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ARTICLE

Investigation on the Effect of Length and Amplitude of Sinusoidal Wavy Vortex Generators on the Heat Transfer Rate, Pressure Drop, and London Factor in Compact Heat Exchangers

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ABSTRACT

Compact heat exchangers (CHE) improve the heat transfer rate with lighter weight and lower volume than other counterparts. An important point in CHEs is their higher pressure drop relative to conventional heat exchangers. This study aims to investigate the heat transfer rate and pressure drop in some proposed models of these heat exchangers with/without vortex generators (VGs) in different cases. A hot fluid of temperature 350 °C flowing through tubes and a cold fluid of temperature 300 °C circulating inside the shell is assumed. To this end, several VGs with sinusoidal wavy shapes are designed and examined with different amplitudes of the sine wave and different lengths to determine the effects of these parameters on the heat transfer rate of tubes and pressure drop along the heat exchanger length. In the 2D steady-state laminar fluid flow, governing equations are discretized using the finite element method and analyzed for Reynolds numbers 400 to 1000 in the ANSYS software. Finally, with a 5.06% increase in the Nusselt number, the sinusoidal VGs of amplitude 1 and length 6 mm quantitatively indicated the best performance in terms of the heat transfer rate and pressure drop (London factor) among the studied cases.

Keywords: Pressure drop; Shell-and-tube heat exchanger; Sinusoidal wavy VG; London factor

1. Introduction

Shell-and-tube heat exchangers are among the most widely-used heat exchangers in the industry, and their optimal performance is a matter of impor-

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tance. In this regard, wide research has been conducted on these heat exchangers in past years. In the recent two decades, with the emergence of VGs, the efficiency of heat exchangers has been improved impressively due to increased heat transfer rate and decreased pressure drop [1]. These studies are generally divided into experimental and numerical.

In 1987, Žkauskas [2] experimentally investigated the fin-and-tube heat exchangers and repeated experiments for different Reynolds numbers to observe the variation of the Nusselt number (Nu) and define a relationship between these two parameters. It should be noted that no VG was considered in Žkauskas's model, and only different arrangements and numbers of tubes were examined in this model.

After the experimental work of Žkauskas [2], many numerical studies were carried out in this field, most of which are based on Žkauskas's model by adding VGs to enhance the heat transfer rate and reduce the pressure drop. In 2004, Leu et al. [3] studied the block-shaped VGs experimentally and numerically and, finally, selected 45° as the best angle of attack in fin and tube heat exchangers.

Gholami et al. [4] presented a numerical study where three VGs, including flat, wavy with upward concavity, and wavy with downward concavity, were used in the heat exchanger. Then, each VG was examined individually. Finally, results indicated that wavy VGs have a significant effect on the heat exchanger performance. He et al. [5] conducted valuable research work in this field. They investigated the effects of winglet-type VGs in three steps as follows: 1) Under inclination angles of 10, 20, and 30 degrees, 2) Different numbers of VGs, and 3) Positioning and arrangement of VGs, inline and staggered. Finally, the results indicated that VGs with staggered placement and inclination angle of 10 degrees outperform other arrangements.

In 2019, three studies are conducted in this field. Modi and Rathod [6] studied the effect of the different common flow-down (CFD) configurations of wavy-up, wavy-down, curved-up, curved-down, and flat rectangular winglets for VG. Finally, the wavy-up configuration presented the best heat transfer rate, and the curved-down one provided the lowest pressure drop.

Awais and Bhuiyan [7] used the staggered arrangement for tubes of the heat exchanger and VGs to evaluate the heat exchanger performance in these conditions. They examine the flat VG at two different angles of attack on time in the beginning part of the tube and another time in the end part of the tube. In addition, they placed oval and square tubes in the shell of the heat exchanger and compared the obtained results. At last, the staggered arrangement of circular and oval tubes delivered the best results among other cases.

Lu and Zhai [8] studied the staggered arrangement of tubes and curved VGs with different values of the angle of attack and curvature. Finally, the VG with the greatest curvature results in better heat transfer relative to others.

Wang et al. [9] investigated the effect of VGs on the thermal performance of elliptical fin-and-tube heat exchangers under different inclination angles of 15 to 75 degrees and Reynolds numbers of 1300 to 2100. The main result of this study was that the friction and Colburn factors increased by increasing the inclination angle of the VG. This study examined the effect of secondary flow intensity on the heat transfer mechanism to acquire a better understanding of this parameter. From examining the secondary flow, it was observed that the secondary flow was complementary to the traditional method of heat transfer performance analysis.

Sahel et al. [10] investigated fin-and-tube heat exchangers in two sections: the first section examined the effects of the inclination angle of the tube between 0-90 degrees, and the second section optimized the best case determined in the first section by placing flat VGs in the heat exchanger. Finally, the optimization was conducted by changing the number and position of VGs on the upper region of the tubes. The obtained results of the first section showed that the inclination angle of 20° ensured a 1-4% increase in the heat transfer coefficient compared to the baseline case (Inclination angle of 0°).

Caliskan et al. [11] investigated the impact of punched triangular longitudinal vortex generators and sinusoidal wavy plates on convective heat transfer using a variety of approaches. The experiments were performed in a rectangular duct with flat plate-punched triangular VGs and sinusoidal wavy plate-punched triangular VGs under different Re numbers (10000, 20000, 30000, and 50000), four different wavelengths (40 mm, 50 mm, 60 mm, and 70 mm), and attack angles of punched winglet (15°, 30°, 45°, and 75°). Finally, the highest thermal performance factor was determined for a wavelength of 50 mm and an attack angle of 30°. Also, it was observed that the Reynolds number has the greatest effect on Nu.

