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Development of a Simplified Vortex Generator using CFD Technique for Optimization of Heat Transfer

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Abstract

Vortex Generators (VGs) are small flow control devices designed to create swirling motions within a fluid flow, thereby enhancing fluid mixing and improving heat transfer efficiency. Due to these advantages, VGs are widely applied in cooling systems and thermal management processes. This study numerically investigates the thermal performance of a trowel wing vortex generator (TWVG) installed within a rectangular duct. The primary objective is to evaluate the influence of TWVG geometrical parameters on heat transfer enhancement and flow resistance. Numerical simulations were performed using ANSYS Fluent 2022 R2, incorporating experimentally obtained boundary conditions at a Reynolds number of 5,000. The analysis focuses on key performance indicators, including the Nusselt number and friction factor, to comprehensively assess thermal and hydraulic performance. The study explores the effects of varying attack angles (α = 5°, 10°, and 15°) and vortex generator lengths (L = 50 mm, 75 mm, and 100 mm) over a Reynolds number range of 5,000 to 30,000. Results indicate that the geometrical configuration of the TWVG plays a crucial role in determining heat transfer enhancement and flow behaviour. The optimal configuration, providing the highest Performance Evaluation Criteria (PEC) value of 1.08, was achieved at an attack angle of 15° and a length of 100 mm. Furthermore, flow visualisation analyses were conducted to understand the interaction between flow structures and convective heat transfer mechanisms.

Major Findings: *The study presents detailed visualisations of Reynolds-averaged velocity fields, vorticity distributions, and surface temperature and Nusselt number profiles, offering valuable insights for the design of efficient thermal systems.*

Keywords: *Nusselt Number (Nu), Performance Evaluation Criteria (PEC), Reynolds Number (Re), Trowel Wing Vortex Generator (TWVG)*

1.0 Introduction

Efficient thermal management is a critical requirement across a wide range of engineering applications, including electronics cooling, aerospace systems, and energy systems. As the demand for compact and high-performance heat exchangers increases, enhancing convective heat transfer without significantly increasing pressure losses has become a key research focus. Passive heat transfer enhancement techniques, which do not require external energy input, have gained popularity due to their simplicity and reliability. Among these techniques, the use of Vortex Generators (VGs) has emerged as an effective approach for promoting fluid mixing and disrupting thermal boundary layers[1-3].

Vortex generators are small flow-altering devices that introduce longitudinal or transverse vortices into the fluid stream. These vortices increase momentum exchange near the surface, thereby enhancing convective heat transfer[4-5]. The effectiveness of VGs, however, is highly dependent on

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their geometrical configuration, placement, and orientation relative to the flow[6-8]. In recent years, novel VG designs have been proposed to optimise heat transfer performance while maintaining manageable pressure drops[9-11].

This study focuses on a Trowel Wing Vortex Generator (TWVG), a modified form of VG designed to improve thermal performance within a rectangular duct under turbulent flow conditions. The objective is to evaluate how varying the attack angle and length of the TWVG influences heat transfer and flow resistance. Numerical simulations are conducted using ANSYS Fluent 2022 R2 over a Reynolds number range of 5,000 to 30,000. Key performance indicators such as the Nusselt number, friction factor, and Performance Evaluation Criterion (PEC) are analysed to determine the optimal geometric configuration. The findings provide insights into the fluid dynamic and thermal behaviour induced by TWVGs and contribute to the development of more effective passive heat transfer enhancement methods[12-16].

2.0 Methodology

2.1 Geometric Details

The computational domain modelled in this study represents a rectangular air duct, a configuration frequently found in compact heat exchanger setups[1,5]. The duct spans 2040 mm in length, 750 mm in width, and 40 mm in height. It is segmented into three distinct regions to replicate realistic flow development and thermal transfer conditions: a 500 mm inlet section allows for the development of the velocity profile; a 540 mm test section houses the vortex generators and serves as the primary region for thermal performance assessment; and a 1000 mm outlet section facilitates flow stabilization and pressure recovery before discharge[11,17].

