



CFD analysis of fin tube heat exchanger using rectangular winglet vortex generator

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Abstract: Among tubular heat exchanger, fin tube types are the most widely used in refrigeration and air conditioning equipment. Efforts to enhance the performance of these heat exchangers included variations in the fin shape from a plain fin to a slit and louver type. In the context of heat transfer augmentation, the performance of vortex generator has also been investigated. Rectangular winglet vortex generators have recently attracted research interest, partly due to experimental data showing that their addition increases the performance of fin-tube heat exchanger. The efficiency of the rectangular winglet vortex generators widely varies depending on their size and shape, as well as locations where they are implemented. The present work represents a two-dimensional numerical investigation of 4-row tube bank in staggered arrangement with rectangular vortex generators placed in common flow up configuration. The effects of the angle of attack (30° and 45°), width (5mm, 5.5mm, 6mm) of vortex generators at position ($\Delta x/\Delta y = 1.3$) are examined. It has been observed that overall Nusselt number of four row tubes increases by 18.68 - 46.38% and 38 - 55.86% with angles of 30° and 45° respectively with increase in pressure drop 9.26-56.35% and 38-190.3% with two angles used respectively, in comparison with case without vortex generators.

Keywords: Fin-tube heat exchanger; Rectangular winglet; Vortex generators; heat transfer enhancement

I. Introduction

Fin and tube heat exchangers are widely used in various engineering fields, such as heating, ventilation, air conditioning, and refrigeration (HVACR) systems. High heat exchanger performance is very important in meeting efficiency standards with low cost and environmental impact. For liquid to air and phase-change heat exchangers that are typical for HVAC&R systems, the air side convection resistance is usually dominant due to thermo physical property of air. Thus many efforts have been made to enhance air-side heat transfer performance and variants of fin patterns like wave, louver and slit fin have been adopted. However, with significant heat transfer enhancement, the associated penalty of pressure drop is also tremendous for those conventional heat transfer enhancement methods.

In recent years, a very promising strategy of enhancing air side heat transfer performance is using flow manipulator, known as vortex generator. When the fluid flows through vortex generators, stream wise vortices are generated in the flow field due to flow separation on the leading edge of vortex generators (VGs), causing bulk flow mixing, boundary-layer modification, and flow destabilization; heat transfer is enhanced due to these vortices. The longitudinal vortex generators applied in various heat exchangers have received considerable attentions for a modest pressure drop penalty.

The vortex generators incorporated in fin-tube heat exchangers can be categorized based on their configurations with respect to heat transfer tubes, whether 'common flow down' and 'common flow up' as illustrated in fig. 1. The common flow down configuration has pair of vortex generators mounted behind a tube (at the downstream). Fiebig [1] contended that this configuration would delay flow separation and reduce drag. In common flow up configuration, a pair of vortex generators is installed in front of the tube (at the upstream). Torri et al. [2] speculated that this configuration accelerates the flow between a tube and its vortex generators as well as delays flow separation; thereby reducing the wake region behind the tube.

Fiebig et al. [3] experimentally investigated three tubes rows heat exchanger element with delta winglets. They reported a 55-65% heat transfer augmentation with corresponding 20-45% increase in the apparent friction factor for an inline arrangement. Biswas et al. [4] performed numerical investigations of flow structure and heat transfer characteristics in a channel with a built-in tube and a winglet type vortex generator. Fiebig et al. [5] measured a 10% local heat transfer enhancement for round tubes but a 100% enhancement for flat tubes under certain geometry. Deb et al. [6] numerically investigated heat transfer characteristics and flow structure in laminar and turbulent flows through a rectangular channel containing built in vortex generators. Jacobi and Shah [7] reviewed previous works until the early 1990s. They explained physical phenomena and vortex characteristics associated with vortex generators on a fin as well as introduced associated with experiments and analysis on the performance of heat exchangers with vortex generators. They categorized active and passive

methods of vortex generation. Active methods generate vortices using external energy, such as electric or acoustic fields, mechanical device, or surface vibration. Passive methods generate vortices through structures and additional fluids. Regardless of methods, vortex generators enhance heat transfer and simultaneous pressure drop due to form of loss.