Naik et al. [12] conducted a 3D numerical simulation for investigating the performance of a curved rectangular winglet VG with concave and convex shapes in a rectangular channel. This study examines the effect of altering the arc angle of curved winglet VGs on the flow and thermal characteristics.

DL Hu [13] investigated the effect of different transverse locations of the concave curved vortex generators on thermal performance.

KW Song [14] studied the effect of the concave curved vortex generator pairs with different transverse locations and attack angles on the thermal-hydraulic characteristics of the heat exchanger.

W Luo [15] investigated three novel structural designs based on wavy-louvred (WL) fin and vortex generators.

BAA Shlash [16] investigated the flow and heat transfer of turbulent fluid flow through channels with various vortex generator designs mathematically (triangular, half-circle, and quarter-circle).

AÜ Tepe [17] surveyed the heat transfer performance of circular-slice-shaped-winglet (*CSSW*) for tube bank heat exchangers (*TBHE*).

TT Göksu [18] presented an experimental study, in twelve different double-sided curved blade vortex generators were produced at two blade angles (15° and 30°) based on the turbulator blade angle, blade direction, and the placement of the turbulators in the pipe.

XL Zhong [19] examined the cases of inverted flags with different thicknesses in a channel flow.

S Akcay [20] numerically investigated the effects on the hydraulic and thermal performance of pulsating flow in a periodically corrugated channel with discrete V-type winglets.

Y Wang [21] studied a novel type of tube inserts, symmetrical wing longitudinal swirl generators (SWLSGs) and its thermal hydraulic performance was numerically investigated under laminar flow.

A Berber [22] surveyed the effects of the curved winglet vortex generator inserts (CWVGs) on the forced convection heat transfer, friction factor, and performance evaluation criteria (PEC) interior flow.

S Suman [23] investigated the effect of different shapes and positions of a wavy flag vortex generator on the heat transfer in a rectangular channel using Computational Fluid Dynamics (CFD) analysis.

J He [24] numerically studied the effect of a V-shaped protrusion vortex generator on flow and heat transfer of impingement cooling in crossflow.

M Sheikholeslami and M Nimafar [25] examined the impact of melting heat transfer on nanofluid flow in the presence of Lorentz forces, they also considered different shapes of nanoparticles.

According to research previously conducted on VGs, it can be concluded that sinusoidal wavy VGs have better efficiency and performance than flat ones. Indeed, the novelty of the present study is the design and analysis of sinusoidal VGs, which have not been considered sufficiently in the literature. Therefore, the main goal of this study is to select the best case of positioning a sinusoidal wavy VF in a shell-and-tube heat exchanger to achieve the highest heat transfer rate and lowest pressure drop. To this end, the sinusoidal wave is considered for designing VGs, which are examined with different amplitudes and lengths in the following.

2. Problem statement

As explained in the introduction section, the heat exchanger or converter is a device that is used to transfer heat between two fluids. One of the most conventional heat exchangers is the shell-and-tube type having many applications in the industry. So, investigation of these heat exchangers in terms of heat transfer rate and pressure drop for minimizing related costs is a hot topic in the industry. This study aims to investigate the heat transfer in the compact type of these heat exchangers. For this purpose, a shell-and-tube heat exchanger is considered, as shown in **Figure 1**.

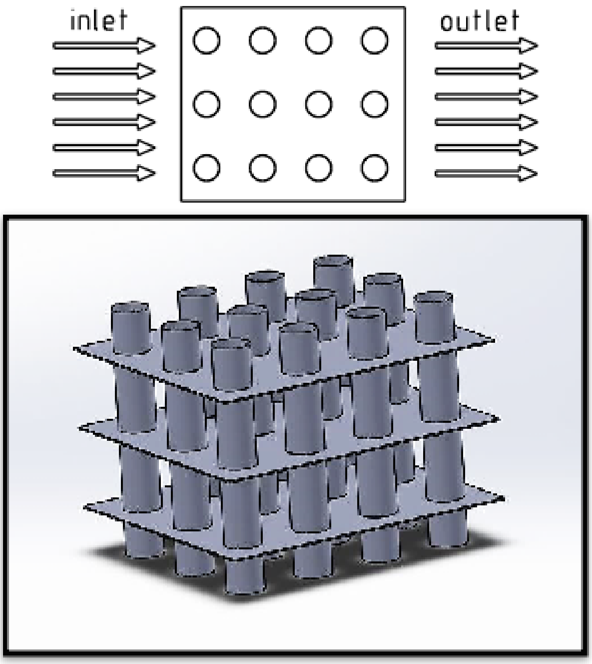


Figure 1. General schematic of shell-and-tube heat exchanger without VG.

This heat exchanger transfers the heat from the hot fluid inside tubes (constant temperature of 350 K) to the cold fluid (air) inside the shell, which results in the fluid cooling inside tubes. The fluids used in this study are incompressible with constant thermodynamic properties. The fluid flow regime in the shell is considered laminar and steady-state due to the low velocity at the inlet section of the heat exchanger.

As mentioned in the introduction section, the presence of VGs in the shell can perturb the cold fluid flow (air) and increase the heat transfer rate. On the other hand, increasing the heat transfer rate causes an increase in the pressure drop along the shell length, which will show itself as an additional cost and energy consumption for the intended project. Therefore, the present study aims to design a series of VGs to achieve acceptable values of heat transfer (maximum) and pressure drop (minimum). Regarding the dimension and size of the heat exchanger in **Figure 2**, an initial model (computational domain) of the heat exchanger (without VG) is produced in 2D space; because of transverse symmetry, the model is divided into two parts of width 12.7 mm, and only one is used for computations. In the next steps, by analyzing this initial model, the results obtained in this study can be compared with the results of other studies, including Gholami et al. [4] and Žkauskas [2], to ensure the validity of the solution method.

