Performance Improvement of a Fin-Tube Heat Exchanger Using Rectangular Winglet as Vortex Generator

Sachin Gupta, Arvind Gupta, Munish Gupta

Abstract: Heat transfer performance of fin-tube heat exchanger can be augmented by using longitudinal vortex generators. Numerical simulations have been performed in the present work for investigating the effect of punching a rectangular winglet having hole from fin surface, on the heat transfer and flow resistance characteristics in a fin-tube heat exchanger. The concept of punching out a rectangular winglet having hole from the fin-plate surface is being proposed here and studied in two configurations namely, common flow down and common flow up. Comparisons on the basis of heat transfer and flow resistance characteristics have been drawn for all the configurations under consideration using Colburn’s factor (j), friction factor (f) and performance evaluation criterion (PEC) also known as area goodness factor (jf). Investigations have been performed considering Reynolds number in the range of 1500 to 9000 and angle of attack as 45°. The result clearly indicates that punched out rectangular winglet with hole having common flow down configuration at upstream location as exhibiting the best thermal performance, followed by common flow up at upstream location and, common flow down at downstream location.

Keywords: Rectangular Winglet, Vortex generator, Numerical Investigation, Fin-tube heat exchanger.

Nomenclature

| Symbol | Description |
|--------|-------------|
| $A_c$ | Cross Section Area (m$^2$) |
| $A_f$ | Surface Area (m$^2$) |
| $C_p$ | Specific heat at constant pressure (J Kg$^{-1}$K$^{-1}$) |
| $D_h$ | Hydraulic Diameter (m) |
| $k$ | Thermal conductivity (W m$^{-1}$K$^{-1}$) |
| $Q$ | Heat Transfer Capacity (W) |
| $m^*$ | Mass flow rate of air (kg s$^{-1}$) |
| $u$ | Component of velocity in x-direction (m s$^{-1}$) |
| $v$ | Component of velocity in y-direction (m s$^{-1}$) |
| $w$ | Velocity component in z-direction (m s$^{-1}$) |
| $Re$ | Reynolds number |
| $Nu$ | Nusselt number |
| $Pr$ | Prandtl number |
| $f$ | Friction factor |
| $j$ | Colburn’s factor |
| $P_m$ | Mean Press. for a cross section (Pa) |
| $\Delta P_m$ | Mean pressure drop (Pa) |
| $T_m$ | Mean temp. for a cross section (K) |
| $T_{m,in}$ | Mean temperature at inlet cross section (K) |

Greek Symbols

| Symbol | Description |
|--------|-------------|
| $\mu$ | Dynamic Viscosity (Pa.s) |
| $\nu$ | Kinematic Viscosity (m$^2$ s$^{-1}$) |
| $\rho$ | Density (kg m$^{-3}$) |

Abbreviations

- CFD$_D$: Common Flow Down at downstream location
- CFD$_U$: Common Flow Down at upstream location
- CFU$_U$: Common Flow Up at upstream location
- RWPH: Rectangular Winglet Pair with hole

