

Heat Transfer and Pressure Drop Assessment of a Vortex Generator Supported Fin-And-Tube Heat Exchanger



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Abstract This paper aims at enhancing the thermo-fluid performance of a plain fin-and-tube heat exchanger. Influence of the height of rectangular type vortex generator on the heat exchanger performance has been analysed. In dimensionless form, the heat transfer is expressed as Colburn factor (j) and pressure drop in terms of the friction factor (f). The arrangement of all the vortex generators follows a “common-flow-up” configuration which tends to counter-rotate the vortices in the wake region of the tubes, thereby, enhancing the thermal mixing to improve the heat transfer. Another parameter, called quality factor (a ratio of j and f), is also evaluated to compare the results between different heights of the winglets. It has been observed that the j factor is the highest for the height 2.8 mm and f factor is the lowest for the height 0.7 mm, whereas the quality factor suggests the height of 0.7 and 1.4 mm to be the appropriate ones.

Keywords Vortex generators · Fin-and-tube heat exchanger · Heat transfer enhancement · Rectangular winglet

1 Introduction

Compact plain fin-and-tube heat exchangers find numerous applications such as in domestic and commercial cooling systems, petrochemical industries and for cooling of several electronic equipment. However, the heat transfer surface area is reduced in the compact heat exchangers compared to the conventional heat exchangers. Also, the low heat transfer coefficient (h) of air makes the air side thermal resistance substantially high. Therefore, to make these devices more efficient, the air side thermal resistance needs to be lowered. An effective way of increasing heat transfer coefficient is to place the winglets in the channel (guiding the flow)—this diverts the flow towards the wake region behind the tubes, thereby minimizing the tube wake region and increases the effective surface contact of the fluid. Several orientations

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and types of winglets have been studied to improve the performance parameters of heat exchangers, viz. the pressure drop and heat transfer coefficient.

The concept of increasing heat transfer performance using VGs is not new, and many researchers have investigated the potential of different VG shape and location. Edward and Alker [1], after investigating the effect of steam wise vortices on heat exchanger performance, reported that delta winglets performed superior to cubes placed on a flat plate. Generally, two types of VG configurations are used, i.e. “common-flow-up” and “common-flow-down”. Earlier literature [2–4] reveals that in “common-flow-down” configuration, pair of VGs is placed downstream to the tube to introduce turbulence in the flow for greater thermal mixing. For “common-flow-up” configuration [5–7], the placements of VGs are slightly upstream of the tube forming a narrow passage and accelerating the fluid near the tube. This leads to delay in separation and minimization of the wake. Previous literature reports that common-flow-down configuration is more efficient with in-line tube geometry than with staggered-tube geometry under identical conditions [8]. VGs enhance the heat transfer, but on the other hand, it also increases pressure drop—this means that more pumping power is required with VGs. Russell et al. [9] compared the in-line rectangular and delta winglets with staggered arrangements and concluded that the effectiveness of rectangular winglets improves in staggered arrangement with two rows. Delta and rectangular wing and winglets were examined experimentally by Fiebig et al. [10] from heat transfer viewpoint and they found the delta winglet to be more effective. Torii et al. [11] examined the in-line and staggered arrangement of tubes at $Re = 350\text{--}2200$ and reported the maximum enhancement of 40% in heat transfer, whereas the associated penalty of pressure drop was about 50% in case of staggered arrangement. Joardar and Jacobi [12] employed delta winglets in “common-flow-up” configuration with Re ranging from 220–960 and reported that the air side performance of heat transfer improved by 44% and 69%, respectively, for single and three-row VG pairs. Few researchers [13, 14] also used the concept of punched triangular and rectangular vortex generators to enhance the heat transfer performance. Agarwal and Sharma [15] used rectangular winglets at different roll angle and reported the optimized roll angle of 15° that increased the heat transfer by 17.5%. Sarangi and Mishra [16] worked on the location of rectangular winglet pairs and presented optimum stream and span wise locations. On increasing the winglet number and attack angle, they reported an increase in the heat transfer as well as pressure drop. In their recent work [17], they also investigated the effect of locations of plain and wavy rectangular winglets for improved thermo-fluid performance.

In view of the above, it becomes apparent that no study, based on the effect of height of winglets (corresponding to the channel height), on heat transfer and pressure drop has been performed. Thus, the objective of the present work is to investigate the impact of winglet height on the thermo-fluid performance of the heat exchanger.