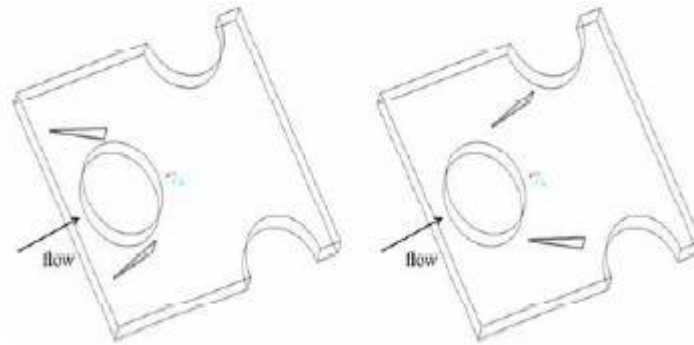


Fig.1. Vortex generator configuration: (a) common flow up; (b) common flow down

Biswas *et al.* [8] found that the flow behind a winglet type vortex generator is consists of main vortex, a corner vortex and induced vortices. Lee *et al.* [9] carried out a numerical study of heat transfer characteristics and turbulent structure in a three dimensional boundary layer with longitudinal vortices. Chen *et al.* [10] numerically investigated heat transfer and flow in an oval-tube heat transfer element in both in-line and staggered arrangements with punched winglets type vortex generators. Wang *et al.* [11] utilized a dye-injection technique to visualize the flow structure for enlarged plain fin and tube heat exchanger with annular and delta winglet vortex generators. Torri *et al.* [12] proposed a common flow up arrangement strategy which can augment heat transfer while reducing pressure penalty in a fin and tube heat exchanger at relatively low Reynolds number. Leu *et al.* [13] numerically and experimentally studied the heat transfer and flow in the plate fin and tube heat exchangers with inclined block shape vortex generators mounted behind tubes. Wu and Tao [14] conducted a numerical simulation to investigate the heat transfer performance of a rectangular channel with pair of rectangular winglets VGs punched out from the wall of the channel. They analyzed the results from the view of field synergy principle.

Joarder and Jacobi [15] experimentally assessed the potential of winglet type vortex generator “arrays” for multi-row inline-tube heat exchangers. They found that the air-side heat transfer coefficient increased from 16.5% to 44% for the single row winglet arrangement with an increase in pressure drop of less than 12%; for three row vortex generator array the heat transfer coefficient increases from 29.9% to 68.8% with pressure drop penalty from 26% to 87.5%. Joarder and Jacobi [16] also numerically investigated the flow and heat transfer enhancement using an array of VGs in a fin and tube exchanger with common flow up arrangement. They observed that the impingement of winglet redirected flow on the downstream tube is an important heat transfer augmentation mechanism for the inline tube geometry.

In this work, numerical study of fin tube heat exchanger with rectangular VG arranged in common flow up configuration are performed. Effects of attack angle, width and placement location of winglet on the fluid flow and heat transfer characteristics were examined.

Nomenclature

A_c	minimum flow cross-sectional area (m ²)
A_f	fin surface area (m ²)
A_o	total surface area (m ²)
D_o	tube outer diameter (m)
D_c	fin collar diameter (m) = $D_o + 2\delta$
D_h	hydraulic diameter, $4A_cL/A_o$ (m)
F_p	fin pitch (m)
P_l	longitudinal tube pitch (m)
P_t	transverse tube pitch (m)
L	fin length along the main flow direction (m)
w	Length of rectangular winglet
f	friction factor
Δp	pressure drop
Nu	Nusselt number
Re_{Dc}	Reynolds number based on tube collar diameter