I. INTRODUCTION

Heat exchangers are indispensable for achieving heat dissipation from the generation source point in applications such as heating, ventilation and air conditioning (HVAC), refrigeration devices and automobile radiators. Efforts for improving transfer of heat between two fluids flowing over and into the heat exchanger has been a topic of research since many years. Fin-tube type of heat exchanger is one of the most commonly used types of heat exchangers and there are numerous studies which have been carried out to increase the air side heat transfer, all of which involve either the use of active methods or passive methods for heat transfer augmentation. Since making use of passive methods does not require any extra power, they are commonly used and preferred. Among all such methods, vortex generators (VGs) are of prime interest. Various studies have been performed over the years which make use of various types of vortex generators for increasing air-side convective heat transfer coefficient in a heat exchanger. Numerous studies, both numerical and experimental have been performed in recent times considering winglet as the vortex generator for investigating the thermo-hydraulic behaviour of heat exchangers. [1] presented an exhaustive review of thermo-hydraulic characteristics for a fin-tube heat exchanger as well as for a channel flow by using delta and rectangular winglet as VGs. [2] performed numerical simulations considering delta winglet pair and concluded that thermal performance can be improved by as high as up to 240% behind the tube in wake region. [3] performed a number of experiments and concluded...
that by using vortex generators on fin surface, there was a considerable improvement in heat transfer. This was in agreement with the findings of [2], [4] performed numerical simulation and experimentation for a channel flow considering rectangular and delta wings as well as winglet both and concluded that winglets furnished superior heat transfer characteristics. In his study, he also observed that for identical dimensionless parameters, performance of rectangular and delta winglets was same. [5] considered channel flow to compare the performance of winglets and wings and concluded that winglets outperformed wings for heat transfer enhancement. Studies involving numerical investigations have been taken up over the years to examine the effect of different shapes of winglets and to find the best one for a particular application. Earlier studies were focused on rectangular and delta winglet but [6] simulated numerically and suggested three novel types of winglets as angle rectangular winglet (ARW), curved angle rectangular winglet (CARW) and rectangular trapezoidal winglet (RTW). They reported that for small attack angle, CARW as VG outperforms the other two types of winglet. [7] made experimental investigations and proposed two configurations for winglets known as common flow down (CFD) and common flow up (CFU). [8] performed numerical investigations for both the aforementioned configurations, that is CFU and CFD for rectangular as well as delta winglet and reported that better results are realized using rectangular winglet as compared to the latter in terms of superior heat transfer augmentation. The present work is the first to study the effects of punching out a rectangular winglet with hole, on the heat transfer and flow resistance characteristics in a fin-tube heat exchanger. Rectangular winglet pair having hole has been punched out in two configurations namely Common Flow Down and Common Flow Up. Numerical simulations have been performed using SST $k-\omega$ turbulence model which has a demonstrated capability of capturing flow separation in a superior way than $k-c$ turbulence model. Heat transfer and flow resistance characteristics have been compared for all the cases using Colburn’s factor ($j$), friction factor ($f$) and performance evaluation criterion (PEC) also known as area goodness factor ($j/f$). Investigations have been performed considering the Reynolds number in the range of 1500 to 9000 and angle of attack as 45°.

II. MODEL DESCRIPTION

A. Physical Model

In the present investigation, experiments were performed on a fin-and-tube heat exchanger installed in a wind tunnel. Fig. 1 shows experimental setup with test prototype. The test model has been made up of 25 parallel aluminum fin plates having dimensions as 300×200 mm$^2$ and fin pitch as 24 mm. A circular tube made up of copper with outer diameter as 52 mm and effective length as 300 mm has been fitted through the center of the series of parallel aluminium fin plates. In addition to circulating hot water inside the copper tube, an electric heater element has been used for heat generation and kept inside the copper tube. Fin plate number 13 with 3 mm thickness at middle has been chosen for the experimentation work among all the fin plates and fitted with winglets. Rectangular winglet with circular hole at center was the VG under consideration as represented in fig. 2(d). Diameter of tube ($D$), length of fin-plate ($L$), width of fin-plate ($W$), thickness of fin-plate ($t$) and fin pitch ($P$), respectively are 52 mm, 300 mm, 200 mm, 3 mm and 24 mm. Longitudinally and transversally, VG has been positioned at 0.5$D$ from the center of the plate. Angle of attack ($\alpha$), length ($L$), height ($H$) and thickness ($t$) of VG is equal to 45°, 18 mm, 12 mm and 3 mm respectively. Longitudinal ($x$) and transverse position ($y$) of hole have been taken as 9 mm and 6 mm, respectively. RWPH was attached in both the configurations viz., CFU and CFD at upstream as well as downstream location. Size of VGs, also represented by a measure of aspect ratio has been taken as 1.33 and positions of winglet have been taken as suggested by [9]. As reported by [10] the diameter of the hole has been considered as 4mm. Reynolds number was varied from 1500 to 9000 and angle of attack was taken as 45° for optimum thermal performance enhancement.

![Fig. 1 – Experimental Setup](image)

![Fig. 2(a) – Common Flow Down (CFD) Downstream](image)

![Fig. 2(b) – Common Flow Down (CFD) Upstream](image)
B. Governing Differential Equations and Boundary Conditions

The following governing equations in Cartesian coordinates have been used for numerical investigations.