Trowel Wing Vortex Generators (TWVGs) are installed on the bottom surface of the test section to generate strong longitudinal vortices. These structures promote enhanced mixing and disrupt the thermal boundary layer, leading to improved heat transfer[2,4,6]. Arranged in a staggered configuration, the vortex generators ensure optimal vortex interaction and consistent thermal performance across the duct cross-section[3,10,18]. The layout consists of three columns spanning the duct's width and five rows along its length, totalling 15 TWVGs. Their placement and orientation follow the specifications defined in the CAD model and design documentation[7,9].

Each TWVG is designed with a unique, curved, blade-like profile that resembles a plastering trowel—hence the name. The primary geometric parameters defining each generator are the Attack Angle (α) and the projected Length (L). The angle of attack, which indicates the tilt of the device to the flow direction, is tested at three values: 5°, 10°, and 15°, to analyse how vortex intensity influences heat transfer[8,12,13]. Similarly, the projected length—affecting vortex penetration and size—is examined for lengths of 50 mm, 75 mm, and 100 mm[14,15].

All dimensions, orientations, and placements of the TWVGs are based on finalised CAD models, developed following standard engineering design methodologies. These parameters ensure precision, consistency in simulation, and relevance to actual industrial duct-based heat exchanger systems[16,19,20].

Figure 1 illustrates the 2D views of the TWVG design, including the front, side, and top perspectives. Figure 2 shows the 3D isometric view of a single TWVG, while Figure 3 depicts the 3D isometric view of a vortex plate incorporating an array of TWVGs.

2.2 Boundary conditions

To analyse the impact of the Trowel Wing Vortex Generator (TWVG) on heat transfer characteristics in a rectangular duct, numerical simulations were performed using ANSYS Fluent 2022 R2. The computational domain was subjected to the following boundary conditions:

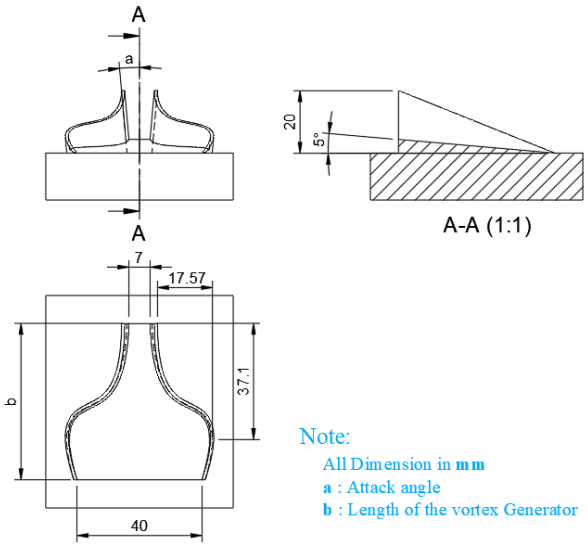


Figure 1. 2D Drawing of the vortex generator.

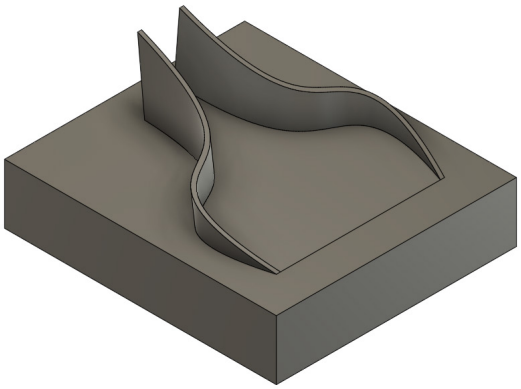


Figure 2. 3D Isometric view of the single vortex generator.

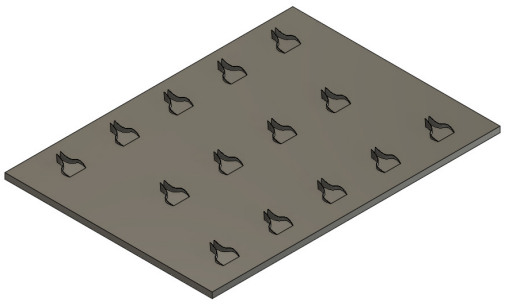


Figure 3. 3D Isometric view of the vortex plate.