Re_{Dh}	Reynolds number based on hydraulic diameter
Pr	Prandtl number
K	Thermal conductivity ($W\ m^{-1}\ K^{-1}$)
T	Temperature (K)
T_{in}	Inlet temperature (K)
T_{out}	Outlet temperature (K)
V	Velocity (m/s)
V_{max}	maximum velocity (m/s)
C_p	specific heat of air (KJ/Kg^*K)
Greek symbols	
α	attack angle of winglet (deg)
∂_f	fin thickness (m)
μ	dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
Subscripts	
a	air
f	fin
in	inlet
out	outlet

II. Model description

A. Physical Model

Fig. 2 shows the representative geometry of a four-row fin-tube heat exchanger considered for the numerical analysis. Rectangular winglets are installed around the tubes in common flow up configuration. Tubes are in staggered arrangement. Fin thickness, fin spacing, and tube size are similar to those of fin tube heat exchangers that are widely used in condensers of industrial refrigeration system. Due to the symmetric arrangement, the region occupied by dashed line is selected for computational domain, which is, considered as a channel of height $H = 15.875\ mm$ and $L = 111.76\ mm$.

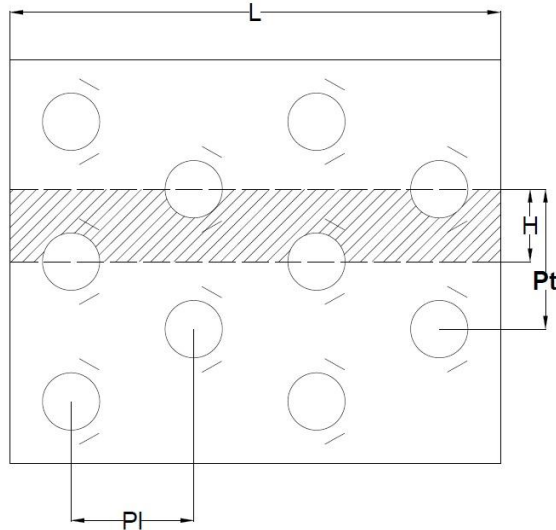


Fig. 2: Schematic diagram of computational domain

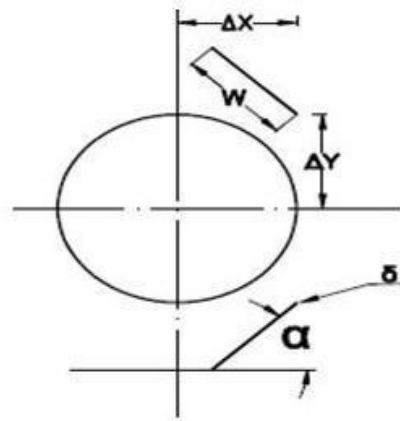


Fig. 3: Tube and VGs parameters

The tube outside diameter D_o is 12.7 mm, the transverse pitch P_t is 1.25 inch, the longitudinal pitch P_l is 1.1 inch, the fin pitch F_p is 2.12 mm, and the fin thickness ∂_f is 0.15mm. The length w of rectangular winglets has three different values ($w = 5\ mm, 5.5\ mm$ and $6\ mm$), thickness $\partial = 0.15$ and are located at ($\Delta X/\Delta Y = 1.3$). Also the angles of attack α of winglets are varied by 30° and 45° . The actual computational domain is extended by 30 mm at exit to ensure a recirculation-free flow there. The tube and rectangular vortex generators parameters are shown in fig. 3

B. Boundary conditions

Air flow is assumed as incompressible and turbulent. RNG k- ϵ model was used as a turbulence model, considering complex flows (wake or separation) and anisotropic turbulence. All the boundary conditions are given in fig. 4.

