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Review Article

Air-Side Heat Transfer Enhancement Using Vortex Generators on Heat Transfer Surfaces- A Comprehensive Review

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ABSTRACT

The main purpose of this article is to provide a critical analysis of published research on these heat transfer surfaces. Important experimental methods and numerical procedures are explained, and many types of vortex generators are described. The phenomenon of flow attributed to vortex generators mounted, connected, pierced, or placed inside surfaces that transmit heat was also examined. In addition, recommendations for applying vortex generator (VGs) technology to improve air-side heat transfer are provided, as well as information on the thermal performance of newly proposed VG heat transfer surfaces. The performance of air-side heating surfaces can often be significantly improved through the use of vortex generators. However, their effectiveness can be greatly affected by many factors, including fluid flow rate, pipe geometry (diameter, shape, pitch, in-line or staggered configuration), fin type, and geometry of the vortex generator (height, length, shape, angle of attack, etc.). Circular fin-tube heat exchangers generally perform worse in terms of thermal-hydraulic efficiency than flat-tube-fin and oval-tube-fin heat-exchanging devices, and more recently, suggested vortex generators. Most current heat exchanger optimization methods focus only on thermal-hydraulic performance.

1. Introduction

In modern heat exchangers, thermal energy is frequently transferred between a moving fluid and its enclosing channel. To enhance this heat transfer, various methods are employed, including increasing the fluid's contact area with the channel wall, disrupting fluid flow to improve

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circulation, and inducing turbulence. By incorporating these methods, the size of heat exchangers can be reduced while maintaining performance, thus minimizing the need for costly operational fluids and mitigating potential safety issues related to system fluid volumes. Heat transfer can be improved using both active and passive techniques. Active methods, such as Electrohydrodynamics (EHD), Magnetohydrodynamics (MHD), and mechanical motion, require external power to enhance heat transfer. In contrast, passive methods like dimples, fins, and tape inserts rely on surface modifications and do not need external energy input. Although active technologies are more complex and costly to implement than passive ones, they offer the advantage of precise control over heat transfer enhancements. Since Joule's seminal 1861 paper on heat transfer, significant research has focused on enhancing heat exchanger performance [1]. Air-cooled heat exchangers, commonly used in various applications, face challenges due to air's lower thermal conductivity compared to liquids or two-phase flows, resulting in airside resistance often accounting for 85% or more of the total heat transfer resistance.

To address this, a prevalent method for improving performance involves employing surfaces with periodic interruptions in the flow direction, which helps achieve high efficiency and cost-effectiveness [2,3]. Heat exchangers, which transfer energy between multiple fluids at different temperatures, are widely used across industries such as industrial logistics, climate control, cooling, and chemicals [4]. Typical vortex generators (VGs) and heat transfer surfaces, including fins, ribs, wings, offset fins, offset strip fins, and rectangular plate fins, aim to enhance heat transfer. Their effectiveness is largely due to their ability to reduce secondary flow formations, such as twists and vortices, by increasing turbulence in the boundary layer [5-7]. Interrupted surfaces in heat exchangers, despite their significant pressure-drop penalty, considerably improve heat transfer performance. Compared to louvered or strip-finned surfaces that disrupt the main flow, secondary stream interruption surfaces with vortex generators (VGs) achieve comparable low-pressure drops while offering substantially better heat transfer enhancements [6–9]. These surfaces are particularly useful in industrial thermal energy recovery systems, where high-pressure internal fluids and high-temperature, low-pressure exhaust gases interact with them [10–11]. VGs can create complex flow patterns over heat-conducting surfaces, enhancing heat transfer through various scales and geometric features [12]. Understanding the thermohydraulic characteristics of surfaces equipped with VGs is essential for evaluating their effectiveness and practicality in improving airside heat transfer for specific applications. This study offers a comprehensive comparison of earlier research concerning the thermohydraulic performance of heat transfer surfaces equipped with various vortex generators (VGs). Its goal is to provide a systematic analysis of how these vortex generators enhance airside heat transfer by inducing vortices that improve thermal mixing and reduce boundary layer thickness.

2. Vortex Generators

Heat exchangers can be categorized based on their construction, including regenerators, heat exchangers with enlarged surfaces, and tubular heat exchangers [4]. Among these, finned-tube and plate-fin heat exchangers are the most common extended surface types.

To improve heat transfer rates, vortex generators (VGs) are often added to extended fin surfaces. VGs disrupt the thermal boundary layer between the heat exchanger surface and the secondary fluid, enhancing heat transfer. The design and integration of VGs vary by application, leading to two main types of vortices: transverse and longitudinal. Longitudinal vortices generally offer better heat transfer with only a slight increase in pressure drop [13]. Four common designs for surface protrusions that enhance heat transfer through longitudinal vortices are rectangular winglet, delta winglet, rectangular wing, and delta wing [8, 12, 14], as shown in Figure 1.

The geometrical attributes of vortex generators (VGs), such as aspect ratio (L) and angle of attack (α), significantly impact their heat transfer performance, making them an important area of scientific research. Along with the primary four categories, various VGs are currently recommended and researched.