2 Mathematical Formulation

A model of plain fin-and-tube type heat exchanger is numerically investigated in this paper. Figure 1 shows the schematic diagram of the flow domain. The assembly of heat exchanger is such that the tubes are expanded within the collars. Since the tubes and collars have very high thermal conductivity, the temperature is maintained constant there.

2.1 Governing Equations

The fluid is Newtonian with constant properties, and the flow is in laminar regime where Re ranges from 200–1000. The continuity, momentum, energy and conduction equations according to flow conditions are shown below.

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum equation:

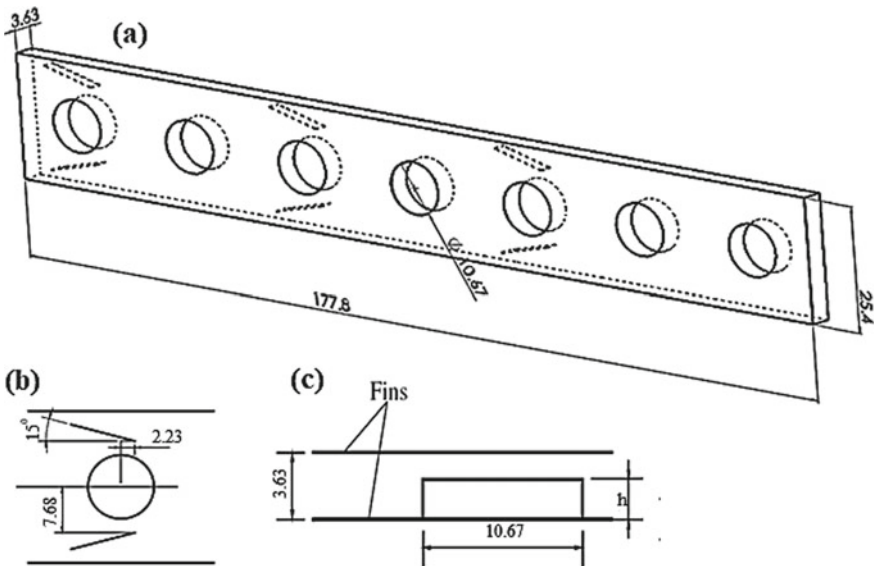


Fig. 1 Winglet orientation (a) computational domain, (b) top view, and (c) side view (dimensions shown are in mm)

$$\frac{D}{Dt}(\rho u_i) = \frac{\partial}{\partial x_i} \left(\mu \frac{\partial u_k}{\partial x_i} \right) - \frac{\partial p}{\partial x_i} \quad (2)$$

Energy equation:

$$\frac{D}{Dt}(\rho T) = \frac{\partial}{\partial x_i} \left(\frac{k_a}{C_p} \frac{\partial T}{\partial x_i} \right) \quad (3)$$

Conduction equation:

$$\frac{\partial^2 T}{\partial x_i^2} = \frac{1}{\alpha} \frac{\partial T}{\partial t} \quad (4)$$

where

$$\alpha = \frac{k_{al}}{(\rho_{al} C)} \quad (5)$$

2.2 Boundary Conditions

The boundary conditions are described as follows and also represented in Fig. 2:

- At the inlet surface of the flow domain, Dirichlet boundary condition is applied for both velocity as well as temperature:
- $U = U_{in} = \text{constant}$, $v = 0$, $w = 0$; $T_{in} = 310.6 \text{ K}$.
- On top and bottom surfaces, periodic boundary condition is applied:
- $u_{up} = u_{down}$ and $T_{up} = T_{down}$.
- On the side surfaces, we use the symmetry boundary condition:

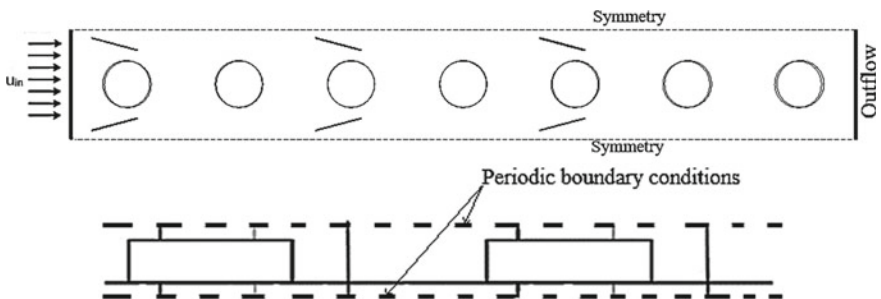


Fig. 2 Boundary conditions applied on the computational domain

$$\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, v = 0, \frac{\partial T}{\partial y} = 0$$

- At the outlet boundary, Neumann boundary condition is applied:

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0$$

- At the fin, tube and winglet surfaces, we use

$$T_{\text{tube}} = 291.77 \text{ K}, u = v = w = 0, \text{ (no slip) and coupled boundary condition.}$$

2.3 Numerical Method

The fluid domain was discretized using hexahedral cells (Cartesian mesh, in particular) using ANSYS workbench 16.2. Three levels of mesh were evaluated; Fig. 3 presents the (level 1) mesh over a small section of the heat exchanger consisting of 385,542 hexahedra in total. Level 2 and level 3 mesh consisted of 548,664 and 872,382 hexahedra.

