrifugal force weakens when the fluid flows over a concave surface, CRWVGs intensify the longitudinal vortex and produce a better convection heat transfer coefficient than RWVGs. For CRWVGs with a 60° angle of attack and a 2 m/s flow velocity, the convection heat transfer coefficient can reach 78.89% of the baseline case. When CRWVGs are perforated, the flow through the hole is disturbed, which results in the weakening of LV. Similar to what was found in the case of CRWVGs, it is found that the convection heat transfer coefficient in perforated RWVG cases decreases considerably at the same flow velocity. A maximum 2.1% drop in the convection heat transfer coefficient in perforated RWVG cases is seen when compared with cases of CRWVGs. These findings suggest that using perforated RWVGs to lower the coefficient of convection heat transfer is negligible.

3.2. Impact of VG's Angle of Attack on the Pressure Loss Penalty

A drop in pressure for the RWVGs with an attack angles of 30°, 45°, and 60° is displayed in Fig. 8. The coefficient of convection heat transfer can be raised by using VG; however, there will also be a rise in pressure drop. As air velocity and angle of attack increase, pressure drop rises. A drag force addition results in a greater pressure drop as air velocity increases. In the baseline case with a 2 m/s flow velocity, pressure drop rises by 35.37% over the RWVGs with an angle of attack of 60°. In comparison to the baseline channel, the average pressure drop rises by roughly 65% for α = 30°, 110% for α = 45°, and 150% for α = 60° for the whole inlet velocity range.

Fig. 9 shows the decrease in pressure for the CRWVGs with angles of attack of 30°, 45°, and 60°. CRWVGs indicate a larger reduction in pressure than RWVGs, as CRWVGs offer a greater surface area than RWVGs; consequently, the flow resistance increases. The angle of attack of the VG can increase the barrier to flow brought on by the longitudinal vortex, resulting in a higher pressure reduction as compared to the baseline case. Pressure drops increase with increasing angle of attack. The biggest reduction in pressure occurs on CRWVGs with an angle of attack of 60°. The pressure drop increases significantly over the velocity range (0.4–2.0 m/s) when compared to the baseline: roughly 400–580% for α = 30°, 450–630% for α = 45°, and 500–680% for α = 60°. The largest increase is continuously seen at α = 60°.

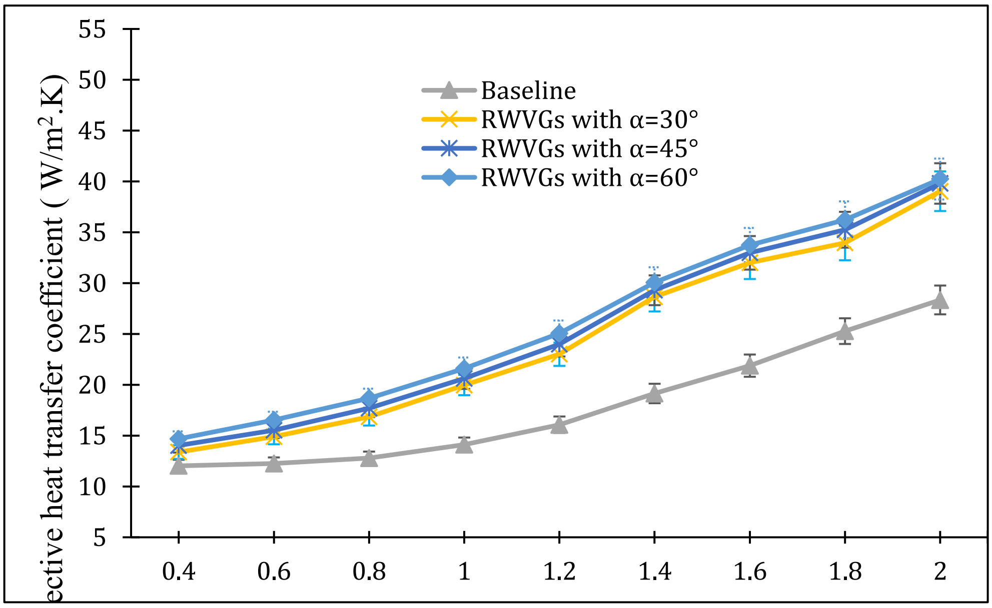


Fig. 6. Coefficient of convection heat transfer comparison between baseline and RWVGs at 30°, 45°, and 60° attack angles

