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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 (j/f). 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

A_c	Cross Section Area (m^2)
A_s	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)
$\dot{m}$	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)
ΔP_m	Mean pressure drop (Pa)
T_m	Mean temp. for a cross section (K)
$T_{m,in}$	Mean temperature at inlet cross section (K)
$T_{m,out}$	Mean temperature at outlet cross section (K)
$T_{m,wall}$	Mean wall temp. of fin surface (K)
ΔT_m	Mean temperature difference (K)

Greek Symbols

μ	Dynamic Viscosity (Pa.s)
ν	Kinematic Viscosity (m^2 s^{-1})
ρ	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.

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[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-ω turbulence model which has a demonstrated capability of capturing flow separation in a superior way than k-ε 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 rig. 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 (F_p), respectively are 52 mm, 300 mm, 200 mm, 3 mm and 24 mm. Longitudinally and transversally, VG has been positioned at 0.5D from the center of the plate. Angle of attack (α), length (L_g), height (H_g) and thickness (t_g) 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.

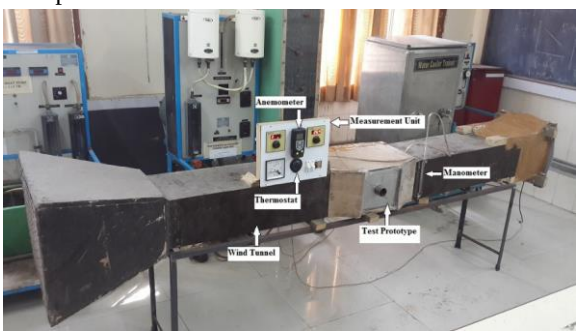

Fig. 1 – Experimental Setup

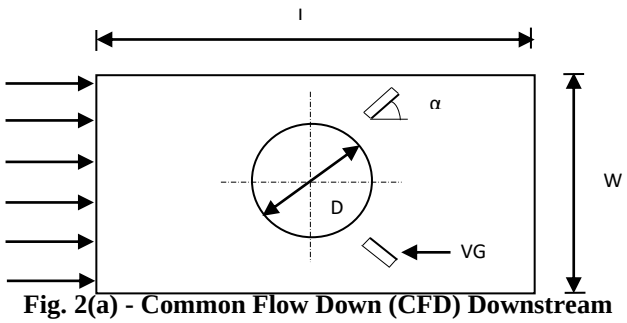

Fig. 2(a) - Common Flow Down (CFD) Downstream

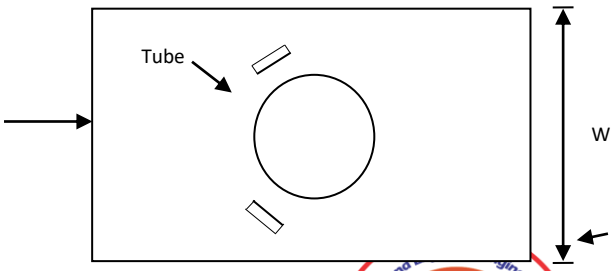

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Fig. 2(b) - Common Flow Down (CFD) Upstream

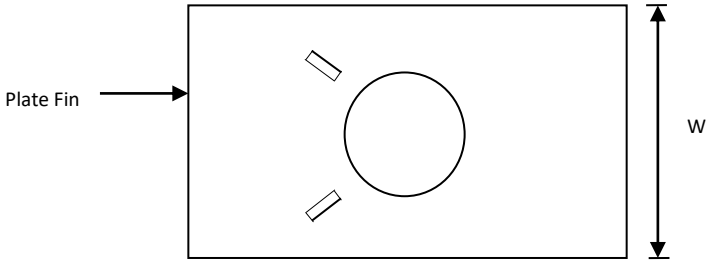

Fig. 2(c) - Common Flow Up (CFU) Upstream

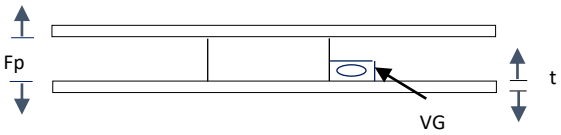

Fig. 2(d) - Geometrical Configuration Of Test Plate (Front View)