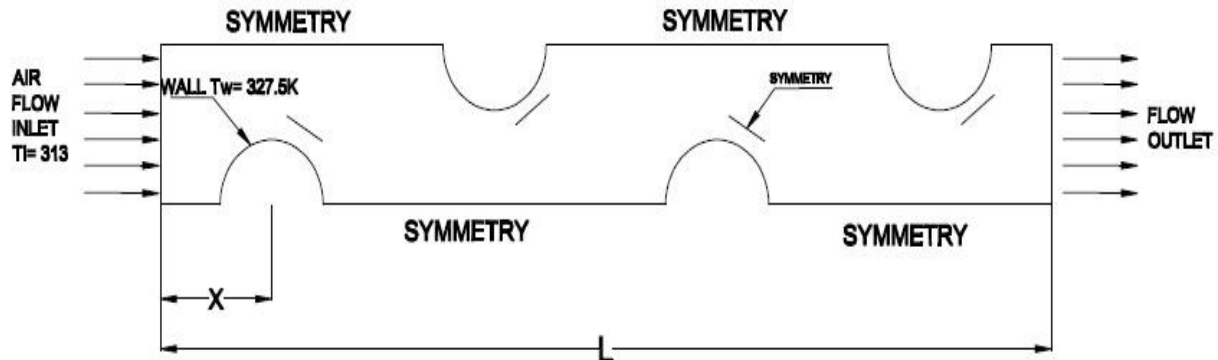


Fig. 4: The computational domain and boundary conditions.

The boundary conditions are as follows. An inlet velocity boundary condition (3.8 m/s) is established for the front end of the fluid domain, whereas a pressure outlet boundary condition is used for the rear end. Symmetric boundary condition is specified at both side-ends of flow path. A constant temperature condition was set at the inner tube wall. The temperature of the solid surface was set such that the heat flux through the solid part balances with that through the adjacent air. Top and bottom surface were set to adiabatic. The air inlet temperature was fixed at 313 K for all simulations. The heat transfer tube wall temperature was set at 327.5 K to represent working condition of refrigerant condenser. Symmetric boundary condition is used for Rectangular winglet VG.

C. Governing equations

The governing equation in Cartesian coordinates can be expressed as follows:

$$\text{Continuity equation: } \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0 \quad (1)$$

$$\text{Momentum equations: } \rho \left[U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} \right] = -\frac{\partial P}{\partial x} + \mu \left[\frac{\partial^2 U}{\partial x^2} + \frac{\partial^2 U}{\partial y^2} \right] \quad (2.1)$$

$$\rho \left[U \frac{\partial V}{\partial x} + V \frac{\partial V}{\partial y} \right] = -\frac{\partial P}{\partial y} + \mu \left[\frac{\partial^2 V}{\partial x^2} + \frac{\partial^2 V}{\partial y^2} \right] \quad (2.2)$$

$$\text{Energy equation: } \rho C_p \left[U \frac{\partial T}{\partial x} + V \frac{\partial T}{\partial y} \right] = k_f \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right] \quad (3)$$

D. Parameter definitions

The following expressions analysed the data. The heat transfer coefficient is related with the temperature difference, as follows:

$$Q = \dot{m} C_p (T_{in} - T_{out}) \quad (4)$$

$$h = \frac{Q}{(A_c + A_{fin}) \Delta T_{LM}} \quad (5)$$

$$\Delta T_{LM} = \frac{(T_c - T_{in}) - (T_c - T_{out})}{\ln \left[\frac{T_c - T_{in}}{T_c - T_{out}} \right]} \quad (6)$$

$$Nu = \frac{h D_c}{K} \quad (7)$$

$$\Delta p = p_{in} - p_{out} \quad (8)$$

$$f = \frac{\Delta p}{\frac{1}{2} \rho V_{max}^2} \left(\frac{D_c}{4L} \right) \quad (9)$$

Where $\dot{m}$, A_c , A_{fin} , T_{in} , T_{out} , T_c , D_c , Δp and Q represent the mass flow rate, fin collar area, fin surface area, inlet air temperature, outlet air mean temperature, fin collar temperature, pressure drop and total heat transfer rate respectively. ΔT is the log-mean temperature difference and f is the friction factor.