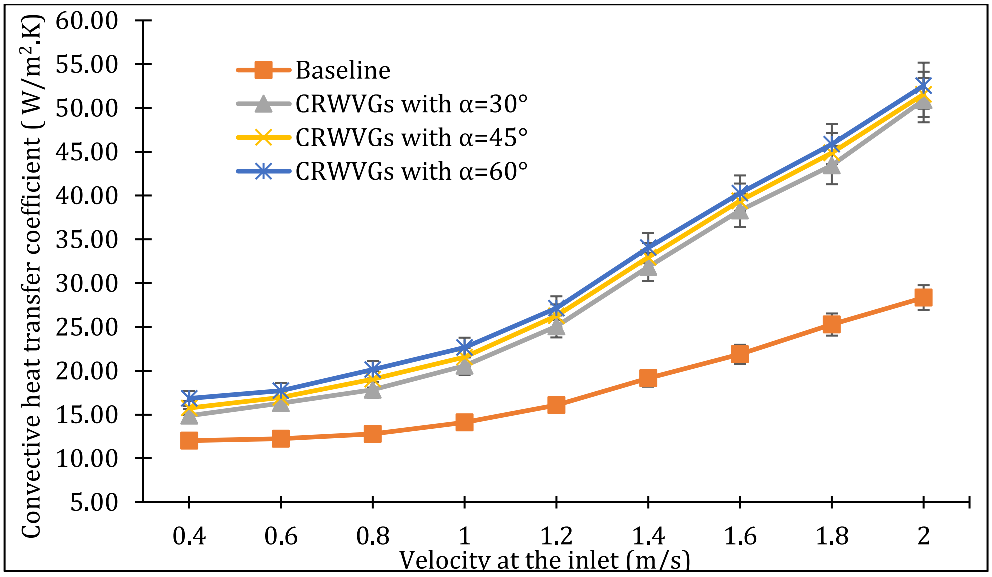


Fig. 7. Coefficient of convection heat transfer comparison between baseline and CRWVGs at 30°, 45°, and 60° attack angles

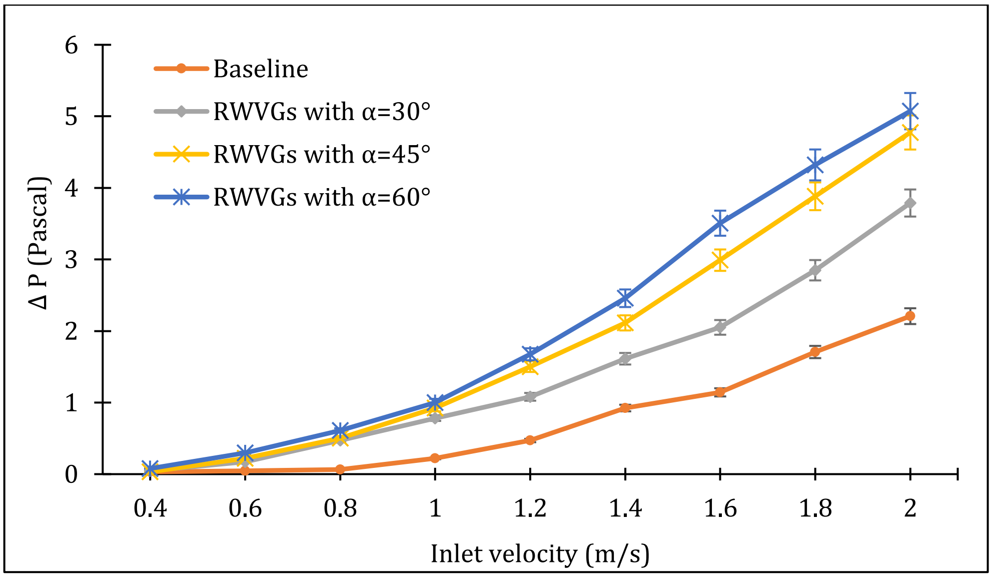


Fig. 8. Comparison of baseline and RWVGs pressure decrease at 30°, 45°, and 60° attack angles