B. Governing Differential Equations and Boundary Conditions

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

Continuity Equation:

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

Momentum equation:

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

Energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial T}{\partial x_i}\right) \quad (3)$$

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

Extended Upstream region:

- At inlet boundary, $u = u_{in} = constant, v = w = 0, T = T_{in} = constant$
- At side boundaries, $\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, v = 0, \frac{\partial T}{\partial y} = 0$
- At the top and bottom boundaries,

  Velocity periodic condition $u_{up} = u_{down}$

  Temperature periodic condition $T_{up} = T_{down}$

Fin coil region:

- At side boundaries,

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

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

  Tube region $u = v = w = 0, \; T = T_w = constant$

- At 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, $\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, v = 0, \frac{\partial T}{\partial y} = 0$
- At the top and bottom boundaries

  Velocity periodic condition $u_{up} = u_{down}$

  Temperature periodic condition $T_{up} = T_{down}$

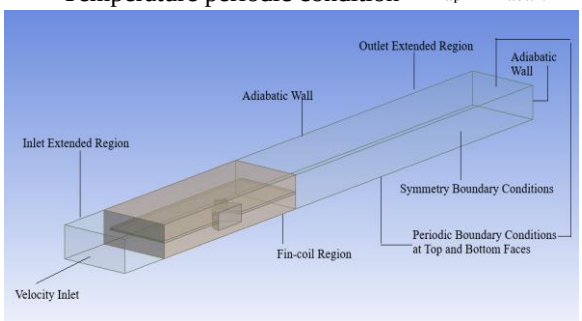

Fig. 3 - Computational Domain

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III. DATA REDUCTION

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

$$T_m = \frac{\iint_A uTdA}{\iint_A udA} \quad (4)$$

$$P_m = \frac{\iint_A PdA}{\iint_A dA} \quad (5)$$

Q represents the total heat transfer rate and ΔT_M denotes temperature difference.

$$Q = \dot{m}.C_p.(T_{m,out} - T_{m,in}) \quad (6)$$

$$\Delta T_M = \frac{(T_{m,wall} - T_{m,in}) - (T_{m,wall} - T_{m,out})}{\ln \frac{(T_{m,wall} - T_{m,in})}{(T_{m,wall} - T_{m,out})}} \quad (7)$$

Heat transfer coefficient due to convection (h) becomes

$$h = \frac{Q}{A_S.\Delta T_M} \quad (8)$$

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

The hydraulic diameter (D_h) is given by,

$$D_h = 4.A_c/P_w \quad (9)$$

The kinematic viscosity (ν) is given by,

$$\nu = \mu/\rho \quad (10)$$

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

$$Re = uD_h/\nu \quad (11)$$

In the above equations, u, A_c and P_w denote the air inlet velocity, cross-sectional area of fluid flow and perimeter of wetted surface, respectively. μ and ρ are dynamic viscosity and density of air respectively.

The Nusselt number (Nu) is defined by,

$$Nu = \frac{h.D_h}{k} \quad (12)$$

The Prandtl number (Pr) is defined by,

$$Pr = \mu.C_p/k \quad (13)$$

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

$$f = \Delta P_m.D_e/(2L.\rho.u^2) \quad (14)$$

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

$$j = \frac{Nu}{Re.Pr^{1/3}} \quad (15)$$

Here, ΔP_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.

$$PEC = j/f \quad (16)$$

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..