Continuity Equation:

\[
\frac{\partial}{\partial x_i} (\rho u_i) = 0
\]  

Momentum equation:

\[
\frac{\partial}{\partial x_i} (\rho u_i u_k) = \frac{\partial}{\partial x_i} \left( \mu \frac{\partial u_k}{\partial x_i} \right) - \frac{\partial p}{\partial x_i}
\]  

Energy equation:

\[
\frac{\partial}{\partial x_i} (\rho u_i T) = \frac{\partial}{\partial x_i} \left( \frac{k}{c_p} \frac{\partial T}{\partial x_i} \right)
\]

Boundary conditions applied to different regions in the computational domain shown in fig. 3 are described as under:

Extended Upstream region:

- At inlet boundary, \( u = u_{in} = \text{constant} \), \( v = w = 0 \), \( T = T_{in} = \text{constant} \)
- At side boundaries, \( \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0 \), \( \nu = 0 \), \( \frac{\partial T}{\partial y} = 0 \)
- At the top and bottom boundaries, Velocity periodic condition \( u_{up} = u_{down} \)
- Temperature periodic condition \( T_{up} = T_{down} \)

Extended downstream region:

- At the outlet boundary, \( \frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0 \)
- At side boundaries, \( \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0 \), \( \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0 \), \( \frac{\partial T}{\partial y} = 0 \)
- At the top and bottom boundaries

Velocity periodic condition \( u_{up} = u_{down} \)

Temperature periodic condition \( T_{up} = T_{down} \)

III. DATA REDUCTION

The mean temperature \( (T_m) \) and mean pressure \( (P_m) \) for a cross section are defined as follows:

\[
T_m = \frac{\iint u T dA}{\iint u dA}
\]
Performance Improvement of a Fin-Tube Heat Exchanger Using Rectangular Winglet as Vortex Generator

\[ p_m = \frac{\int P \, dA}{\int dA} \]  \hspace{1cm} (5)

\( Q \) represents the total heat transfer rate and \( \Delta T_m \) denotes temperature difference.

\[ Q = \dot{m} \cdot C_p \cdot (T_{m,\text{out}} - T_{m,\text{in}}) \]  \hspace{1cm} (6)

\[ \Delta T_m = \frac{(T_{m,\text{wall}} - T_{m,\text{in}}) - (T_{m,\text{wall}} - T_{m,\text{out}})}{\ln\left(\frac{T_{m,\text{wall}} - T_{m,\text{in}}}{T_{m,\text{wall}} - T_{m,\text{out}}}\right)} \]  \hspace{1cm} (7)

Heat transfer coefficient due to convection \((h)\) becomes

\[ h = \frac{Q}{A_v \cdot \Delta T_m} \]  \hspace{1cm} (8)

where, \( \dot{m} \) is mass flow rate of working fluid, \( T_{m,\text{out}} \) is the fluid temperature at outlet, \( T_{m,\text{in}} \) is the fluid temperature at inlet, \( T_{m,\text{wall}} \) is the wall temperature. Specific heat capacity of working fluid is represented by \( C_p \) and \( A_v \) denotes the total surface area of heat transfer.

The hydraulic diameter \((D_h)\) is given by,

\[ D_h = 4 \cdot A_v / A_w \]  \hspace{1cm} (9)

The kinematic viscosity \((\nu)\) is given by,

\[ \nu = \mu / \rho \]  \hspace{1cm} (10)

The hydraulic diameter-based Reynolds number \((Re)\) is given by

\[ Re = uD_h / \nu \]  \hspace{1cm} (11)

In the above equations, \( u \), \( A_v \), and \( A_w \) denote the air inlet velocity, cross-sectional area of fluid flow and perimeter of wetted surface, respectively. \( \mu \) and \( \rho \) are dynamic viscosity and density of air respectively.