III. Validation of model and numerical method

The commercial computational fluid dynamics code FLUENT was used to solve the governing equations, which were discretised by the finite volume method. The simple (Semi Implicit Method for Pressure –Linked Equations) algorithm was used for the velocity-pressure coupling. The second-order upwind scheme was employed for discretization of the convection terms. An unstructured quadrilateral mesh was generated using Gambit, with fine mesh around the tube and a coarse mesh in the extended regions having mesh size of 63,007 as shown in the fig.5.

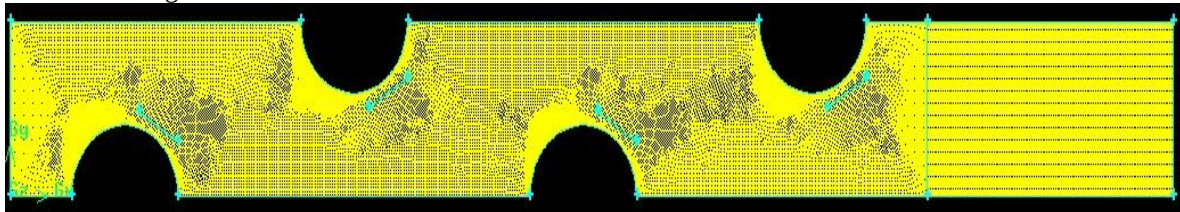


Fig.5. Computational model at 30° attack angle and winglet length 5.5mm in Gambit

The inlet air velocity is 3.8 m/s and the corresponding Reynolds number is 2908. The discrepancy between the predicted Nusselt number and that calculated using Grimison's correlations [17] is 13% and discrepancy between the predicted pressure drop and that calculated using Wang's correlations [18] is less than 2%. The excellent agreement between the numerical results and correlation data indicates the present numerical results are reliable to predict the heat exchanger performance.

IV. Results and discussion

A. Influence of VGs

In order to study the influence of VGs on the heat transfer characteristics and the fluid flow over a bank of tubes, a comparative investigation for four tube-row model with and without VGs is performed. Firstly, the heat exchanger model without VGs will be referred to as “baseline” case and heat exchanger with VGs will be referred to as “modified” case as shown in fig.6.

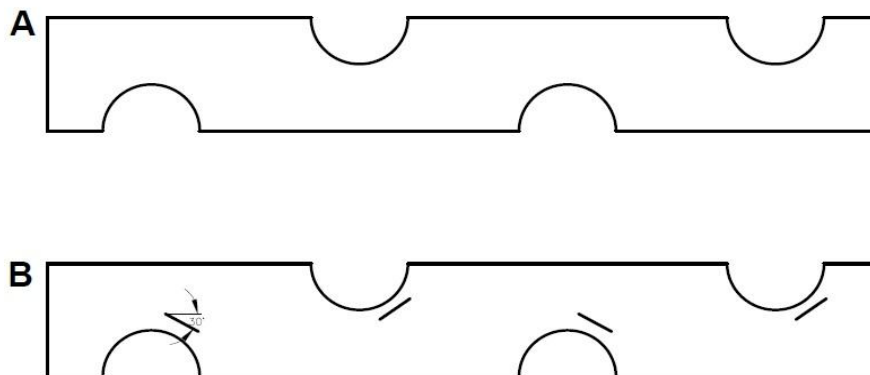


Fig.6. Different configuration for 4-rows tube bank with and without VGs: (A) Baseline case, (B) Modified case

For the comparison test, the VGs are placed in the position of $\Delta X/\Delta Y = 1.3$, length of VG = 5.5mm and the angle of attack is set as 30°. The Re number based on hydraulic diameter is 2908. From the CFD simulation it is found that the overall Nu number for the modified case is improved by 38.50% in comparison with baseline case.