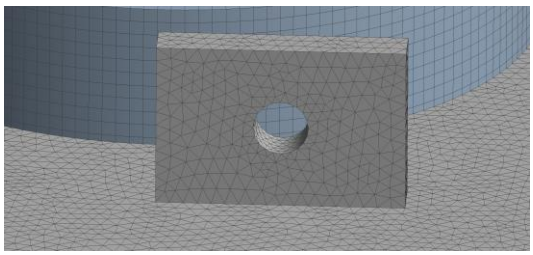

Fig. 4 – Representation Of Multizone Method For Meshed Plate And Winglet

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.

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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_D case but a similar study was performed for the other cases.

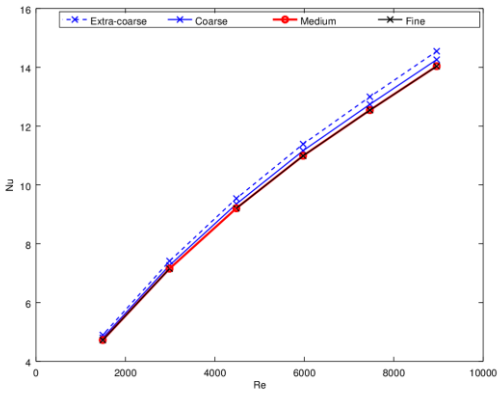

Fig. 5– Grid Dependence Study

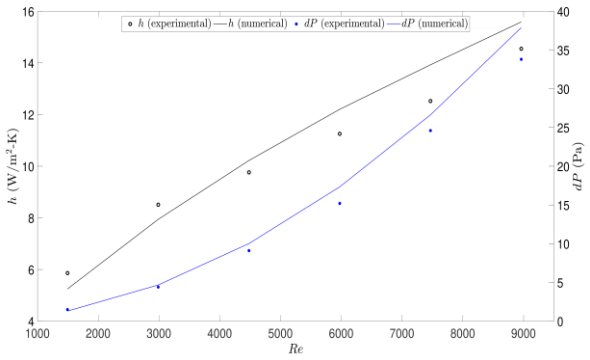

Fig. 6– Experimental Validation

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_D 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.

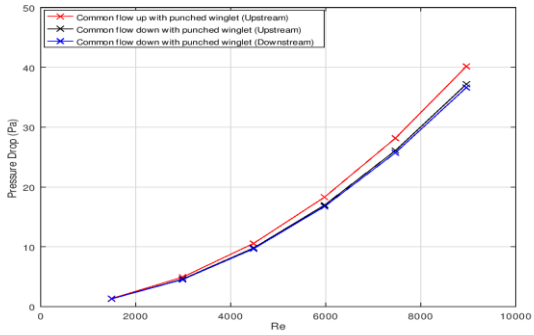

Fig. 7 – (a) Variation of Pressure drop with Reynold's number

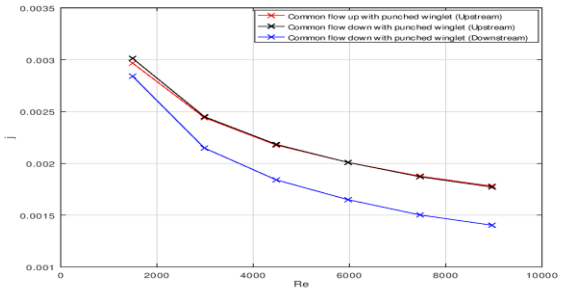

Fig. 7 – (b) Variation of Colburn's factor with Reynold's number for all investigated cases

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Fig. 7b shows a variation of Colburn's factor with *Re* for all the cases under 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 CFU_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_D 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.