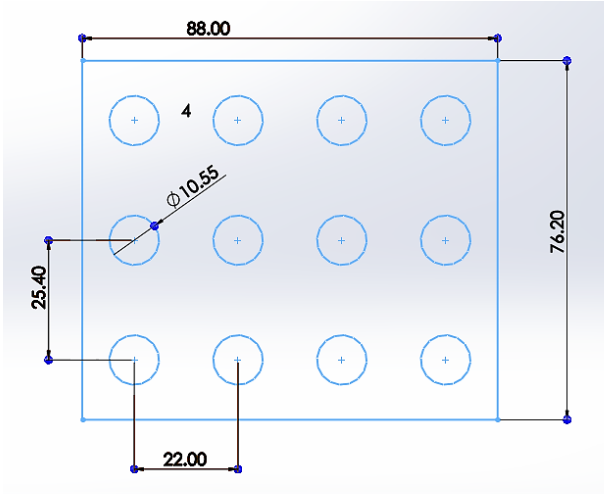


Figure 2. Heat exchanger dimensions from the top view (mm).

2.1 Boundary conditions

In this study, the boundary conditions used in the domain are as follows: Velocity inlet at the cold fluid inlet with the temperature of 300 K, pressure outlet at the cold fluid outlet, constant temperature boundary condition (350 K) at walls of four tubes, wall boundary condition for cases with VG, and two symmetry boundary conditions. All boundary conditions are presented in **Figure 3**.

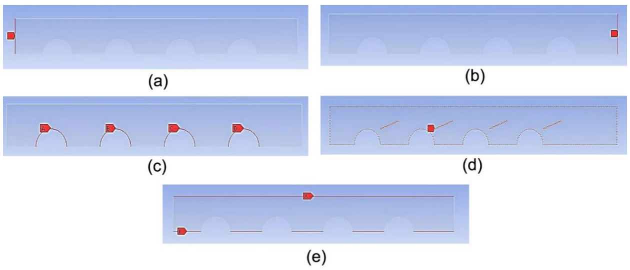


Figure 3. (a) Cold fluid outlet boundary condition, (b) Cold fluid outlet boundary condition, (c) Constant temperature wall boundary condition for tubes, (d) Wall boundary condition for VGs, and (e) Up and down symmetry boundary conditions.

2.2 Governing equations

The numerical solutions of mass and momentum conservation equations are addressed in the analysis of this problem. For this purpose, different methods can be used, including finite difference, finite volume, and finite element. This problem is analyzed using the finite volume method in ANSYS software. Like other heat transfer and fluid dynamics problems, the momentum, mass conservation, and energy equations in the incompressible flow are important in this problem. These equations are expressed as follows:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{3}$$

$$k\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right)=\rho C_p\left(\frac{\partial (uT)}{\partial x}+\frac{\partial (vT)}{\partial x}\right) \tag{4}$$

Other parameters are also used in this problem, from which Reynolds number, London factor, friction factor, and pressure drop can be referred:

$$Re=\frac{\rho U_m D_H}{\mu} \tag{5}$$

where ρ is the cold fluid density (air), U_m is the cold fluid velocity at the minimum surface area, μ is the dynamic viscosity of the cold fluid, and D_H is the hydraulic diameter considered equal to the tube diameter in this problem. Another important equation is the relationship of the Nusselt number in the heat exchanger defining the ratio of the convection to conduction heat transfer in the thermal boundary layer, which is defined as follows:

$$Nu=\frac{hD_H}{k} \tag{6}$$

where h is the average convective heat transfer coefficient, and k is the thermal conductivity. The pressure drop along the heat exchanger length is given by Equation (7).

$$\Delta P=P_{in}-P_{out} \tag{7}$$

where P_{in} and P_{out} are the average fluid pressures in the inlet and outlet, respectively. A dimensionless number, named friction factor, can be obtained from Equation (8) using this pressure drop.

$$f=\frac{\Delta P}{(0.5\rho U_m^2)} \tag{8}$$

where ΔP is the pressure drop along the heat exchanger length in Pa. Now, another dimensionless number can be defined using this equation. Equation (9) defines the London factor, which is a dimensionless number for measuring the overall performance of the heat exchanger in terms of heat transfer rate and pressure drop:

$$London\,factor={}^{j}/_{f} \tag{9}$$

$$j=st.pr^{2/3} \tag{10}$$

$$St=\frac{h}{\rho U_m C_p} \tag{11}$$

$$Pr=\frac{C_p\mu}{k} \tag{12}$$

In Equation (9), j is the Colburn number, St is the Stanton number, and Pr is the Prandtl number, which are important dimensionless numbers in the heat transfer subject (Equations (10) and (11)). Also, Equation (12), C_p denotes the specific heat capacity.

2.3 VG dimensions

This study utilizes some VGs designed using sine functions, as shown in **Figure 4**. The thickness of all VGs is 0.2 mm (t = 0.2 mm), and they are designed according to Equation (13).

$$y=Asin\left(\frac{2\pi x}{d}\right) \tag{13}$$

where values of A, d, and θ are presented in **Figure 4** and **Table 1**.