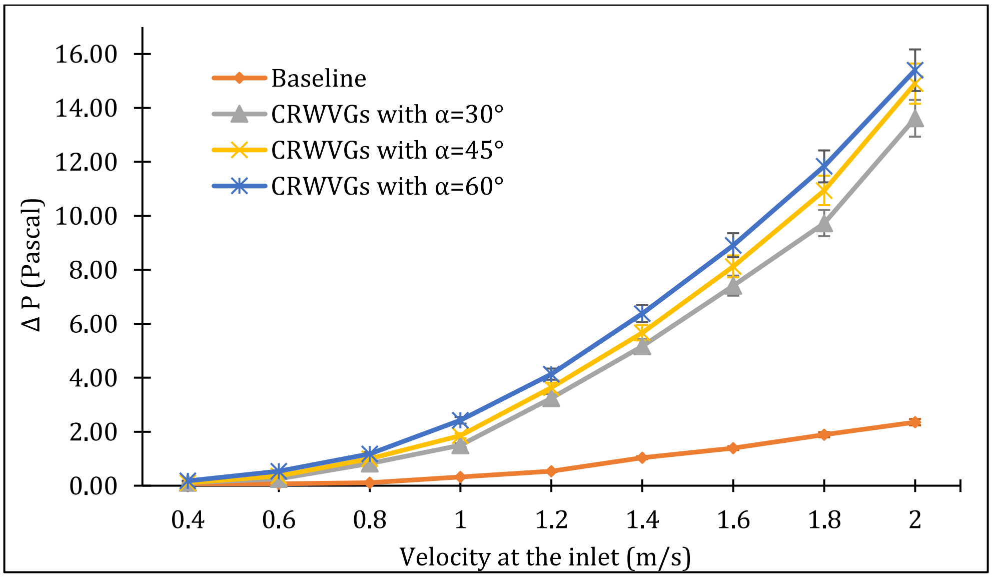


Fig. 9. Comparison of baseline and CRWVGs pressure decrease at 30°, 45°, and 60° attack angles

In Figures 10 and 11, comparison of heat transfer coefficient per pressure drop for RWVGs and CRWVGs at 30°, 45°, and 60° attack angles is shown. RWVGs with a 30° attack angle routinely beat all other designs across the velocity range, according to a quantitative evaluation of the heat transfer coefficient per pressure drop. The performance ratio of RWVGs at 30° reaches approximately 180 at low velocities (0.4–0.8 m/s), which is roughly 35-45% greater than that of RWVGs at 45° and 60% higher than that of CRWVGs at 45°. When contrasted with CRWVGs at 60°, the difference is much more noticeable, with improvements ranging from 60 to 80% in the low-to-moderate velocity range (0.6–1.2 m/s). The advantage of RWVGs at 30° rapidly diminishes with increasing velocity; at 2.0 m/s, the performance ratio values for all configurations converge in the range of 7–12, leaving only a 0–25% difference between them. This makes it quite evident that RWVGs with smaller attack angles are more effective at lower speeds, whereas the relative advantage decreases at higher speeds because pressure drop has a greater effect. The most advantageous trade-off is provided by RWVGs at 30°, which exhibit up to 80% more efficiency than the weakest configuration and consistently outperform the others across the velocity range.

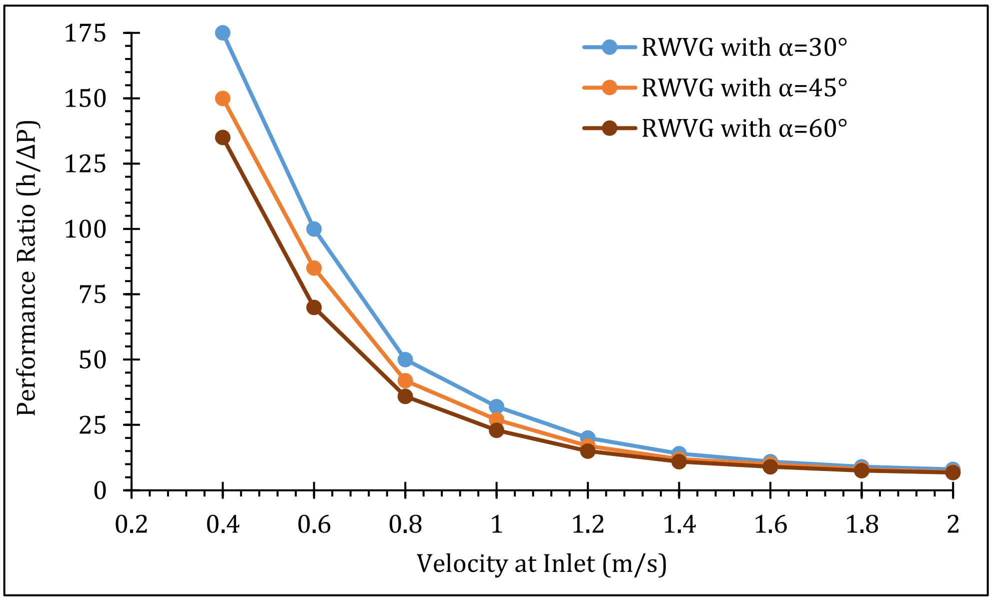


Fig. 10. Comparison of heat transfer coefficient per pressure drop for RWVGs at 30°, 45° and 60° attack angles