- Inlet: A velocity inlet boundary condition was applied, with the inlet velocity varied to achieve Reynolds numbers in the range of 5000 to 30,000. The inlet temperature was fixed at 300 K, and the turbulence intensity was set to 5%, based on the hydraulic diameter of the duct[1,9].
- Outlet: A pressure outlet boundary condition was implemented with the gauge pressure set to 0 Pa, representing atmospheric pressure. The backflow temperature was set to match the inlet temperature to maintain thermal stability[3,17].
- Duct Walls: All walls of the duct were treated as no-slip boundaries. The bottom wall, which was the primary heat-transferring surface, was subjected to a constant heat flux of 500 W/m². The remaining walls were assumed to be adiabatic, ensuring no heat transfer across them[4,5,13].
- Vortex Generator Surface: The surface of the TWVG was modelled as a no-slip wall. It was assumed to be thermally insulated (adiabatic), contributing to flow mixing but not acting as a heat source[6,10].
- Symmetry Conditions: Where applicable, symmetry boundary conditions were employed to reduce computational cost by simulating only a representative section of the geometry without loss of generality[11,14].

These conditions ensured that the flow and thermal fields were accurately captured, facilitating the evaluation of the TWVG's effect on convective heat transfer and pressure drop across the duct[2,12,15].

2.3 Mesh Details

To ensure the accuracy and reliability of the numerical results, a comprehensive mesh independence study was conducted. Multiple computational simulations were carried out with varying mesh densities to evaluate their influence on the predicted nusselt number and friction factor. The objective was to achieve a mesh resolution that ensured the thermal and flow characteristics remained insensitive to further refinement, while also maintaining computational efficiency.

The study involved mesh sizes ranging from 105,361 to 678,255 elements. For each case, the average Nusselt number and friction factor were computed. It was observed that beyond a certain mesh density, the changes in these performance indicators became negligible. Among the tested configurations, the mesh consisting of 325,659 elements was selected for further simulations. This mesh yielded a stable Nusselt number of 17.0329 and a corresponding friction factor of 0.1834, indicating a good compromise between accuracy and computational cost.

Figure 4 presents the variation of Nusselt number and friction factor with respect to mesh density, clearly showing convergence behavior and justifying the choice of the third mesh configuration as optimal for capturing

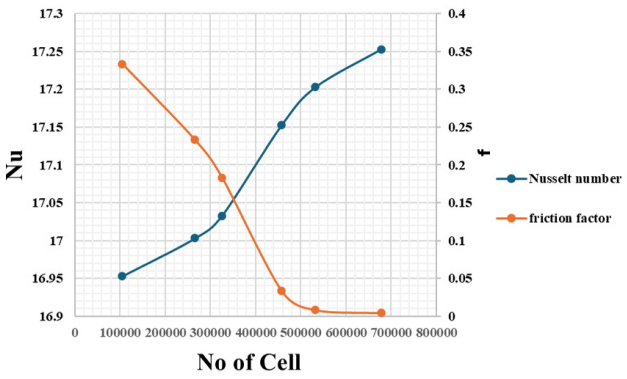


Figure 4. Grid analysis test for vortex generator.

detailed flow and heat transfer characteristics without incurring excessive computational expense.

2.4 Numerical scheme

Steady-state, 3D simulations were performed using the pressure-based solver in ANSYS Fluent 2022 R2. The Realisable k–ε model was used to resolve turbulent effects, especially those induced by the TWVG. SIMPLE was applied for pressure–velocity coupling.

A Second-Order Upwind scheme was chosen for all transport equations to ensure accurate resolution of flow and temperature fields. The finite volume method solved the governing equations for continuity, momentum, energy, and turbulence.

Convergence was based on residuals of 10^{-6} for energy and 10^{-4} for other variables. Nusselt number and pressure drop were monitored for additional validation.

Air was modelled with constant properties: density = 1.177 kg/m³, viscosity = 1.85 × 10^{-5} Pa.s, and thermal conductivity = 0.02624 W/m.K Operating pressure was set to 101325 Pa. A heat flux of 2000 W/m² was applied to the bottom wall, with other walls treated as adiabatic and no-slip.