Fig.7. illustrates the temperature profiles for baseline case and modified case with two angles of attack respectively for VG length = 5.5mm and Re = 2908. Comparing the three temperature profiles, we can see that

the temperature distribution in the vicinity of the inlet region is almost identical for all cases. As the air approaches the VGs, the stream wise vortices are generated and the heat transfer from the tubes is significantly enhanced. As we see, the temperature profiles around the tube surface are different between the baseline case and modified cases due to the stream wise vortices of VGs and flow acceleration in the region between the VGs and oval-tubes. Thus, the temperature behind the tube surface for modified cases is distinctly lower than in the corresponding region for baseline case.

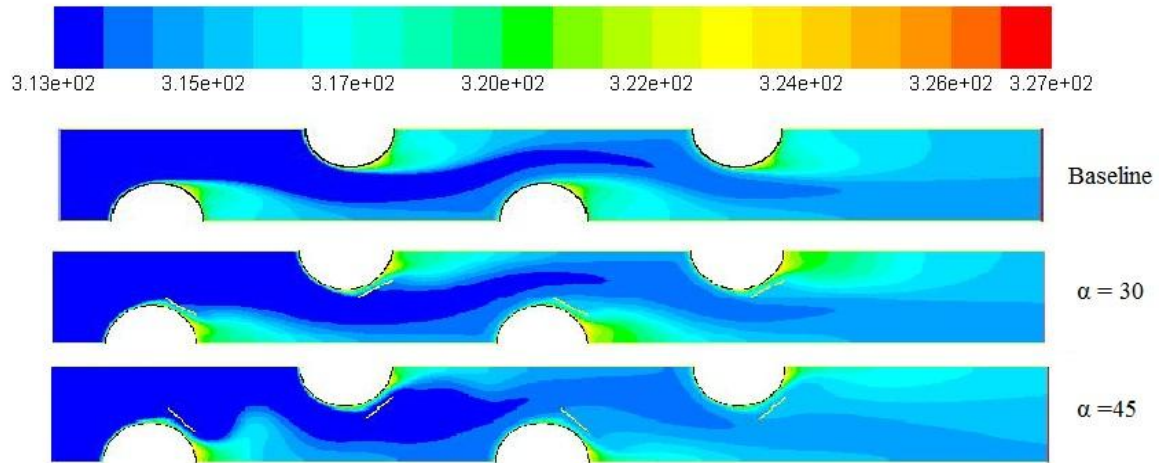


Fig.7. Temperature distribution on the four rows tube banks for both baseline and modified cases at position $\Delta X/\Delta Y = 1.3$, VG length = 5.5mm and two angles of attack, for $Re = 2908$.

B. Effect of Winglet Size and Angle of Attack on Heat Transfer (Nu)

The influence of both winglet size and angle of attack of rectangular VGs on the heat transfer and fluid flow characteristic for the tube banks is investigated. Fig.8. shows that the overall Nu number, for modified cases at position ($\Delta X/\Delta Y = 1.3$) increases with increasing winglet length w and angle of attack α . The enhancement in overall Nu number with angle of attack of 45° is more as compared with angle of attack 30° . The overall Nu number for three different winglet sizes (5mm, 5.5mm and 6mm) increases by 18.68-46.38% with angle of attack 30° and by 38-55.86% with angle of attack 45° comparing with baseline case.