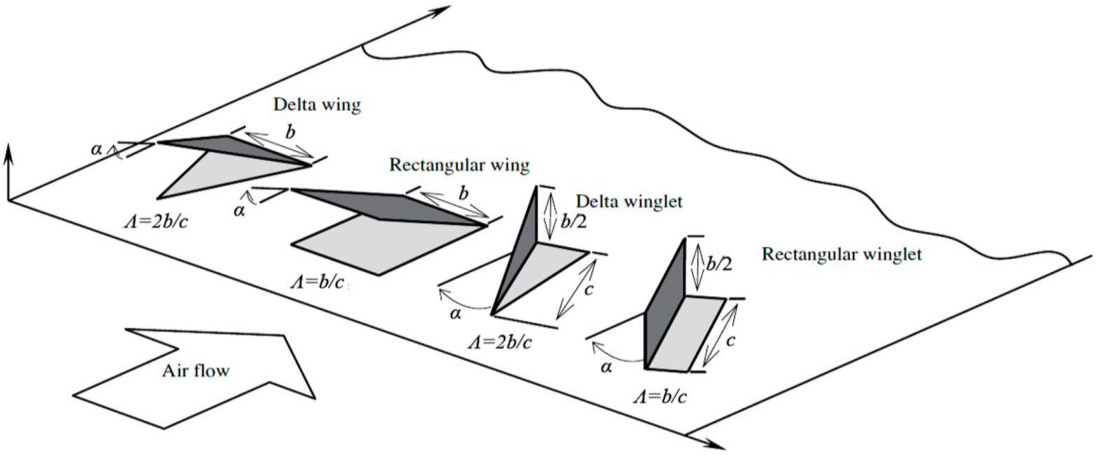

Fig. 1. Typical VGs and the corresponding geometric descriptions [8 , 12 , 14]

Wang et al. [6, 9], Lin et al. [15], and Gong et al. [16] designed wave-element and annular winglets VGs, which are shown in Figure 2a, specifically for use with finned tube heat exchangers. He and colleagues [17] suggested V-deployed VG arrays, as seen in Figure 2b, to mimic animal mobility in the wild. Zhou et al. [18, 19] investigated several types of straight and curved surface winglet vortex generators (VGs), some with punched holes. These included: Delta winglet, Curved delta winglet, Trapezoidal winglet, Rectangular winglet, Curved trapezoidal winglet, Curved rectangular winglet. These VG designs were explored to understand their impact on heat transfer enhancement, as illustrated in Figure 2c. Lotfi et al. (references [20, 21]) examined the applicability of four distinct vortex generator (VG) types in heat exchangers. These VG types included: Angle rectangle winglet, Curved angle rectangular winglet, Rectangular trapezoidal winglet, and Wheeler wishbone on smooth wavy finned oval tube. These studies, illustrated in Figure 2d, aimed to assess how these VG configurations impact heat transfer enhancement in practical applications. Figure 2e illustrates a composite fin design by Wu et al. [22] that incorporates VGs on slit fins. All of the recently proposed VGs have improved the thermohydraulic performance of thermal exchangers. Figure 2f's innovative concepts for longitudinal tube fins, which are proposed by Stehlk et al. [10] and to enhance heat transfer, it is crucial to strengthen the surface of heat transmission and increase flow turbulence, is particularly noteworthy. In power plants, when fin shape and augmentation may successfully reduce heat exchanger size and costs, these designs may be employed for heat recovery.

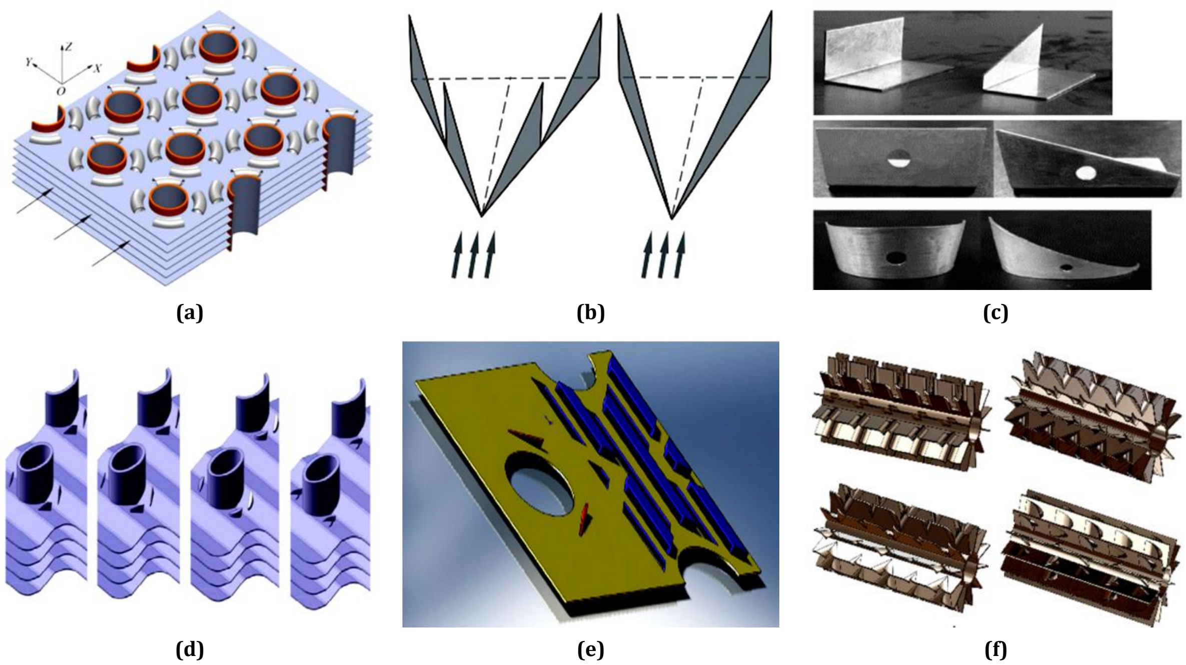

Fig. 2. Latest suggested VGs: (a) Wave-element and annular winglet Vertex genertors[6, 9, 15, 16]; (b) V-deployed array of VG [17]; (c) Curved winglet VGs with or without punched holes [18, 19]; (d) Oval tube bank and wavy fin with VGs attached [20, 21]; (e) Slit fin and composite fin with VGs [22]; (f) Fins on tubes that are longitudinal by Stehlík et al. [10]