As specified in **Table 1**, VGs with the attack angle of 30°, three different lengths (6, 8, and 10 mm), and five different amplitudes result in 15 different models in this problem. The goal is to investigate the effect of attack angle and different lengths on other parameters of the problem, including Nusselt number, pressure drop, and London factor. It should be noted that these VGs are installed in the point R = (x, y) relative to the center of tubes, where R denotes the radius of tubes.

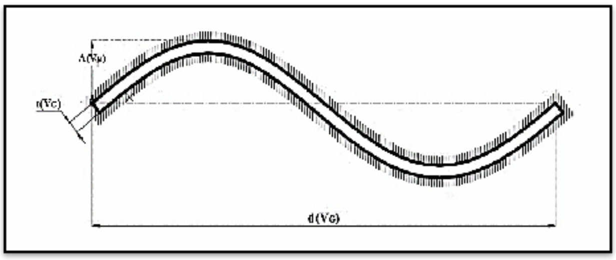


Figure 4. Dimensional characteristics of sinusoidal wavy VG.

Table 1. Characteristics of the sine function related to VGs.

θ (degree)			30°		
A	1-	5/0-	0	5/0	1
d (mm)	6		8	10	

2.4 Output parameter in post-processing

After calculations and obtaining the main output parameters, including the pressure field, velocity field, and convection heat transfer coefficient in the ANSYS, the parameters required for analysis of results are calculated. These parameters are the Nusselt number (local and average for tubes) that is calculated according to governing equations in Section 2.2 (Equation (6)) by considering the tube diameter as the hydraulic diameter; pressure drop in the length of heat exchanger that is determined using the pressure field obtained from Equation (7) in Section 2.2; finally, London factor that is obtained from Equation (9) in Section 2.2. These parameters are determined to examine the overall performance of the heat exchanger in this study.

3. Solution validation through comparing Nusselt numbers in the simple model (without VG)

The solution validation is an important part of numerical simulations in each project. This section compares the initial model (without considering VGs) with two other studies [2,4] to verify the numerical solution. It should be noted that the experimental results are adopted from Žkauskas [2], and the numerical results are extracted from Gholami et al. for comparison with obtained results. In the paper of Žkauskas [2], Equation (4) has been proposed for calculating the Nusselt number in the shell-and-tube heat exchanger composed of several tubes.

$$Nu_D = 0.92(0.52 \times {Re_D}^{0.5} \times Pr^{0.36} \times (\frac{Pr}{Pr_w})^{0.25}) \qquad (14)$$

According to **Table 2** and **Figure 5**, which compare the obtained results, in each paper, it is observed that increasing the Reynolds number increases Nusselt number. Since Žkauskas [2] has conducted experimental work, the results of Žkauskas [2] are promising. Based on numerical computations, Golami et al. [4] have also presented acceptable results. **Figure 5** illustrates that the final results of the present study are close to the results of the experimental work, which is superior to other studies in the literature. Now, Nusselt numbers at Reynolds numbers 400, 600, 800, and 1000 are compared between the present study and Žkauskas [2] and Gholami et al. [4].

Table 2. Comparison of Nusselt number at different Reynolds numbers in papers.

Reynolds number	**Nusselt number**				
	Žkauskas [2]	Gholami et al. [4]	Present study	Difference with Žkauskas [2]	Difference with Gholami et al. [4]
400	8.258	6.701	6.855	20.4%	2.24%
600	10.176	8.666	8.836	15.16%	1.92%
800	11.764	10.16	10.633	10.63%	4.44%
1000	13.116	11.513	12.281	6.79%	6.8%

Figure 5 presents the difference between the numerical solutions of the present study and those of Žkauskas [2] and Gholami et al. [4]. All of these results are obtained from Equation (14). After comparison, the validity of the numerical solution can be ensured regarding the agreement between the results.

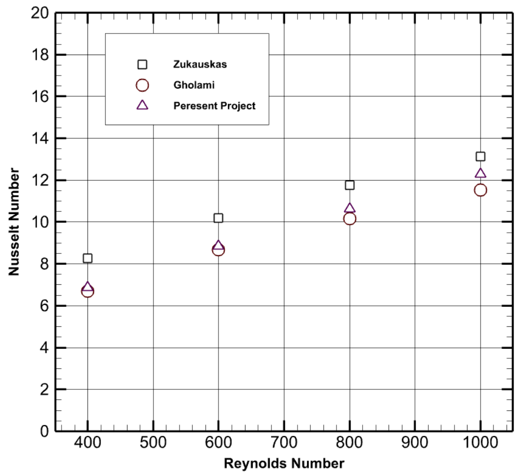


Figure 5. Nusselt number versus Reynolds number.

4. Grid introduction

The grid is generated in ANSYS software, and an unstructured grid with tetrahedral elements is used as the simulation model has a relatively simple geometry. As presented in **Figure 6**, one variable is examined for different numbers of elements (Nusselt number in this study due to the importance of heat transfer), and the number of computational nodes is increased until no tangible change occurs in the Nusselt number (when the selected variable has no difference between two consecutive grids). For example, the variation of the curve slope in **Figure 6** is lower for the number of elements more than 18916, compared with that of lower numbers of elements. So, a grid with 18916 elements can be a good selection for the grid independence of results. The general schematic of this grid can be seen in **Figure 7**.

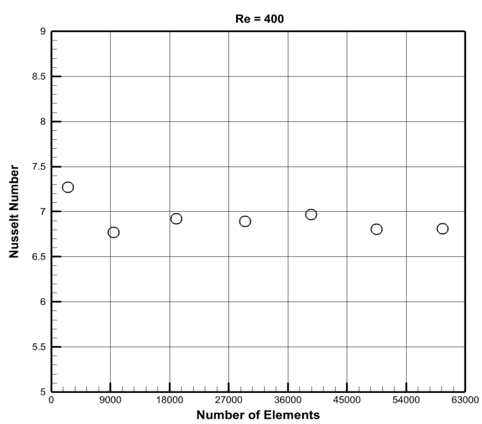


Figure 6. Nusselt number variation for different numbers of elements at Re = 400.