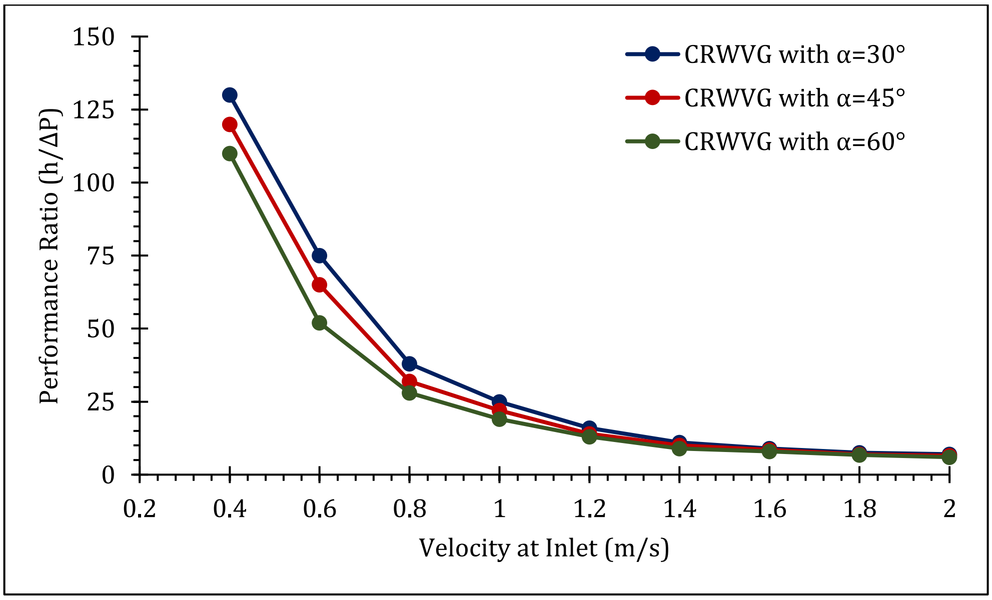


Fig. 11. Comparison of heat transfer coefficient per pressure drop for CRWVGs at 30°, 45° and 60° attack angles

4. Conclusions

- The pressure drop can be increased by increasing the convection heat transfer coefficient with a vortex generator. Compared to CRWVGs, VGs with a higher pressure drop have a higher convective heat transfer coefficient.
- Convection heat transfer coefficients using RWVGs and CRWVGs were compared, and the RWVG resulted in a slight decrease in value. Perforated VGs, however, can lessen the pressure drop. The flow visualization shows that CRWVGs generate a greater LV than RWVGs.
- CRWVGs improved the convection heat transfer coefficient by a significantly greater 78.89% at 60° than RWVGs, which improved it by as much as 40.56%.
- Heat transfer performance was only slightly decreased (by about 2.1%) when holes were added to RWVGs. But because of increased flow resistance, these improvements came with a higher pressure drop, which for RWVGs reached 35.37% and for CRWVGs was even higher at 60°.
- In order to assess total thermo-hydraulic performance, it is necessary to take into account the trade-off in pressure drop, even though CRWVGs offer the most efficient heat transfer improvement.
- Although vortex intensity was not evaluated directly (for example, using vorticity contours or synergy angle), its qualitative impact is clear from the trends that were observed, suggesting that secondary flow strength is a key factor in enhancing heat transfer.

5. Limitations of the Study

- Depending on the hydraulic diameter of the test channel and the fluid characteristics, the current analysis has been conducted over a restricted range of flow velocities (0.4–2.0 m/s), which corresponds to Reynolds numbers roughly in the range of 4000–20,000.
- This suggests that the study focuses on the turbulent flow regime and ignores extremely high Reynolds numbers that might be found in industrial settings as well as very low Reynolds numbers (laminar or transitional flow). As a result, the results are only applicable within this particular set of operating conditions, and extrapolation outside of these bounds should be done carefully. For wider applicability, future research might examine different thermal and geometric parameters and broaden the Reynolds number range.

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Conflicts of Interest

The author declares that there is no conflict of interest regarding the publication of this article.

Authors Contribution Statement

Vaibhav Bansode: Written original manuscript and Review.

Munna Verma: Conceptualization and Data Curation.

Krishna Borate: Resources and Software.

Venkatesh T. Lamani: Methodology and Review.

Shekar Kannaiah: Writing – Original Draft.

Channa Keshava Naik N.: Validation and Visualization.

Hasan Sh. Majdi: Formal Analysis and data curation .

Salah J. Mohammad: Methodology and Review.

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