3. Experimentative and Computational Techniques

3.1. Method of Experimentation

The researchers should be knowledgeable about measuring techniques and data reduction strategies because experimentation is so crucial in engineering and science. Accurate measurements of temperature, pressure and flow rate are typically needed for evaluating thermohydraulic performance. Among the experimental techniques used to look into the thermohydraulic efficiency of the airside heat exchange surfaces are laser doppler velocimetry (LDV), laser light sheets (LLS), particle image velocimetry (PIV), liquid crystal thermography (LCT), hot-wire anemometry, the naphthalene sublimation technique, and infrared thermography.

The illustrative experiment methods for temperature, pressure and flow rate measurements are included in Table 1. Whichever tube type heat exchanger is used, these experiment procedures frequently focus on the extended fin surfaces. Since one methodology may be used for a variety of heat exchange surfaces and VGs, these strategies are not provided individually depending on either tube or VG kinds. Focusing on the measurements of temperature variation and flow visualisations as they are crucial to a deeper comprehension of the connections between flow and heat exchnge, we want to illustrate the experimental methodologies and generalised experimental systems in this section. Chen and Shu [38] used Laser Doppler Velocimetry (LDV) to analyze axial vortices and axial mean velocity near the bottom wall of a duct with a heated plate. They measured three-component mean and fluctuation velocities. Upstream of a smoke wire, a device producing sub-micron particles was installed. Axial vorticity directed towards the wall was calculated from averaged component velocities. Flow patterns were measured at four cross-sectional planes and two planes perpendicular to them vertically to determine these parameters. Hernon and Patten [49] performed hot-wire tests to obtain velocity profiles near a pair of delta winglets (DWP) installed on a flat, unheated surface. They used a TSI IFA300 system with a steady-temperature anemometer to measure both mean and fluctuating velocities, allowing for the determination of boundary layer profiles and thickness. The authors emphasized that hot-wire measurements offer superior temporal and spatial precision, providing a more detailed exploration of key flow characteristics and a deeper understanding of flow and heat transfer processes compared to other measurement techniques. Min et al. [53] utilized Particle Image Velocimetry (PIV) to visualize flow on a heated flat plate, employing glycerol particles and a charged coupled device (CCD) camera. The glycerol particle generator produced particles with a high concentration and approximately 1- micron diameter to ensure adequate illumination for the CCD camera to capture the visualization region. Fiebig et al. [24, 27, 29] and Tiggelbeck et al. [25] employed Transient Liquid Crystal Thermography (LCT) to investigate the influence of longitudinal vortices on local heat transfer coefficients. By coating the heat transfer surface with a thin layer of thermochromic liquid crystals, which refract laser light differently depending on temperature levels, they could observe the formation of specific isotherms during transient heating. The relationships between the swirl and boundary layer were investigated by Gentry and Jacobi [30], Yoo et al. [36], and Shi et al. [43] using naphthalene sublimation procedures. Naphthalene solidified and polished into a portion of the plate surface during the experiment. The whole Plate's streamwise length can have a naphthalene surface that forms a narrow lip at the preceding and leaving edges. In order to get the local and average mass transfer coefficients and the local and average coefficients of heat transfer using the mass and heat analogies, the depths of local sublimation were determined using a non-contact optical method called laser triangulation. In order to assess heat transport, O'Brien et al. [40] used a transient approach in which the test section was quickly filled with warm air, and an imaging infrared camera (FLIR PRISM DS) was used to record excellent quality temperature variations on the local fin surface. Infrared thermography was also used by Leu et al. [39], Min et al. [53], and Aris et al. [14] to measure surface temperature distributions. In comparison to thermochromic liquid crystals, this approach offers several benefits., including the ability to operate over a wide temperature range, the ability to obtain excellent thermal and high spatial resolution, and the ability to use full field direct digital data acquisition and processing.

Table 1. Illustrational experimental heat transfer surface studies for airside thermohydraulic performance with VGs