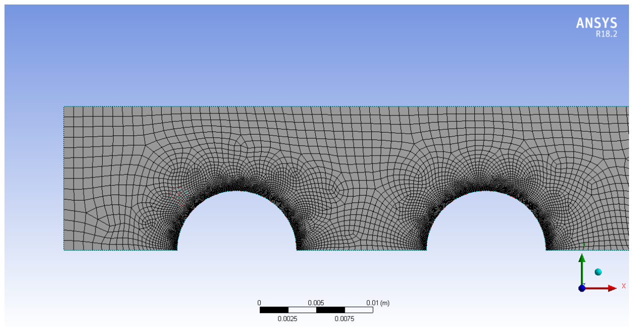


Figure 7. Selected grid with 18916 Quad/Tri elements.

5. Results and discussions

Before interpreting the results, it should be noted that there is a recirculation zone in front of each tube in the inline arrangement without VGs. The reason is that the flow is separated in front of the tube, and separated flows reach each other again behind the adjacent tube to create a large recirculation zone between two adjacent tubes. This phenomenon is demonstrated in **Figure 8**. Because of this phenomenon, it is of great importance to use VGs to direct the fluid over tubes.

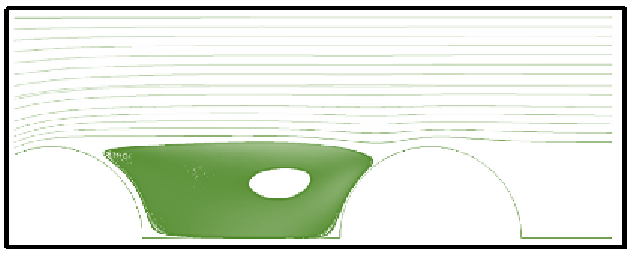


Figure 8. Recirculation zone between two adjacent tubes.

In comparison with the flat VGs in the same range of Re, using sinusoidal wavy VGs leads to a relatively constant reduction of the wake area behind tubes. This phenomenon controls and reduces drag force by directing the high-momentum fluid and postponing the flow separation behind the tube. Hence, using VGs improves the heat transfer rate over tubes.

5.1 Effect of VG's geometric parameters on Nusselt number over tubes

As the fluid flow approaches VGs, longitudinal vortices are created, and the heat transfer rate is improved significantly. **Figure 9** shows that increasing the VG length increases Nu. Therefore, VGs with a length of 10 mm and VGs with a length of 6 mm exhibit the best and worst performances in terms of heat transfer, respectively. Also, increasing Re magnifies the effects of length change on Nu. From different amplitudes, $A_3 = 1 > A_2 = 0 > A_1 =- 1$ deliver the greatest efficiency respectively. Quantitatively, at a constant length, Nu increases by 47-88% by increasing the Reynolds number. Generally, it can be said that using VGs causes the fluid flow to be disturbed and the heat transfer rate to increase by decreasing the wake area that is formed after each tube.

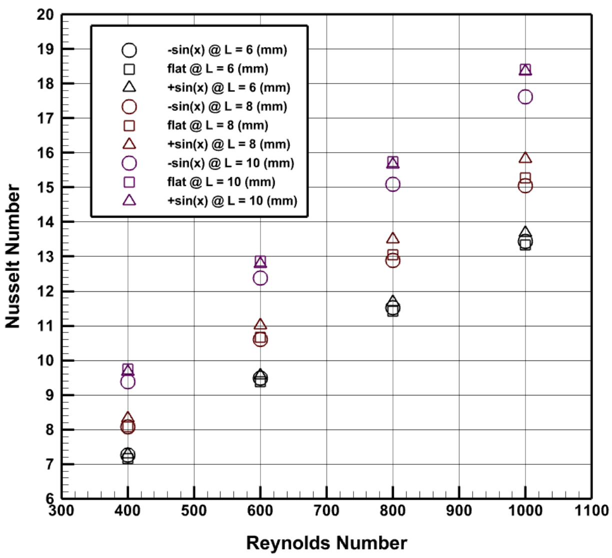


Figure 9. Average Nu variation versus Re for different VG lengths.

5.2 Effect of VG's geometric parameters on the pressure drop along the heat exchanger length

After examining the Nu variation, another important parameter, namely the pressure drop, is evaluated to select the best VG model.

According to results obtained from obtained diagrams and contours, it can be said that all proposed VG models create pressure drop along the heat exchanger length.

As illustrated in **Figure 10**, pressure drop has a direct relationship with Re and increases by increasing Re. This variation is more significant at higher lengths of the VG.

Moreover, increasing the length increases the pressure drop, and it is evident that higher lengths are not suitable for VG design. The sine wave amplitudes, $A_2 = 0 > A_3 = 1 > A_1 =- 1$ result in the greatest pressure drop, respectively, and the first value of amplitude ($A_1 =- 1$) is the best as it has the lowest pressure drop. Another notable point that can be seen in **Figure 10** is the insignificant effect of amplitude on the pressure for a VG length of 8 mm. In other words, the effect of amplitude on the pressure drop is more considerable at low and high lengths (6 mm and 10 mm) clearly. Quantitatively, at a constant length, increasing Re increases the pressure drop by 82-457%.