CFD code Fluent is utilized to compute the governing equations with the prescribed boundary conditions. The second-order upwind scheme is applied to discretize the convective terms appearing in the governing equations. The SIMPLE algorithm is used for the pressure–velocity coupling. The convergence is attained when residuals fall below 10^{-6} for energy and 10^{-3} for momentum and continuity. For all the cases, the pseudo-transient approach was adopted for the simulations. These settings are consistent with most of the studies reported in the literature [16, 17].

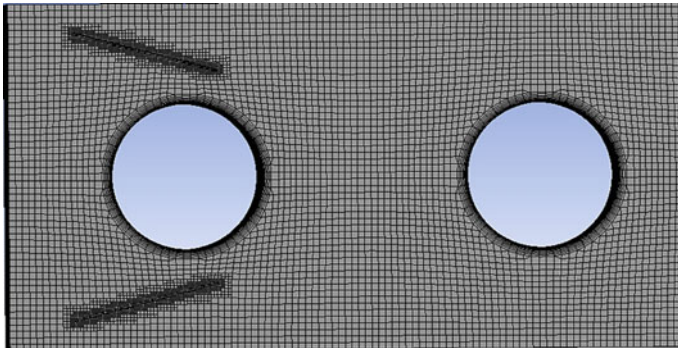


Fig. 3 Top view of the generated mesh

2.4 Parameter Definitions

The following parameters are evaluated to assess the heat exchanger's performance [14].

Total heat transfer

$$Q = \dot{m}C_p(\dot{T}_o - \dot{T}_{in}) \quad (6)$$

Here, $Q \equiv$ heat transfer, $\dot{m} \equiv$ inlet mass flow rate, $\dot{T}_o \equiv$ outlet temperature, $C_p \equiv$ specific heat capacity, and $\dot{T}_{in} \equiv$ inlet temperature.

Hydraulic diameter,

$$D_h = 4(A_{min}L)/A_T \quad (7)$$

A_T and A_{min} are the total heat transfer surface area and minimum free flow area, and L is the flow length.

$$Re = \rho U_{in} D_h / \mu \quad (8)$$

where ρ and μ are the fluid density and dynamic viscosity, respectively. U_{in} is the frontal velocity.

Colburn factor,

$$j = St \cdot Pr^{2/3} \quad (9)$$

Stanton number,

$$St = \frac{h}{\rho C_p V_m} \quad (10)$$

h is the heat transfer coefficient, and Pr is the Prandtl number.

Friction factor,

$$f = \frac{\Delta p}{\frac{\rho V_m^2}{2} \frac{A_T}{A_{min}}} \quad (11)$$

Δp is the pressure drop.

3 Validation of Model and Numerical Method

Validation is an important part in simulations as one justifies the numerical settings used and also evaluates the effect of the grid density on the results. Simulations are

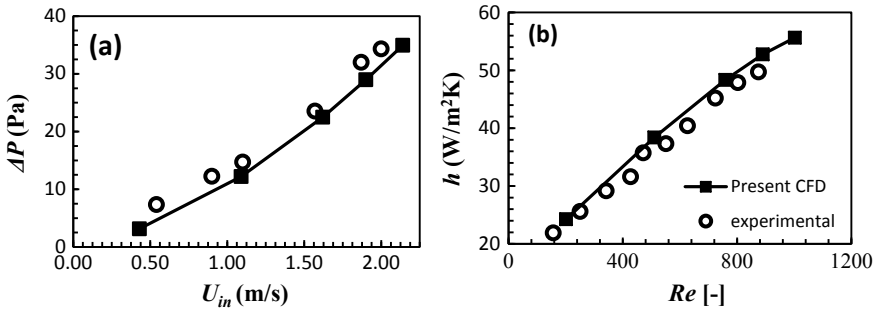


Fig. 4 Validation of (a) pressure drop and (b) heat transfer coefficient with experimental data [12] for fin-and-tube heat exchanger without winglets