Bibliography	Vortex Generators (VG's)	Reynolds Number	Approach	Testing
Garimella and Eibeck [23]	Horizontal channel with two distinct delta wings in half	700-5200	Heat transfer is facilitated by a system of 30 heated copper components positioned in six spanwise rows on the removable hatch.	Pressure reduction and improved heat transmission
Fiebig et al. [24]	Flat plate with rectangle and delta wings	1360-2270	Flow visualisation using a laser light sheet, and calculating the local heat transfer coefficients using unsteady liquid crystal thermography	Drag coefficient, Colburn j-factor, flow pattern, local heat transfer coefficient, and normalised improvement of heat transmission.
Tiggelbeck et al. [25,26]	On a flat plate, half-delta wings are arrayed in single and double rows.	1600-8000	In order to assess local heat transfer, a liquid crystal thermography is utilised. Glycerine evaporating tracer particles are also employed to investigate the flow structure and see the flow field.	Drag coefficient, local Nusselt number, average Nusselt number, flow pattern, and vortex characteristic.
Fiebig et al. [27]	DWPs in a exchanger of heat with a circular tube finned with fins.	600–2700	Local heat transfer is measured using liquid crystal thermography, and axial velocity is measured using a hot wire anemometer at a intervals of 2 mm.	Distribution of Nusselt number, local Nusselt number average, Nusselt number, and apparent friction factor.
Tiggelbeck et al. [28]	DWPs and RWPs on flat plate with a delta wing and a rectangular wing.	2000–9000	To measure the local heat transmission on the wall, use thermochromic liquid crystal thermography.	Local Nusselt number.
Fiebig et al. [29]	DWPs in flat tube and circular finned heat exchangers.	600 - 3000	For monitoring local heat transmission, use of liquid crystal thermography.	Nusselt number distribution, Nusselt number average , Nusselt number locally and apparent friction coefficient.
Gentry and Jacobi [30]	Flat plate of wings of delta.	600, 800, 1000	An experiment using naphthalene sublimation to determine convection coefficients, and a laser light sheet for visualising flows.	Drag coefficient, Sherwood number.
Kotcioglu et al. [31]	Flat plate with RWPs.	3000-30000	A smoke producer was utilised in a Hele-Shaw device to see laminar main flow.	Friction factor, flow pattern and Nu number average.
Torii et al. [32–34]	DWPs in thermal exchanger with a circular tube with fins.	350–2100	At the test portion's input, a heating screen made of stainless steel ribbons was evenly distributed throughout a cross section to fast and uniformly heat the flow.	Colburn j-factor, friction factor.
Gentry and Jacobi [35]	Flat plate with delta wings.	300–2000	A potential flow model with flow visualisation and a vane-type vortex metres to gauge vortex strength. naphthalene sublimation tests to yield convection coefficients.	Heat transfer enhancement, the distribution of Sherwood number, Sherwood number average and the penalty in pressure drop.
Yoo et al. [36]	RWPs in heat exchanger with a finned circular tube.	800–4500	Using the naphthalene sublimation method, local mass transfer coefficients can be measured and heat transfer coefficients may be calculated using an analogy equation between mass and heat transfer.	Distribution of Nusselt number, Nusselt number local, average Nusselt number, and apparent friction factor.
Yuan et al. [37]	Flat plate with RWPs .	5000–47000	To assess the local heat transfer coefficients, 25 rows of copper-constantan thermocouples are installed evenly in the flow direction.	Average Nusselt number, and apparent Darcy friction factor correlations.
Chen and Shu [38]	Flat plate with delta wings.	4430–11820	The flow pattern were described and the three component mean and fluctuation velocities were calculated using laser	Average Nusselt number, near-wall flow characteristics, and velocity structure.

			doppler velocimetry. The surface temperature was measured using twenty flow thermocouples that were positioned in the flow direction.	
Leu et al. [39]	RWPs within a heat exchanger with a finned circular tube.	400–3000	A precise hot-wire apparatus is utilised to measure inflow velocity, an infrared thermal camera is used to detect temperature distribution, and to examine the flow, a dye injection approach is employed.	Temperature field, pressure drop, average coefficient of heat transfer and numerical distribution.
O'Brien et al. [40]	DWPs in a heat exchanger with an oval tube and fins.	600–6500	The local distributions of the temperatures of the fin surfaces and the airflow rates are measured, respectively, using inline precision mass flowmeter and a precision imaging infrared camera.	Distribution of heat transfer coefficients, average Nusselt number and friction coefficient.
Sommers and Jacobi [41]	a heat exchanger with delta wings and a circular tube fin design.	500 - 1300	Hot bulb anemometry is used to detect the air flow velocity, and the temperature and humidity of the air are regulated via a controlled steam injection system and an upstream cooling coil..	Total temperature resistance, penalty for pressure loss, Fanning friction factor, Colburn j-factor.
Pesteei et al. [42]	DWPs with a single cylindrical barrier in parallel plates.	2250	To measure the regional temperature distribution, a total of 23 thermocouples were installed on the fin surface in half.	The apparent friction factor, average Nusselt number and the local heat transfer coefficient.
Shi et al. [43]	DWPs in a heat exchanger with a finned flat tube.	< 3000	To demonstrate heat-mass transfer, use the naphthalene sublimation method.	Local Nu number and friction factor
Allison and Dally [44]	DWPs in heat exchanger with a finned circular tube.	2600, 3400, 4600	Using a digital video camera and the dye-injection method, the flow was visualised.	Fanning friction factor, Colburn j- factor and JF factor.
Wang et al. [45]	Flat plate with RWPs.	3000–20000	Working fluid is deionized water.	Friction factor and average Nu number correlations.
Joardar and Jacobi [46]	DWPs in a thermal exchanger with a finned circular tube.	220–960	An upstream cooling coil is used in conjunction with a static mixer near the exhaust of the fan and a centrifugal mixer for ensuring a well mixed, flow with homogeneous temperature.	Thermal resistance, pressure drop, the Colburn j-factor and average heat transfer coefficients.
Tang et al. [47,48]	Heat exchanger in circular shape with mixed fins and DWPs.	4000–10,000	A steam-air system is utilised to accomplish steam to air heat exchange. A Pitot-tube metre is connected to either a U-tube water column manometer or an inclined draught gauge in order to measure the air flow rate.	Comparison criteria for heat transfer performance, average Nusselt number, friction factor, and relationships between friction factor and Nusselt number.
Hernon and Patten [49]	Flat plates with DWPs.	3000-4200	A set of anemometers with a constant temperature for measuring mean and fluctuation velocities.	Local velocity, averaged over time, and boundary layer thickness.
He et al. [17]	a heat exchanger with a finned circular tube with V type DWPs.	1400–3400	Using a second, 12- junction thermocouple grid and a 6- junction, equal-spaced grid of thermocouples, respectively, temperature of the air at intake and behind of the sample were determined.	Pressure gradient, Fanning friction factor, area and volume goodness factors, heat transfer coefficient mean, and heat transfer coefficient all play a role.
Yang et al. [50]	Flat plate with delta wings, semicircular wings, triangular wings, and dimple wings.	120–600	The effects of flow maldistribution are avoided and reduced by using a mixer, an air straightener equaliser, and numerous nozzle code testers to assess the air flow rate.	Inverse Graetz number, Colburn j-factor, friction factor, mean heat transfer coefficient and pressure gradient.
Promvonge et al. [51, 52]	Winglets and ribs working together to form a triangle ribbed canal.	5000–22000	A mixer, an air straightener equaliser, and a multiple nozzle code tester based on the ASHRAE 41.2 standard prevent and reduce the impact of flow maldistribution.	Friction factor and average Nu are performance assessment parameters.