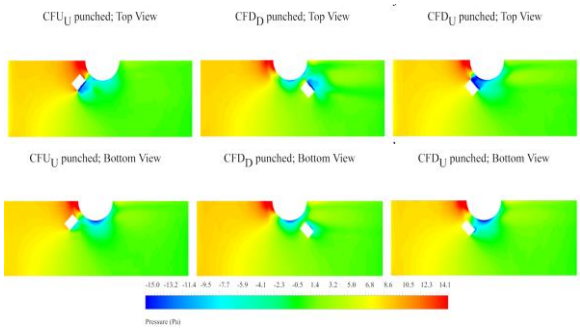

Fig. 8 – Pressure contours for all investigated cases from top and bottom view of fin-plate at inlet velocity of 5 m/s

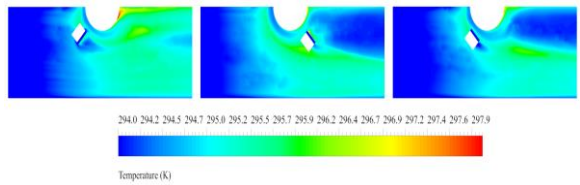

Fig. 9 – Temperature contours for all investigated cases at inlet velocity of 5 m/s

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_D 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.

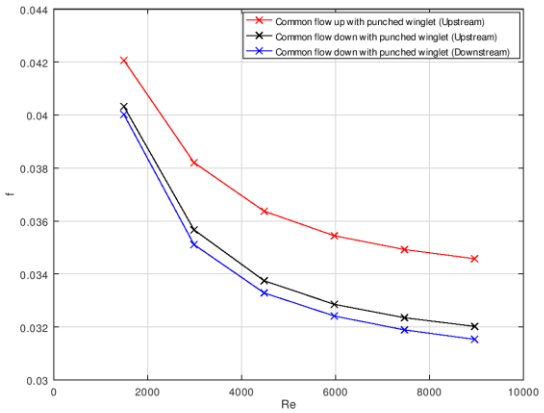

Fig. 10 – (a) Variation of friction factor with Reynold's number and

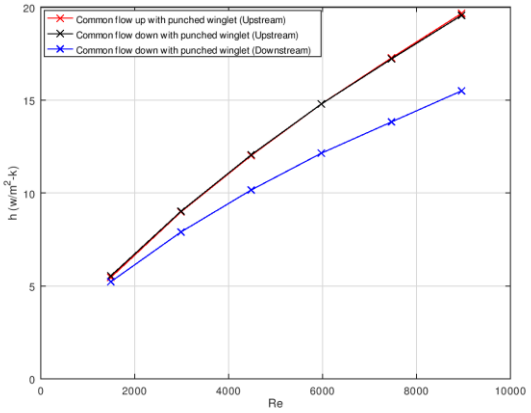

Fig. 10 – (b) Variation of convective heat transfer coefficient with Reynold's number for all investigated cases

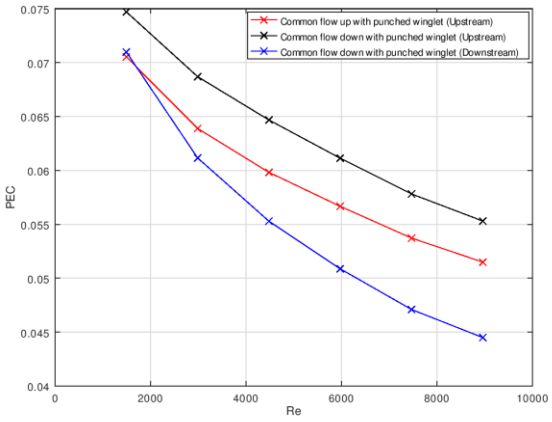

Fig. 11– Variation of performance evaluation criterion with Reynold's number for all investigated cases

Area goodness factor being considered as the scale for performance evaluation in our study provides the best indication of performance comparison between punched cases.

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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 CFD_U 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 CFD_U 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 thermal performance, followed by common flow up at upstream location and, common flow down at downstream location. On the other hand, plate-punching proved to be advantageous in upstream location when considering thermal characteristics The cumulative effect of thermal characteristics and pressure drop was seen with the help of area goodness factor, stated as performance evaluation criterion above. It can thus be concluded that punched winglets with hole having common flow down orientation at upstream location provide maximum enhancement in overall thermohydraulic performance, and can be especially recommended for use in real life applications.

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