The simulation modelled the governing equations for continuity, momentum, energy, and turbulence, as detailed in Equation (1) to Equation (5).

$$\text{Continuity equation}: \nabla.\left(\overrightarrow{\rho v}\right)=0 \tag{1}$$

$$\text{Momentum equation}: \overrightarrow{\rho v}.\nabla\vec{v}=\mu\nabla^2\vec{v}-\Delta P \tag{2}$$

$$\text{Energy equation}: \rho C_p\nabla.\left(\vec{v}\,T\right)=\nabla.(k\nabla) \tag{3}$$

$$\text{Turbulence Kinetic Energy}: \frac{\partial}{\partial x_j}\left(\rho k u_j\right)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k+G_b-\rho_e+Y_M \tag{4}$$

$$\text{Rate of Dissipation}: \frac{\partial}{\partial x_j}\left(\rho\varepsilon u_j\right)=\frac{\partial}{\partial x_j}\left[\left(\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right)\right]-\rho C_2\frac{\varepsilon^2}{k+\sqrt{\nu\varepsilon}}+C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b \tag{5}$$

2.5 Calculation Procedure

The Reynolds number (Re) characterises the flow regime by differentiating between laminar, transitional, and turbulent flows. It is defined mathematically as in Equation (6):

$$Re=\frac{\rho V D_h}{\mu} \tag{6}$$

where ρ is the fluid density, V is velocity, D_h is the hydraulic diameter, and μ is the dynamic viscosity.

The Dittus-Boelter equation is an experimentally derived correlation used to determine the Nusselt number (Nu) for convective heat transfer in fully developed turbulent flow within a smooth duct. The Equation (7) is expressed as:

$$N_u=0.023R_e^{0.8}P_r^{0.4} \tag{7}$$

The Gnielinski equation provides a more precise approach for determining the Nusselt number (Nu) in turbulent flow, particularly in situations where the flow has not yet fully developed. As an enhancement of the Dittus-Boelter equation, it accounts for the influence of entrance length and intermediate Reynolds numbers. The Equation (8) is expressed as:

$$N_u=\frac{\left(f/8\right)\left(R_e-1000\right)P_r}{1+12.7\left(f/8\right)^{0.5}\left(P_r^{2/3}-1\right)} \tag{8}$$

The Blasius equation is an empirical relationship used to approximate the Darcy friction factor (f) for smooth turbulent flow within a duct or pipe. It is expressed as in Equation (9):

$$f=0.316R_e^{-0.25} \tag{9}$$

The Petukhov equation is an enhanced empirical correlation used to determine the Darcy friction factor (f) for fully developed turbulent flow in smooth pipes. It is expressed as in Equation (10):

$$f=\left(0.79\ln Re-1.64\right)^{-2} \tag{10}$$

The Bulk temperature (Tb) represents the mass-averaged temperature of the fluid flowing through a duct and is a key parameter in convective heat transfer analysis. It is determined using the following expression as in Equation (11):

$$T_b=\frac{T_{in}-T_{out}}{2} \tag{11}$$

where, T_{in} is the inlet and T_{out} is the outlet temperature of the fluid flowing through the duct.

The convection heat transfer coefficient (h) quantifies heat transfer between a moving fluid and a surface. In smooth ducts, it depends on boundary layer development and flow properties. In contrast, vortex generators enhance mixing and disrupt the boundary layer, altering heat transfer behavior and influencing the calculation of h as in Equation (12).

$$h = \frac{q}{\left(T_s - T_b\right)} \tag{12}$$

where q is the heat flux, T_s is the surface temperature, and T_b It is the bulk temperature.

The Nusselt number (Nu) is a dimensionless quantity that characterises the ratio of convective to conductive heat transfer in a fluid. It is defined by the fundamental Equation (13):

$$N_u = \frac{hD_h}{k} \tag{13}$$

where, D_h is the hydraulic diameter (m).