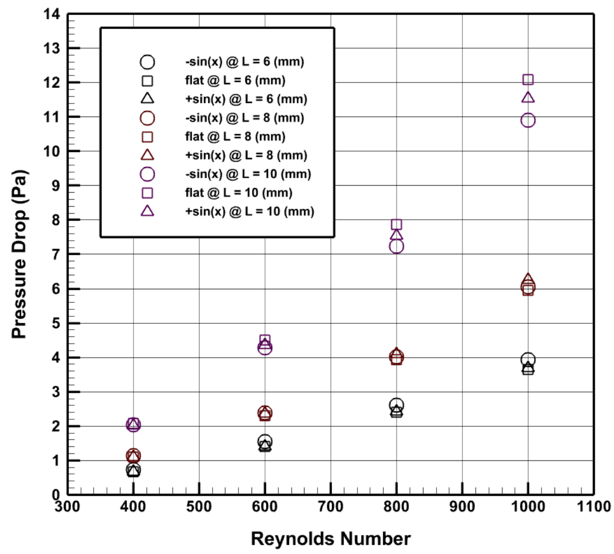


Figure 10. Pressure drop variation versus Re for different VG lengths.

5.3 Effect of VG's geometric parameters on London factor

As discussed in Section 2.2, London factor quantifies the overall performance of the heat exchanger in terms of heat transfer rate and pressure drop. According to Equation (9), a larger London factor means a higher heat transfer rate and lower pressure drop, a better and more suitable case relative to others.

As presented in **Figure 11**, increasing Re decreases London factor. By increasing the VG length, London factor is also reduced. Therefore, among three modeled VGs of different lengths, 6-mm VG outperforms the other ones. In addition, according to **Figure 11**, the effect of sine wave amplitude on London factor becomes more highlighted by decreasing VG length. So, at a VG length of 6 mm, amplitudes of $A_1 = 1 > A_2 = 0 > A_3 = -1$ result in the greatest London factor, respectively. Finally, at a constant length, an increase in Re can decrease London factor by 15-20%.

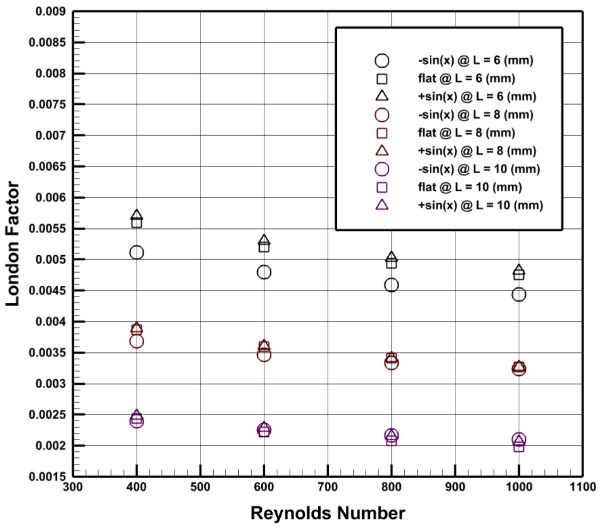


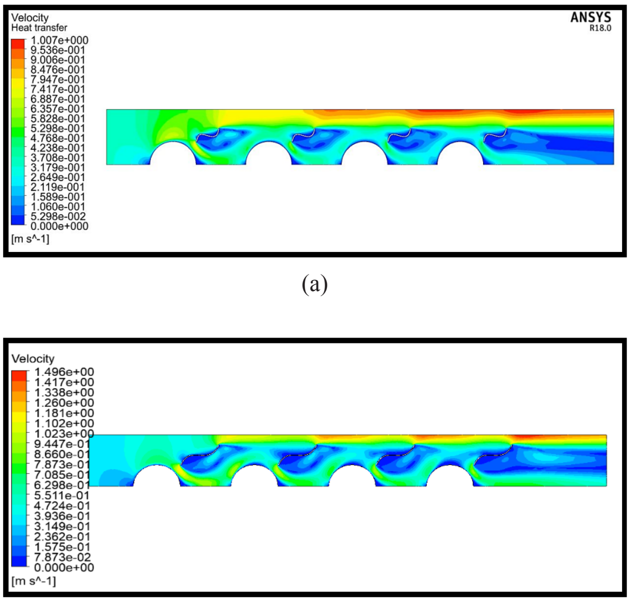
Figure 11. London factor versus Re for different VG lengths.

6. Comparison of pressure, velocity, and temperature contours for different VG lengths

This section compares the pressure, velocity, and temperature contours for different VG lengths. These contours are examined in another way due to the similar behavior of attack angle and length variations against inlet velocity. Two models of designed VGs are examined. One of these VGs is 6 mm in length with an amplitude of 1 and another one is 10 mm in length and flat ($A_2 = 0$), which deliver the highest and lowest efficiencies, respectively.

Based on **Figure 12**, it is observed that after the fluid flow impact on 10-mm VG, a vortex is created inside the heat exchanger. This vortex is weaker in **Figure 12(a)** (6-mm VG) than that of 10-mm VG.

By comparing the pressure contours in **Figure 13**, it is concluded that those VGs with a length of 10 mm have no significant effect on the pressure drop, and the inlet flow pressure is rapidly reduced to reach almost zero at the end of a heat exchanger. However, as shown in **Figure 13(a)**, the performance of 6-mm VG shows that this VG has a key role in pressure drop and decreases the flow pressure in the length of a heat exchanger. The pressure in **Figure 13(a)** is not reduced abruptly in contrast to **Figure 13(b)**, and the pressure drop variation rate in **Figure 13(a)** is low, indicating the superiority of 6-mm VG over other ones.