Min et al. [53]	RWPs with flat plate.	5000–17500	An infrared image camera to create glycerol particles for the flow visualization, a hot wire anemometer to measure the channel's entrance velocity, and 54 thermocouples installed at the channel's exits to gauge the mean outlet temperature.	Local Nusselt number , secondary flow distribution , average Nusselt number and friction factor.
Aris et al. [54]	a circular heat exchanger with fins and delta wings.	330–960	After the fan, an orifice device is used to measure the air mass flow rate.	Darcy friction coefficient and average Nu number.
Wu and Tao [55]	A flat plate with punched delta wing.	500-2000	Sixteen thermocouples in a thermocouple mesh to measure the distribution of the exit temperature, and eight thermocouples in a thermocouple mesh to measure the distribution of the almost uniform input air temperature.	Distribution of temperatures, average Nu number.
Wu et al. [56]	DWPs within a cutting-edge finned circular tube heat exchanger.	680-1200	a device that measures air flow rate using nozzles.	Heat transfer coefficient average and pressure gradient.
Wang et al. [57]	Pairs of semi-dimple winglets in a circular tube heat exchanger with plain or louvre fins.	1480-2000	The effects of flow maldistribution are avoided and reduced by using a mixer, an air straightener equaliser, and numerous nozzle code testers to assess the air flow rate.	Pressure gradient and heat transfer coefficient are typical.
Abdelatie et al. [58]	RWPs in heat exchanger made of wing-shaped tubes.	1850–9700	Eight thermocouples are positioned on two grids to monitor the average air temperatures input and output, and an alcohol thermometer with a wet wick-encircled bulb is used to measure the temperatures of the wet bulb at the entry and exit.	Drag coefficient and Nusselt numbers are average.
Wu et al. [59]	Curved DWPs in a heat exchanger with a finned circular tube.	500–4200	At the entry and exit, there are eight thermocouples on two grids to measure the air entering and leaving the system, as well as an alcohol thermometer with a wet wick- encircled bulb to gauge the temperature of the wet bulb.	Average heat transfer coefficient average and pressure gradient, average Nusselt number and friction factor, and correlations for Nusselt number average and Darcy friction factor.

3.2. Numerical Techniques

Numerical modeling, by solving mathematical equations, is a valuable tool for studying heat transfer and fluid flow. It reduces the need for extensive experiments, offering a reliable and efficient means to analyze the interactions between heat transfer and fluid flow, design novel vortex generators (VGs), and optimize designs before prototype development [60, 61].

This section outlines these numerical methods without specifying the type of heat exchanger, as the same techniques can be applied to various surfaces. Key numerical features include the model, assumptions, discretization, boundary conditions, solver, and result presentation. Table 1 presents a chronological summary of the primary numerical approaches used in published studies. STAR-CD, CFX solver, modified marker-and-cell (MAC) algorithm, SOLA algorithm, and FLUENT solver are a few of the computational solvers. Since the 1960s, time-dependent incompressible fluid flow issues have been solved using the well-known MAC approach. The MAC algorithm was updated by Biswas et al. [64,67,70] to solve the laws regulating fluid flow and heat transfer. The SOLA algorithm, which is based on the MAC approach and used by Fiebig et al. [62] and Zhu et al. [65,66] to determine the temperature gradient and velocity pattern on heat transfer surfaces, is a finite-difference technique for evaluating an incompressible fluid's Navier-Stokes equations. The temperature profile and velocity pattern in wavy fins with various VGs were examined by Lotfi et al. [21] using CFX, which exhibits exceptional precision, durability, and speed for rotating equipment. FLUENT is extensively used in computational simulation and is regarded as a highly potent CFD tool. It features well-validated physical models and is capable of giving quick, accurate results for multiphysics applications. As indicated in Table 1, FLUENT software has been used in the majority of published CFD investigations during the past ten years, including References [4,16,48].