The Performance Evaluation Criterion (PEC) serves as a measure for assessing the thermal-hydraulic efficiency of heat transfer enhancement methods, including vortex generators. It quantifies the trade-off between heat transfer improvement and the resulting pressure drop, offering a comprehensive evaluation of an enhancement technique's overall effectiveness through Equation (14).

$$PEC = \frac{(N_u / N_{u0})}{\left(f / f_0\right)^{1/3}} \tag{14}$$

3.0 Results

The study investigated the impact of Trowel Wing Vortex Generators (TWVGs) on heat transfer performance in a rectangular duct using CFD simulations. Simulations were conducted for nine different configurations, varying in attack angles and lengths, across a Reynolds number range of 5000 to 30,000. Results showed that all vortex generator designs enhanced the Nusselt number compared to a plain duct, indicating improved convective heat transfer. However, this improvement came with an increase in the friction factor, signifying a higher pressure drop. Among the tested designs, the configuration with a 15° attack angle and 100 mm length yielded the highest Performance Evaluation Criterion (PEC) value of 1.08. This indicates that the specific configuration offered the best balance between heat transfer enhancement and flow resistance. Streamline and temperature contour analysis confirmed stronger mixing and disruption of the thermal boundary layer in ducts with vortex generators. At higher Reynolds numbers, the difference in performance between various configurations became more noticeable. Some designs, although excellent in heat transfer, caused excessive pressure drop, making them less efficient overall. The study concludes that proper optimisation of VG geometry is essential for maximising thermal performance without compromising system efficiency.

3.1 Temperature Contours

Figure 5 presents the temperature contour distributions for various Vortex Generator (VG) configurations in a rectangular duct. The Plain Duct (PD) reveals a pronounced temperature gradient, signifying inefficient heat transfer. In comparison, the VG cases (VG1–VG9) display improved temperature uniformity, attributed to vortex-induced mixing that enhances convective heat transfer. Configurations with denser and more organised vortex structures exhibit superior thermal performance, effectively minimising hot spots and promoting better heat dissipation.

Figure 6 illustrates the velocity streamlines for various Vortex Generator (VG) configurations in a rectangular duct. In the Plain Duct (PD) scenario, the streamlines remain largely parallel, indicating minimal flow disruption and limited mixing. Conversely, in the VG cases (VG1–VG7), the streamlines exhibit notable disturbances caused by the Vortex Generators, generating enhanced turbulence and secondary flow structures. These vortices facilitate greater fluid mixing, effectively disrupting the thermal boundary layer and improving heat transfer. The intensified interaction between the main flow and near-wall regions suggests a significant enhancement in convective heat transfer efficiency compared to the plain duct.

3.2 Nusselt Number (Nu)

The Nusselt number (Nu) indicates convective heat transfer efficiency, increasing with Reynolds number due to enhanced turbulence. Plain ducts show the lowest Nu, while vortex generators—especially VG 4, VG 6,