The Nusselt number \((Nu)\) is defined by,

\[ Nu = h / D_h \]  \hspace{1cm} (12)

The Prandtl number \((Pr)\) is defined by,

\[ Pr = \mu / C_p / \rho \]  \hspace{1cm} (13)

Darcy Friction factor \((f)\) which determines the friction characteristics is defined by,

\[ f = \Delta P_{in} \cdot D_f / (2 \cdot L \cdot \rho \cdot u^2) \]  \hspace{1cm} (14)

Heat transfer capacity represented by Colburn’s factor \((j)\) is defined by,

\[ j = \frac{Nu}{Re \cdot Pr} \]  \hspace{1cm} (15)

Here, \( \Delta P_m \) and \( L \) denote the pressure drop between inlet and outlet section of the test section and length of plate along the direction of flow, respectively. Performance evaluation criterion (PEC) mentioned below has been used for estimating the combined effect of thermal performance and pressure drop or flow resistance characteristics of the fin-and-tube heat exchanger.

\[ PEC = j / f \]  \hspace{1cm} (16)

IV. NUMERICAL SOLUTION

A. Grid Discretization

A simple cuboidal domain was defined in order to replicate the flow taking place inside a wind tunnel with unequal domain extensions in order to avoid the effect of reversed flow, as encountered by the solver and widely prescribed in literature. Our domain was extended by 45 times the fin pitch \((45F_p)\) towards outlet and 7.5 times the fin pitch \((7.5F_p)\) towards inlet.

Fig. 4 shows how multizone method has been used to create blocks on the fin-plate and divide it into two zones of structured and unstructured elements. While face sizing functions were implemented on the plate and tube surface for generating mesh of very fine resolution, concept of body of influence was employed for defining preferential element sizing on part of the domain in vicinity of the fin-plate and tube.

B. Grid Dependence Study and Experimental Validation

An investigation of effect of number of grid elements on the Nusselt number, as shown in fig. 5 was made in order to conduct the grid dependence study, wherein four types of grids were made. The extra-coarse, coarse, medium and fine grids consisted of 4.1 lakh, 7.58 lakh, 11.54 lakh and 20 lakh elements respectively. Error percentage between extra coarse and coarse grids was 1.972%, that between coarse and medium grids was 1.155%
whereas the error reduced to an insignificant percentage of 0.014% between medium and fine grids. In order to save on computational resources and allied costs without compromising on accuracy of results, the medium grid was chosen for our simulations. Results of grid dependence shown in fig. 5 are for CFD$_2$ case but a similar study was performed for the other cases.

Fig. 5– Grid Dependence Study

In order to save on computational resources and allied costs without compromising on accuracy of results, the medium grid was chosen for our simulations. Results of grid dependence shown in fig. 5 are for CFD$_2$ case but a similar study was performed for the other cases.

Fig. 6– Experimental Validation

Fig. 6 shows a comparison of convective heat transfer coefficient ($h$) and pressure drop ($dP$) values obtained from numerical simulations and experimentation. Results of experimental validation shown in fig. 6 are for CFD$_0$ case because of experimentation being performed on the same case.

V. RESULTS AND DISCUSSION

Objective of this work is to find the optimum configuration of punched out RWPH from among the considered configurations. Comments on overall performance of investigated cases have been made on the basis of area goodness factor, which is a ratio of Colburn’s factor and Friction factor ($j/f$).

A. Selection of optimum configuration

Effect of punching the winglets out of fin-plate surface will now be discussed. Fig. 7a and fig. 7b show variations of pressure drop and Colburn’s factor with Reynold’s number, for punched cases. Fig. 8 portrays contours of pressure for all cases under consideration from both top and bottom view at an inlet flow velocity of 5 m/s. On comparing all the punched cases, we may say that effect of stagnation zone on pressure drop due to hinderance caused by winglet in the path of flow is less significant in punched-out cases. This is primarily because of the fact that presence of a hole on the plate before winglet acts as a vent which provides an alternate path for air to flow towards the outlet. The second factor that plays a major role in evaluating pressure drop across the fin-plate is formation of recirculation zone behind winglet and tube. When seen from top view, we may notice that all the punched cases have a smaller winglet-dominated recirculation zone. Because of the creation of low-pressure zone of an appreciable size at the back of tube, fluid would always have a tendency to fill in the void if given an opportunity. Another important factor is to consider the flow phenomena taking place near the lower face of plate because of venting. The contours showing bottom view of the fin-plate portray how there is addition of a minor low-pressure region at one of the edges of the hole. An interesting phenomenon taking place here is in case of CFD$_U$ configuration. Orientation of the winglet and hole in this case is such that venting promotes fluid to directly hit the tube on its side. The low-pressure zone on the lower face due to hole happens to get formed at very close proximity to the low-pressure zone of tube on its side which further promotes coalescence. This becomes a dominating factor in the punched CFD$_U$ configuration.