In the numerical studies listed in Table 1, the assumptions are typically Newtonian and incompressible, with constant fluid characteristics, three-dimensional in the context of the computing domain, and with little force of buoyancy and dissipation of viscosity. To the best of the authors' understanding, these presumptions hold true for low-pressure flows with small temperature changes. While certain engineering challenges do not deviate from the assumptions of incompressible fluid, constant characteristics, and little buoyancy force, others do include high-pressure flow and wide temperature variations. In certain situations, the numerical modelling should use thermophysical properties that rely on temperature or even the real-gas thermodynamic property model while taking gravity into account. Additionally, for particularly compressible flows, the temperature distribution at the heat transfer surface may differ significantly from that of the bulk flow; in this situation, the heating brought by the dissipation of viscosity should be taken into account. As seen in Table 1, most studies utilise periodic fully formed conditions for fluid flow at the flow route's intake and exit, whereas others assume at the computational domain's inlet, the velocity and temperature are both constant. A geometrical element with periodic or symmetrical boundary constraints is typically utilised as the simulation domain to streamline the physical model and reduce processing time. The symmetry boundary condition is employed in computing geometries and flow patterns that have mirror symmetry, whilst the periodic boundary condition is used for those that have a periodic repeating nature. Both of these boundary conditions may greatly minimise the amount of needed simulation period. As far as the author's knowledge, it is crucial to critically check the derived solution for periodic and symmetric boundary conditions since it can introduce atypical correlations. The local conservation principle has been effectively applied in simulations using the finite-volume methodology to apply numerical techniques to solve the governing equations. As seen in Table 1, the discretization of the diffusion terms is always done using the second-order central difference scheme, that of the convection terms is usually done using the third-order QUICK scheme, and that of the coupling between pressure and velocity is usually done using SIMPLE or SIMPLEC algorithms. The ratio of the convective term to the diffusive term, accuracy, robustness, and other factors come into play when deciding on whether discretization scheme to choose. The power scheme works for intermediate values, while the QUICK and second-order upwind schemes are suitable for all convective to diffusive component ratios. The second-order upwind scheme can sometimes be outperformed by the power and QUICK schemes, albeit at the sacrifice of numerical stability. In order to calculate the momentum equation, the SIMPLE method uses a beginning approximation for pressure and velocity. However, this initial guess may not be accurate, and as a result, the resultant velocities may not meet continuity.

4. Thermal-hydraulic Performances

Since the 1980s, research has focused heavily on the interactions between the flow pattern, heat transfer, and friction drag brought on the finned-tube and plate-fin heat exchangers by VGs. The sample experimental and numerical published research, taking into account the kind of Heat-transfer surfaces and VGs, Reynolds number range, important numerical or experimental techniques, and the observed features, are summarised in Tables 1 , respectively. The thermohydraulic performance indicated in Tables 1 based on various types of heat-transferring surfaces, such as flat plates and heat exchangers with finned circular tubes, is discussed in greater detail in this section. The type of heat transfer surface that regulates the flow passage of the mainstream has a significant impact on the thermohydraulic performance of heat exchangers. Additionally, we concentrate on the interplay of heat and flow transmission, the impact of geometrical factors, and the effectiveness of recently developed VGs for each type of heat transfer surface. It should be noted that the Reynolds number is determined by the channel height or, in the case of finned-tube heat exchangers, the distance between two fins or, in the case of plate-fin heat exchangers, the distance between two plates.

4.1. Flat Plates with Vortex Generators

4.1.1. Flow and Heat Transfer Interaction

Fiebig et al. [24,62] examined the drag and heat exchange increase of winglet pairs or delta wings between flat plates for Reynolds values ranging from 500 to 2270 for laminar flow and heat transfer. It was discovered that every pair of wings or winglets produced two opposing longitudinal vortices at their leading edges. However, the winglet wake was distinguished by large shear layers near the surface, but the linked wing is unable to produce a trailing edge wake. This is the main area where the flow fields of the wing and winglet pair vary. It was discovered that the drag was unaffected by the VG geometry and Re number. A fin with and without VGs had a heat transfer ratio that was unaffected by the Re number from an up to 60. The area ratio Ra > 50 (Ra = Ac/Av, where Av is the vortex-generator area and Ac is the area of the channel wall) enabled local heat transfer augmentation with a mean value of up to 50%. The delta wing had the greatest improvement in heat transmission per unit area of a vortex generator, followed by DWPs and RWPs. Laminar flows induced by delta winglets or DWPs were quantitatively explored by Biswas et al. [68–70] in terms of heat transmission and flow pattern. The following flow pattern was mentioned: the corner vortex had a horseshoe-like distinctive feature, while the detached flow at the front edge of the winglet formed the primary vortex. It was discovered that these vortices caused the flow to swirl in the mainline direction around the axis, greatly boosting the blending of the cold and hot liquids and, as a result, increasing heat transfer but decreasing fluid friction. Tiggelbeck et al. [25,26] examined a channel with one or two rows of DWPs with up to 8000 Reynolds numbers for turbulent flow and heat transmission. In an aligned two-row setup, the flow configuration in the wake of the second row was substantially the same as that of the initial row. The largest local transmission of heat increase was obtained after the second row of winglets for spacing between rows of 7-10 height of the channel. The localised improvement in heat transfer, normalised in the second row's wake, was substantially depending on how closely the two rows were spaced. Lower heat transfer enhancement was seen for a staggered double-row design than for an aligned arrangement, especially for angles near the critical values of 70o for the first row and 55o for the second.

4.1.2. Effect of Geometry Parameters

Wu and Tao [55] investigated the impacts of four various angles, 150, 300, 450, and 600, for DWP on channel flow heat transfer for Reynolds values in the 500 to 2000 range. In comparison to a simple plate without DWP, it was discovered that the average Nu number rose as α. The heat and flow transmission properties of the built-in delta wing in conjunction with or without a hole projecting from the bottom wall were studied by Biswas et al. [63,64], who discovered that the punched hole decreased the intensity of the longitudinal vortices. Additionally, they discovered that while the friction pressure drop dramatically lowered, the increase in the coefficient of heat transfer was quite modest when in comparison to a scenario without any punched holes. Delta wing studies were conducted in a rectangular duct by Aris et al. [14] to examine the effects of angle α. In order to improve heat transfer at higher temperatures and reduce losses in flow pressure at lower temperatures, alloys with shape memory were used to make the wings they had and fabricated using a selective laser melting method. The angle α changed from 100 to 380 as the surface temperature changed from 20 0C to 65 0C. Using the single as well as double wings in their active positions can increase heat transmission by up to 90% and 80%, respectively. For the single as well as double VGs, the losses in flow pressure over the test section rose when the wings were activated by 7% and by 63% of the losses at their de-activated locations, respectively.