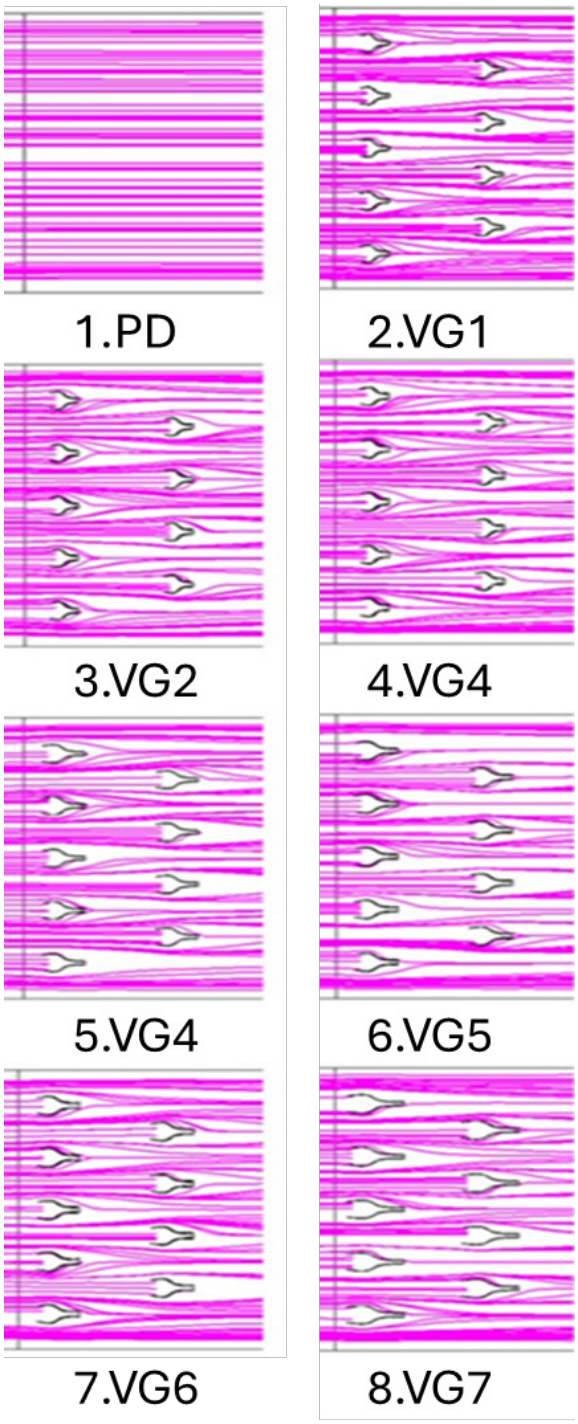


Figure 5. Temperature contour images.

and VG 7—deliver superior heat transfer through strong vortex formation. VG 1 and VG 3 offer moderate gains, and VG 8 and VG 9 balance decent heat transfer with lower friction, highlighting energy efficiency, and the Figure 7 denotes the variation Nusselt number over the Reynolds number.

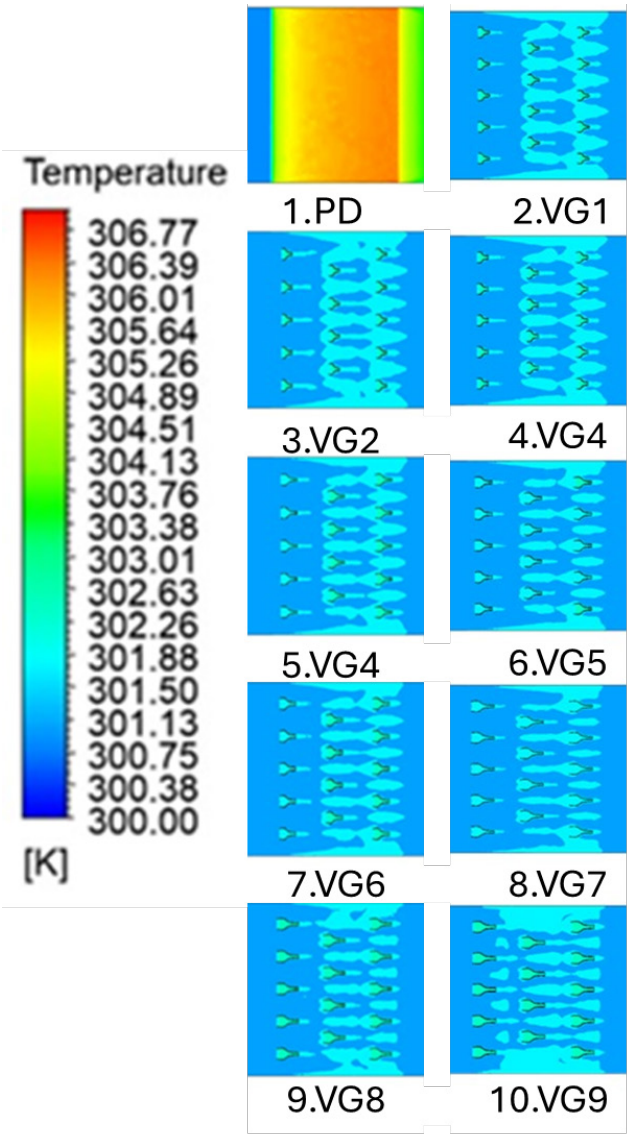


Figure 6. Velocity streamlines.

Overall, vortex generators significantly boost thermal performance, particularly at high Reynolds numbers, though efficiency must be weighed against increased friction.