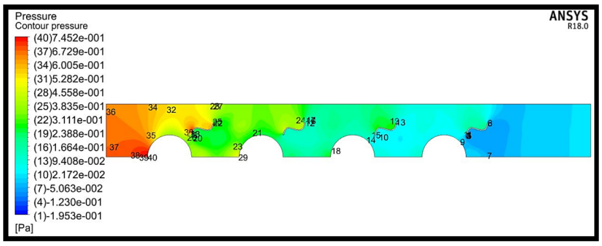


(a)

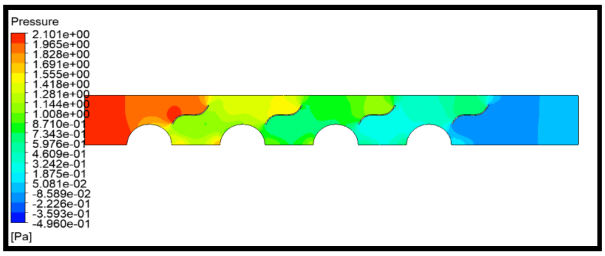
(b)

Figure 12. Velocity contours for different shapes of VG: (a) 6-mm length, (b) 10-mm length.

Another important parameter affecting the overall performance of heat exchangers is the temperature variation inside the heat exchanger. By comparing the contour in **Figure 14**, the performance of two VGs is similar in the middle region of the heat exchanger. However, in the initial and end parts of the heat exchanger, VG of length 6 mm shows better performance than VG of length 10 mm. It is worth mentioning that according to temperature contours, the formation of stagnant hot points is not considered because of VGs.

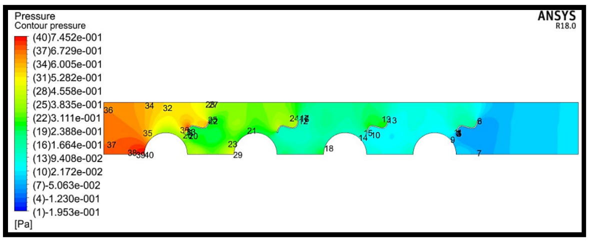


(a)

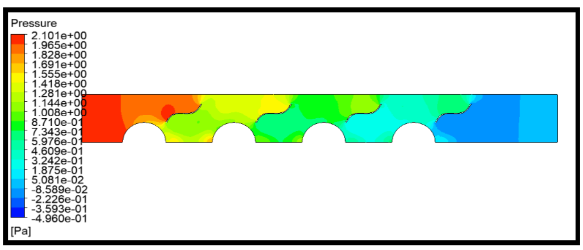


(b)

Figure 13. Pressure contours related to highest and lowest pressure drops for different lengths of VG: (a) 6 mm, (b) 10 mm.



(a)



(b)

Figure 14. Temperature contours for different VG lengths: (a) 6 mm, (b) 10 mm.

7. Conclusions

The present study investigated the heat transfer rate and pressure drop in shell-and-tube heat exchangers in which VGs were mounted. To this end, VGs were designed using sine wave functions and examined under different amplitudes and lengths to specify the effects of length and amplitude on the heat transfer rate from tubes and pressure drop along the heat exchanger length. After evaluating diagrams, contours, and results obtained in this study, it was found that accurately using VGs can impressively help heat transfer augmentation and pressure drop reduction. In the end, the main concluded remarks are as follows:

- In the heat transfer evaluation of tubes, VGs are preferred to be used near the last column tubes since the fluid flow is diverted after impacting on tubes of the first column and the heat transfer rate gradually decreases in the next tubes.

- Heat exchangers of larger lengths have lower London factor and are not suitable options due to higher pressure drop.

- Among different VGs, the sinusoidal wavy shape with an amplitude of 1 provides the greatest heat transfer rate relative to those of amplitudes 0 and –1.

- Considering all diagrams and results, it is found that from the designed VGs, VG with a length of 6 mm and sine wave amplitude of 1 presents the best performance and efficiency relative to others with a 5.06% increase in Nusselt number.

Nomenclature

English symbols	
A	Area (m^2)
C_v	Specific heat at constant volume ($\frac{J}{kg.K}$)
C_p	Specific heat at constant pressure ($\frac{J}{kg.K}$)
D_H	Hydraulic diameter (m)
f	Friction factor for pressure drop calculation
g	Gravitational acceleration ($\frac{m}{s^2}$)
h	Convective heat transfer coefficient ($\frac{W}{m^2.K}$)
J	Colburn number
J/f	London factor
k	Thermal conductivity ($\frac{W}{m.k}$)
Nu	Nusselt number
P	Pressure (Pa)
pr	Prandtl number
$\vec{q}$	Heat transfer rate (W/m^2)
Re	Reynolds number
St	Stanton number
T	Temperature (K)
t	Time (s)
U_m	Velocity at minimum area ($\frac{m}{s}$)
u, v, w	Velocity field components ($\frac{m}{s}$)
Greek symbols	
ρ	Density (kg/m^3)
μ	Dynamic viscosity ($\frac{kg}{m.s}$)
Subscripts	
m	Minimum surface area
v	Volume
p	Pressure
H	Hydraulic

Conflict of Interest

There is no conflict of interest.

References

[1] Gao, S.D., Wang, L.B., Zhang, Y.H., et al., 2003. The optimum height of winglet vortex generators mounted on three-row flat tube bank fin. Journal of Heat Transfer. 125(6), 1007-1016.