Enhancement as well as the losses in flow experienced by the four fundamental types of VGs, including the delta wing, rectangle wing, DWP, and RWP, in the Reynolds number range of 2000 - 9000 and for angles between 30° to 90°. There was an ideal angle for maximal heat transfer for each test VG. However the flow losses rose monotonically with the angle. The drag coefficient b and average Nu number both grew monotonically at greater rates.

4.1.3. Functionality of Newly Suggested Vortex Generators

Figure 2c illustrates a pair of curved trapezoid winglets (CTWPs) that were experimentally tested for thermohydraulic performance, and their results were compared to those of conventional RWPs, DWPs, and TWPs. Due to their streamlined design and lower pressure gradient, DWPs performed best in the laminar and transitional flow areas, whereas CTWPs performed best in the entire turbulent region, according to a comparison of three dimensionless factors (j/j0, f/f0 and (j/j0)/(f/f 0)). Narrower angle α, greater curvature, and greater angle of inclination provided greater efficiency under the investigated circumstances, according to parametric research on CTWPs. For Re values ranging from 650 to 21,000, Zhou and Feng [19] investigated the functionality of rectangular, trapezoidal, and delta winglet VGs with and without punched holes. In both laminar and turbulent flow areas, it was discovered that the curved winglet-type VGs outperformed the comparable plane winglet VGs in terms of heat transfer enhancement and flow resistance. When taking into account all flow zones, the curved DWPs and CTWPs both had the best performance. In all situations, the flow resistance was reduced, and the performance of the VGs was actually increased by the punched holes, but the ideal hole diameter had to match the size of the VG face area.

4.2. Finned Circular-Tube Heat Exchangers with Vortex Generators

4.2.1. Interaction between Fluid Movement and Heat Transfer

In the Re number range 600-2700, Fiebig et al. [27,29] investigated fin-tube heat exchangers' thermohydraulic performance in relation to DWPs. Since the previous row's wake was separated by a low-speed flow, for the inline layout without VGs, the consecutive peaks in front of the second and third tubes grew smaller and flatter. The second peak for the staggered layout without VGs was much greater than the first, where the staggered tubes directed the flow into the wake of the preceding row, generating a horseshoe-shaped vortex and a larger heat transfer peak. The third peak caused by the horseshoe vortex region was smaller than that of the second row since the third tube row came after the first row. For Re values ranging from 500 to 1000, Biswas et al. [67,75] quantitatively examined the heat transmission and flow pattern augmentation using DWPs. In the recirculation zone, which had low-velocity fluid downstream of the circular tube in the absence of VGs, heat transfer was quite limited. However the wake region's heat transmission was significantly amplified there by the existence of DWPs. They said the powerful swirling motion produced by the streamwise longitudinal vortices behind the DWPs and the nozzle-like flow passages were the causes of the improvement. The flow passageways resembled nozzles, which encouraged acceleration and eliminated the region of inefficient transmission of heat from the nearby wake. The thermal boundary layer's expansion was hampered by the fluid layers mixing due to the whirling motion.

4.2.2. Effect of Geometry Parameters

For Re values between 330 and 960, Aris et al.'s [54] investigation looked at the thermohydraulic results of delta wing VGs attached to the surface of the fin. TiNi shape memory alloy served as the material for the delicate wings. This alloy has the ability to alter its attack angle in response to surface temperature. In comparison to the values of the plain fin surface, the highest rise in flow pressure gradient of 15% and a heat transfer enhancement of up to 37% were recorded. The best augmentation effects were attained by an inline and staggered arrangement of thin, punched-out wings with angle α = 14 and ratio L = 4, respectively.

In both the with and without a pair of inclined block-shape VGs three-row plate-fin and tube heat exchangers, Leu et al. [39] examined the impact of span angles (an angle of VG incidence to the streamwise direction) on heat transfer and fluid flow for Re values ranging from 400 to 3000. It was discovered that the inclined block-shape VGs with b = 45 arrangement offered the significantly improved relative heat transfer over the Reynolds range, with the Colburn factor rising from 8% to 30%, yet the friction factor is only increasing by 11–15%. For low and intermediate Re values, it was discovered that the inclined block VG heat transfer improvement was more beneficial.

A comparison of the airside performance of plain, louvered, and semi-dimpled VGs in fin-tube heat exchangers was conducted by Wang et al. [57]. They evaluated the fin pitch, which ranged from 1.6 to 2.0 mm, and furthermore the quantity of tube rows, which were 1, 2, and 4. The heat transfer coefficient for louvre fin geometry was frequently greater than that of semi-dimple VG and plain-fin geometry for all tube rows, with the exception of conditions with one tube row, larger fin pitches, and slower frontal velocities. Rising velocity caused the disparity to widen, whereas a bigger fin pitch caused the difference to narrow. The semi-dimple VG charged the wider gap with a somewhat efficient swirling action. The three-dimensional square channel results show that for both inline and staggered designs, the recirculation zone behind the ribs grows as the Reynolds number increases. The inline attached ribs have a stronger recirculation zone than the staggered attached ribs [75].