performed on a model of a fin-and-tube heat exchanger having the same dimension as described in [12]. The velocity at the inlet ranges from 0.54–2.14 m/s, corresponding to Re ranging from 200–1000. The validation results are compared for the pressure drop (ΔP) and the heat transfer coefficient (h) with the experimental results [12], as shown in Fig. 4a, b, respectively (with level 1 mesh). The CFD predictions are in good agreement with the experimental results. Furthermore, level 2 and level 3 mesh were also evaluated, and the maximum deviations observed were 3.73% in ΔP and 4.13% in h values. Hence, the solution can be considered to be independent of the grid size. For all the cyclone variants, we opt for level 1 mesh to reduce computational overheads.

4 Results and Discussion

This paper attempts to study the influence of winglet height on thermo-fluid characteristics. The winglet height is varied with a step size of 0.7 mm ranging from 0.7–2.8 mm for a fixed winglet length of 10.67 mm. The results are plotted for pressure drop and heat transfer coefficient against Re . Also, an optimizing factor, called quality factor (E), is plotted against Re to summarize the overall performance.

4.1 Effect on Heat Transfer Coefficient

Figure 5 shows the variations in h values against Re for various winglet heights. The height for delta winglets without any reduction is 3.5 mm, and the height is reduced with a step size of 0.7 mm. The heat transfer coefficient is maximum for a height of 2.8 mm (i.e. 20% reduction), and it decreases by further reducing the height of

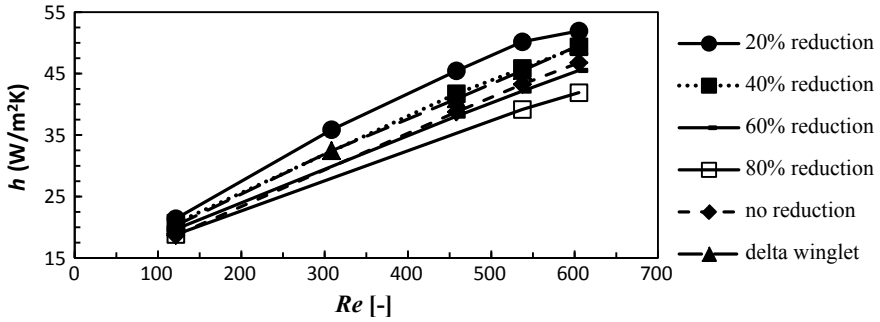


Fig. 5 Influence of Re on h for different height of winglets and comparison with baseline case

winglets. The results are also compared with the experimental results of the three-delta-winglets model, and they were very closely associated at 2.1 mm height of winglets.

Figure 6 presents the temperature distribution over the cut section, which indicates that the greater winglet height helps in providing enriched thermal mixing in the regions of poor heat transfer and thereby enhances local heat transfer coefficient downstream of tubes. With a reduction in the winglet height, a part of the incoming flow does not get diverted towards the thermally isolated region, and relatively lower heat transfer effect is obtained.

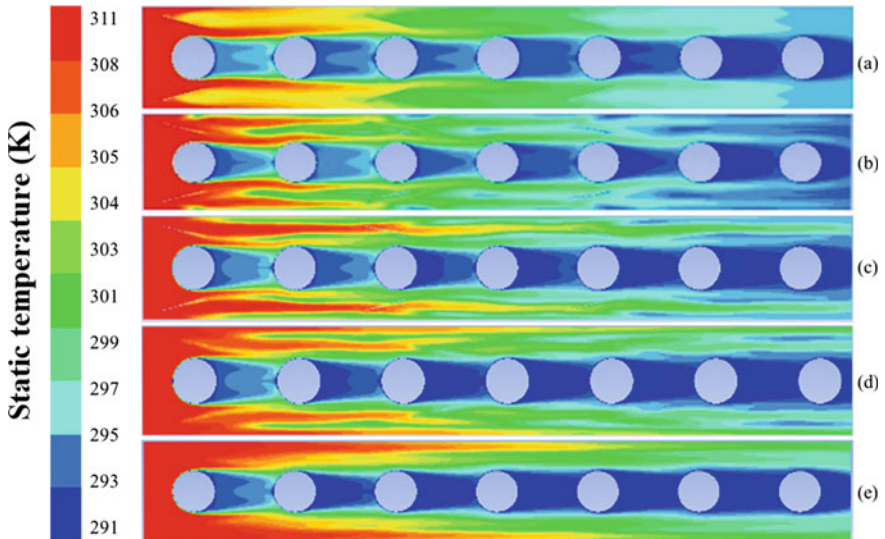


Fig. 6 Temperature contours (unit: K) for different winglet heights at $Re = 1002$. (a) 3.5 mm (No reduction), (b) 2.8 mm (20% reduction), (c) 2.1 mm (40% reduction), (d) 1.4 mm (60% reduction), (e) 0.7 mm (80% reduction)