3.3 Friction Factor (f)

The friction factor (f) is as shown in Figure 8, and it reflects the pressure drop due to vortex generators, increasing with turbulence but reducing as the Reynolds number rises. Plain ducts have the lowest values due to smooth flow, while vortex generators raise friction by disturbing the boundary layer. At low Reynolds numbers, wake and recirculation effects elevate f, with VG 1–3 showing the highest values. In contrast, VG 8 and

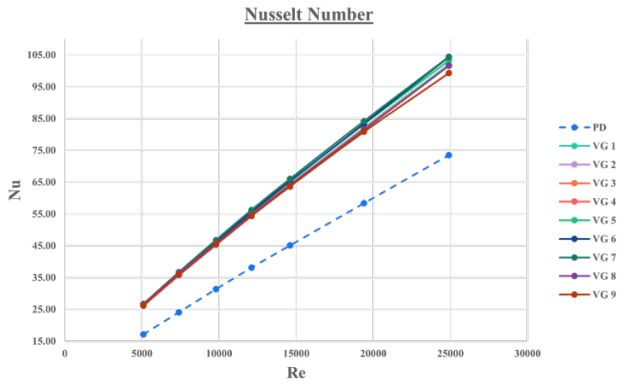


Figure 7. Graph of Nusselt number.

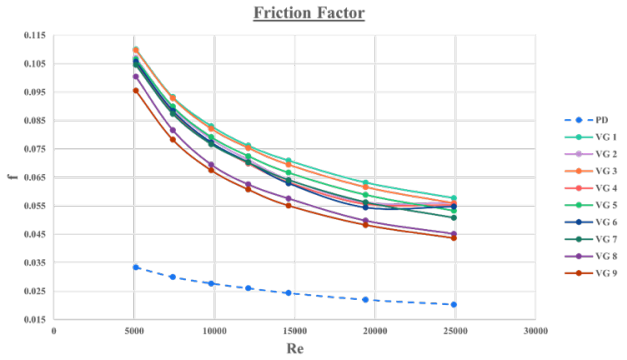


Figure 8. Graph of friction factor.

VG 9 maintain lower friction factors, indicating efficient turbulence generation with minimal resistance. These designs offer a balanced trade-off between heat transfer and aerodynamic efficiency.

3.4 PEC

The Performance Evaluation Criterion (PEC) is shown in Figure 9, and it assesses the efficiency of vortex generators by balancing heat transfer improvement against pressure drop. As the Reynolds number increases, PEC initially rises and then stabilises. VG 8 and VG 9 achieve the highest PEC values at high Reynolds numbers, making them the most efficient designs. VG 1 and VG 3 have lower PEC values, indicating greater pressure losses relative to their heat transfer gains. Although VG 4, VG 6, and VG 7 offer higher Nusselt numbers, their efficiency is reduced by significant flow resistance. Thus, VG 8 and VG 9 stand out as the most practical options, offering optimal thermal performance with minimal pressure loss.

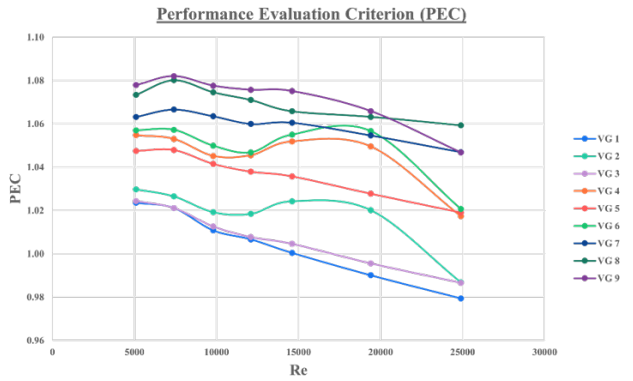


Figure 9. Graph of Performance Evaluation Criterion (PEC).

4.0 Conclusion

In conclusion, vortex generators significantly enhance convective heat transfer by promoting turbulence, with designs like VG 8 and VG 9 offering the best balance of performance. These vortex generators provide superior heat transfer with minimal pressure drop, as indicated by high Nusselt numbers and optimal Performance Evaluation Criterion (PEC) values. While VG 1, VG 3, VG 4, VG 6, and VG 7 show improvements in heat transfer, their higher pressure drop or lower PEC values make them less efficient overall. Therefore, VG 8 and VG 9 emerge as the most practical and energy-efficient choices for real-world applications, offering the best trade-off between heat transfer and flow resistance.

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