4.2.3. Effectiveness of Recently Proposed Vortex Generators

The flow depiction and frictional outcomes for annular and delta winglet VGs in expanded fin-tube heat exchangers with Reynolds values between 500-3000 were given by Wang et al. [6,9]. Two longitudinal vortices developed behind the tube in the presence of annular VGs, and the intensity of the counter-rotating vortices grew, considering the annular height. The horseshoe vortices and other flow streams might swirl with the longitudinal vortices due to their extreme strength. The delta winglet was shown to exhibit more vorticial motion and flow instability than the annular winglet for the same winglet height, resulting in a better blending phenomenon.

The staggered circular-tube bank fin heat exchanger with interrupted annular groove VGs (shown in Figure 2a) was quantitatively studied for its average fluid flow and heat transfer properties by Lin et al. [15] for Re values ranging from 600 to 2500. Under the same pumping power settings, the obstructed annular VG surface could not significantly boost heat transfer at lower Re values, but at higher Re numbers, exceptional performance was possible. The observed Re number range saw an increase in the mean friction coefficient of up to 35%, while the average Nu number boosted by 10% to 40%. For Re values in the range of 800-3000, Gong et al.'s [16] numerical modelling of the heat transfer and flow of fluids of Rectangle Vortex Generators (RVGs) with punched-curve fin-tube heat exchangers was conducted. At various locations, it was discovered that the intensity of secondary flow, average Nu number, and friction factor were higher than those of the reference plain fin.

5. Concluding Remarks

The type of VG, angle of attack, attack angle ratio, and tube configurations significantly affect performance. Winglets often outperform wings, and delta pairs tend to be more effective than rectangular pairs. Maximizing the angle of attack and attack angle ratio generally improves heat transfer enhancement. Most research has concentrated on VGs in finned flat and circular tube heat exchangers, with less attention given to finned oval tubes.

Recent studies suggest that advanced VG designs, such as four-cornered rectangular wings, intermittent annular VGs, V-deployed VG arrays, and others, offer improved thermohydraulic performance compared to traditional VGs. Integrating punched holes into VGs enhances performance by reducing flow resistance. Optimal results depend on matching the hole diameter with the VG face area to achieve aerodynamic goals. Channels with CTWPs show better performance than those with TWPs, with increased Nu numbers and global friction factors compared to smooth channels. The best angle of attack for vortex generators is found to be 45° [83]. For an L/D ratio of 15, as the radius of the cold outlet is increased, the mass flow rate fraction decreases from 0.8 to 0.7 and then to 0.6 for n=1, from 0.65 to 0.58 and then to 0.52 for n=3, and from 0.42 to 0.32 and then to 0.24 for n=5 [84]. Fins in phase change materials can reduce melting time by up to 100 minutes and increase the hot water outlet temperature. Vortex generators impact the charging process by extending the charging time [85]. Based on the performance evaluation parameter ((Nu/Nu0)/(f/f0)), channels using CTWPs showed superior overall performance. Specifically, the average Nu number increased by 6–8% and 9–12%, and the global friction factor increased by 24–29% and 38–48% with CTWPs and TWPs, respectively, compared to a smooth channel.

Key findings:

Winglets generally outperform traditional wings, and delta pairs are more effective than rectangular pairs. Integrating punched holes into VGs can improve performance by reducing flow resistance. Optimal performance is achieved when the hole diameter is matched appropriately with the VG face area. Channels with Continuous Tangential Wing Pairs (CTWPs) exhibit better performance compared to those with Tangential Wing Pairs (TWPs), showing increased Nusselt numbers and global friction factors compared to smooth channels.

6. Scope for Future Work

Based on studies on enhancing airside heat transfer with vortex generators (VGs), several key recommendations for future research have emerged. Finned flat-tube and finned oval-tube heat exchangers with VGs are expected to have lower airside pressure drops and better heat transfer coefficients than finned circular-tube heat exchangers. Therefore, future research should focus on experimental and simulation studies of these two types across a wider range of parameters. Developing universally applicable design correlations for optimizing heat exchangers with VGs on extended surfaces is crucial. These correlations should carefully address geometry parameters and data reduction methods. Before adopting specific correlations, designers and engineers should evaluate detailed data on the heat transfer surface geometry, operational conditions, and the correlation's applicability and limitations. Further research is needed to understand how enhanced heat transfer surfaces affect overall system size and lifecycle costs. This comprehensive approach is vital for advancing heat exchanger technology in engineering applications. Explore how the surface roughness and material properties of heat transfer surfaces interact with vortex generators. Investigate the effects of different coatings, textures, and materials on vortex formation and heat transfer enhancement. The future scope of research in air-side heat transfer enhancement using vortex generators is rich with opportunities for innovation and improvement. By focusing on design optimization, advanced validation techniques, application-specific studies, material interactions, energy efficiency, and technological integration, researchers can significantly advance the field and contribute to more efficient and effective heat transfer systems.

Importance:

As energy efficiency becomes increasingly important, understanding the environmental impact of vortex generators is crucial for their adoption and implementation. The interaction between vortex generators and surface properties can significantly influence heat transfer performance. Understanding these interactions can lead to better design guidelines and material choices.

Potential Impact:

Research in this area could lead to the development of advanced materials and surface treatments that optimize the performance of vortex generators, leading to more efficient heat exchange systems. Combining vortex generators with cutting-edge technologies could create innovative solutions that push the boundaries of heat transfer enhancement and lead to new applications and markets.

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Conflicts of Interest

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

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