Fig. 7– (a) Variation of Pressure drop with Reynold’s number

Fig. 7– (b) Variation of Colburn’s factor with Reynold’s number for all investigated cases

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3877
Performance Improvement of a Fin-Tube Heat Exchanger Using Rectangular Winglet as Vortex Generator

consideration from which, inferences regarding thermal performance of the heat exchanger could be made. Employing the concept of punched-out winglets from plate surface shows the maximum effect in CFD_U configuration. In order to understand the reasons behind these results, temperature contours shown in fig. 9 may be referred to. Contours of CF_U show that winglet dominated recirculation zone formed in case of punched-out winglet is significantly smaller. Due to high velocity air streams, heat transfer characteristics improve at the vicinity of the tube in addition to its wake region. Due to the winglet facing away from tube and being located downstream, high velocity air streams turn more towards the back of the tube in punched-out case. This creates a very small region of higher temperature being formed right behind the winglet, as can be seen in fig. 9. The same attributes to a minor reduction in thermal performance in punched-out case of CFD_U configuration. Along aforementioned lines of reasoning, tube-dominated recirculation zone shows lower temperature values in punched-out case of CFD_U configuration and this accounts for a minor enhancement in thermal performance.

Fig. 8 – Pressure contours for all investigated cases from top and bottom view of fin-plate at inlet velocity of 5 m/s

Fig. 9 – Temperature contours for all investigated cases at inlet velocity of 5 m/s

Fig. 10(a) shows a plot of friction factor (f) against Reynold’s number (Re). Friction factor is directly proportional to pressure drop and inversely proportional to the square of velocity, hence the shape of indicated curves. Due to the nature of proportional relationship, the trend of friction factor is same as that of pressure drop, shown in fig. 7a. The punched-out case of CFD_U configuration exhibits the best performance, when talking in terms of overall pressure drop due to maximum drop in friction factor. Fig. 10b shows a variation of convective heat transfer coefficient (h) with Re. Trend exhibited by these curves is similar to that shown by Colburn’s factor (fig. 7b) because of the direct proportionality of both with Nusselt number.

Fig. 10 – (a) Variation of friction factor with Reynold’s number and

Fig. 10 – (b) Variation of convective heat transfer coefficient with Reynold’s number for all investigated cases

Fig. 11– Variation of performance evaluation criterion with Reynold’s number for all investigated cases

Area goodness factor being considered as the scale for performance evaluation in our study provides the best indication of performance comparison between punched cases. The plot of variation of PEC with Re shown in fig. 11 considers the cumulative effect of Colburn’s factor and friction factor. Although there is a minor enhancement in heat transfer characteristics in
punched case of CFDU configuration, the factor of increment in pressure drop dominates and thus a reduction in PEC is seen in this configuration. It may thus be concluded that punched out winglet having CFDU configuration has best thermohydraulic performance from among the considered configurations.

VI. CONCLUSION

The concept of punching out a winglet being used as vortex generator in a fin-tube heat exchanger has been discussed in the present work. Critical comparisons have been drawn between all the configurations under consideration and the effect of each on thermohydraulic performance of the heat exchanger has been studied. The numerical modeling technique employed in our work pointed towards common flow down at upstream location as exhibiting the best thermohydraulic performance. Critical comparisons have been drawn between all the configurations indicating various parameters effect of each on thermohydraulic performance of the heat exchanger. Development of the concept of punching out a winglet being used among the considered configurations.

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