[2] Žkauskas, A., 1972. Heat transfer from tubes in crossflow. Advances in Heat Transfer. 8, 93-160.

[3] Leu, J.S., Wu, Y.H., Jang, J.Y., 2004. Heat transfer and fluid flow analysis in plate-fin and tube heat exchangers with a pair of block shape vortex generators. International Journal of Heat and Mass Transfer. 47(19), 4327-4338.

[4] Gholami, A.A., Wahid, M.A., Mohammed, H.A., 2014. Heat transfer enhancement and pressure drop for fin-and-tube compact heat exchangers with wavy rectangular winglet-type vortex generators. International Communications in Heat and Mass Transfer. 54, 132-140.

[5] He, Y.L., Chu, P., Tao, W.Q., et al., 2013. Analysis of heat transfer and pressure drop for fin-and-tube heat exchangers with rectangular winglet-type vortex generators. Applied Thermal Engineering. 61(2), 770-783.

[6] Modi, A.J., Rathod, M.K., 2019. Comparative study of heat transfer enhancement and pressure drop for fin-and-circular tube compact heat exchangers with sinusoidal wavy and elliptical curved rectangular winglet vortex generator. International Journal of Heat and Mass Transfer. 141, 310-326.

[7] Awais, M., Bhuiyan, A.A., 2019. Enhancement of thermal and hydraulic performance of compact finned-tube heat exchanger using vortex generators (VGs): A parametric study. International Journal of Thermal Sciences. 140, 154-166.

[8] Lu, G., Zhai, X., 2019. Effects of curved vortex generators on the air-side performance of fin-and-tube heat exchangers. International Journal of Thermal Sciences. 136, 509-518.

[9] Wang, Y., Zhao, W., Wang, P., et al., 2021. Thermal performance of elliptical fin-and-tube heat exchangers with vortex generator under various inclination angles. Journal of Thermal Science. 30, 257-270.

[10] Sahel, D., Ameur, H., Alem, K., 2021. Enhancement of the hydrothermal characteristics of fin-and-tube heat exchangers by vortex generators. Journal of Thermophysics and Heat Transfer. 35(1), 152-163.

[11] Caliskan, S., Şevik, S., Özdilli, Ö., 2022. Heat transfer enhancement by a sinusoidal wavy plate having punched triangular vortex generators. International Journal of Thermal Sciences. 181, 107769.

[12] Naik, H., Tiwari, S., Kim, H.D., 2022. Flow and thermal characteristics produced by a curved rectangular winglet vortex generator in a channel. International Communications in Heat and Mass Transfer. 135, 106103.

[13] Hu, D., Zhang, Q., Song, K., et al., 2023. Performance optimization of a wavy finned-tube heat exchanger with staggered curved vortex generators. International Journal of Thermal Sciences. 183, 107830.

[14] Song, K., Hu, D., Zhang, Q., et al., 2022. Thermal-hydraulic characteristic of a novel wavy fin-and-circle tube heat exchanger with concave curved vortex generators. International Journal of Heat and Mass Transfer. 194, 123023.

[15] Luo, W., Yang, Z., Jiao, K., et al., 2022. Novel structural designs of fin-tube heat exchanger for PEMFC systems based on wavy-louvered fin and vortex generator by a 3D model in OpenFOAM. International Journal of Hydrogen Energy. 47(3), 1820-1832.

[16] Shlash, B.A.A., Koç, I., 2022. Turbulent fluid flow and heat transfer enhancement using novel Vortex Generator. Journal of Advanced Research in Fluid Mechanics and Thermal Sciences. 96(1), 36-52.

[17] Tepe, A.Ü., Yilmaz, H., 2022. Thermal-hydraulic performance of the circular-slice-shaped-winglet for tube bank heat exchanger. International Jour-

nal of Thermal Sciences. 179, 107711.

[18] Göksu, T.T., Behçet, R., 2022. Experimental investigation of the effect of a novel curved winglet vortex generator on heat transfer with a designed controller circuit. International Journal of Thermal Sciences. 180, 107724.

[19] Zhong, X., Fu, S., Chan, K., et al., 2022. Experimental and numerical study of heat transfer performance of a channel flow with an inverted flag. International Journal of Heat and Mass Transfer. 193, 122969.

[20] Akcay, S., 2022. Numerical analysis of heat transfer improvement for pulsating flow in a periodic corrugated channel with discrete V-type winglets. International Communications in Heat and Mass Transfer. 134, 105991.

[21] Wang, Y., Liu, P., Xiao, H., et al., 2022. Design and optimization on symmetrical wing longitudinal swirl generators in circular tube for laminar flow. International Journal of Heat and Mass Transfer. 193, 122961.

[22] Berber, A., Gürdal, M., Yetimoğlu, M., 2022. Experimental study on the heat transfer enhancement in a rectangular channel with curved winglets. Experimental Heat Transfer. 35(6), 797-817.

[23] Suman, S., Uppada, V., Singh, S., et al., 2022. Numerical investigation and experimental validation of shape and position optimisation of a static wavy flag for heat transfer enhancement. International Journal of Ambient Energy. 43(1), 3237-3245.

[24] He, J., Deng, Q., Xiao, K., et al., 2022. Impingement heat transfer enhancement in crossflow by v-shaped protrusion vortex generator. Heat Transfer Engineering. (IF 2.431), 1-26.
DOI: https://doi.org/10.1080/01457632.2022.2134080

[25] Sheikholeslami, M., Nimafar, M., Ganji, D.D., 2017. Analytical approach for the effect of melting heat transfer on nanofluid heat transfer. The European Physical Journal Plus. 132, 385.