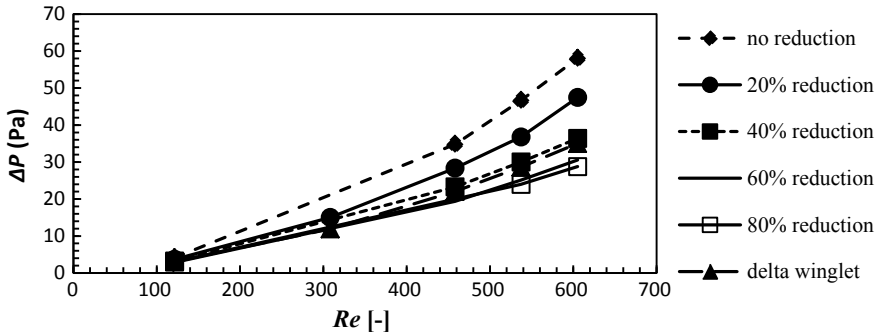


Fig. 7 Effect of Re on pressure drop for different height of winglets and comparison with baseline case

4.2 Pressure Drop

Figure 7 presents the pressure drop values as a function of Re . It represents the power required to pump and maintain the flow along the channel. Two types of drags are associated with pressure drop, i.e. pressure drag and friction drag. Pressure drag dominates when the fluid is restricted to flow due to additional oblique surfaces. Friction drag dominates when surface over which the fluid flows is rough, where friction is the main cause of restriction. Pressure drop for 0.7 mm (80% reduction) and 1.4 mm (60% reduction) height of winglets is very close and minimum of all other reductions. Further changes in height of the winglets lead to an increase in pressure drop. This increase in pressure drop is due to enhanced flow restriction with increasing height of the winglets over a certain channel height.

4.3 Quality Factor

The ratio of Colburn factor to the friction factor is termed as the quality factor. Higher the quality factor, the more will be the effectiveness of the heat exchanger. It becomes apparent from Fig. 8 that the quality factor came out to be maximum for 1.4 mm height of winglets (60% reduction). Therefore, corresponding to this height of winglets, an optimum balance between heat transfer and pressure drop will be achieved.

5 Conclusions

The objective of this paper was to assess the influence of winglet height on thermo-fluid characteristics of a fin-and-tube heat exchanger. An in-line configuration, with

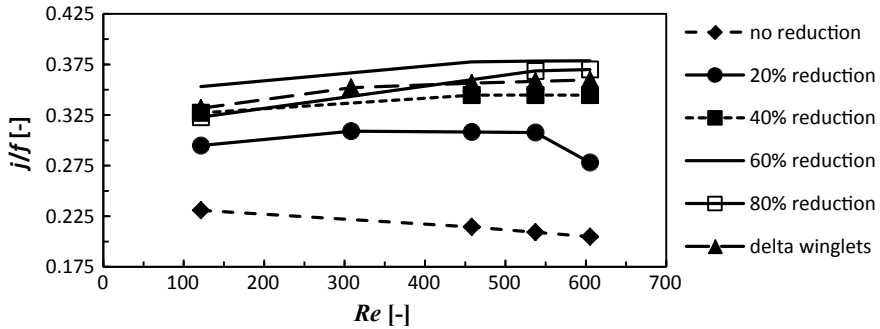


Fig. 8 Quality factor as a function of Re for different height of winglets and comparison with baseline case

seven tubes supported with winglets of different heights, was numerically investigated at different Re . The overall performance of the heat transfer surface has been presented in terms of quality factor. Major findings, in reference to the experimental results of delta winglets, have been listed below:

1. The pressure drop increased with increase in height of winglets. For the winglet height of 1.4 mm, the pressure drop decreased by 12%. For a winglet height of 0.7 mm, the reduction in the pressure drop by 20% was observed.
2. The increased winglet height enhanced the heat transfer performance, and it came out to be the maximum for a height of 2.8 mm. For this height, the heat transfer was raised up to 39% at high Re .
3. Quality factor came out to be maximum for the winglet height of 1.4 mm which was enhanced by 6%, and this height was taken as the optimum winglet height. Height of 0.7 mm also showed an increase in quality factor by 2.8% at higher Re .

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