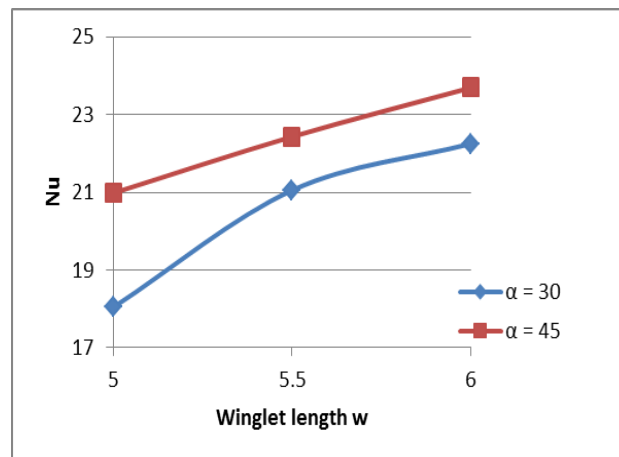


Fig.8. Effect of winglet size and angle of attack on heat transfer

There are some reasons leading to this high enhancement at these parameters of VGs. One of these is influence of vortex generators, which reduce the size of wake region behind each tube and flow will be accelerating at this narrow region. In addition, the vortices generated by VGs around the first tube row will interact with the vortices generated by VGs around the second, third and fourth rows.

C. Effect of Winglet Size and Angle of Attack on Pressure Drop (ΔP)

The effects of winglet size and angle of attack on the pressure drop are studied. As we know that the flow in the channel with tubes has a significant fluid friction in comparison with the smooth channel. Fig.9 and Fig.10. shows that pressure drop for modified cases at position ($\Delta X/\Delta Y = 1.3$) increases with increasing winglet length

w and angle of attack α . It is observed that with the increase in winglet size w and attack angle α , pressure drop also increases. The pressure drop for three different winglet sizes (5mm, 5.5mm and 6mm) increases from 9.26-56.35% for angle of attack 30° and increases from 81.57-190.30% for 45° comparing with baseline case.

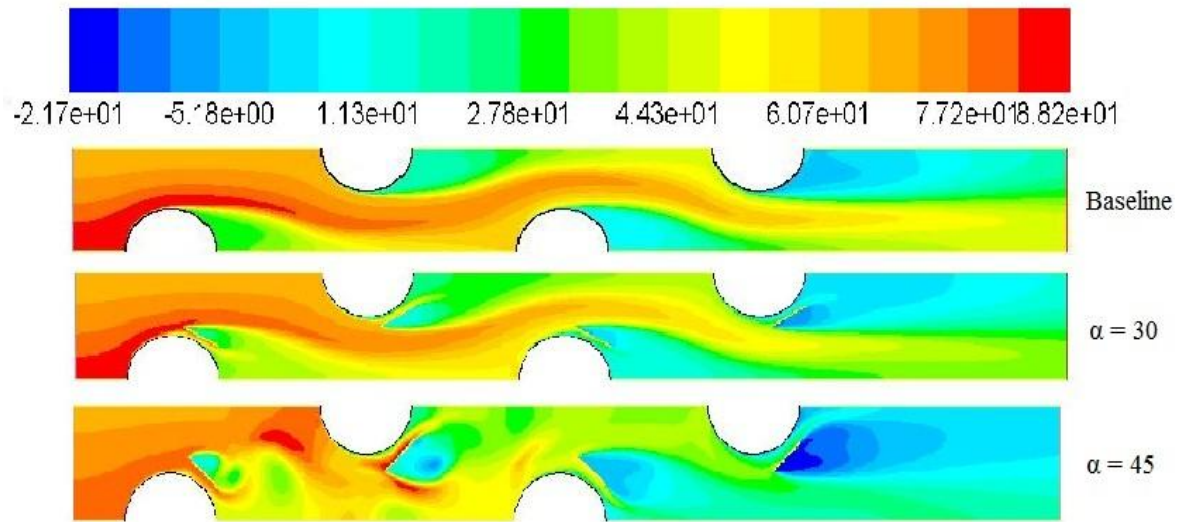


Fig.9. Pressure drop on the four rows tube banks for both baseline and modified cases at position $\Delta X/\Delta Y = 1.3$, VG length = 5.5mm and two angles of attack, for $Re = 2908$.

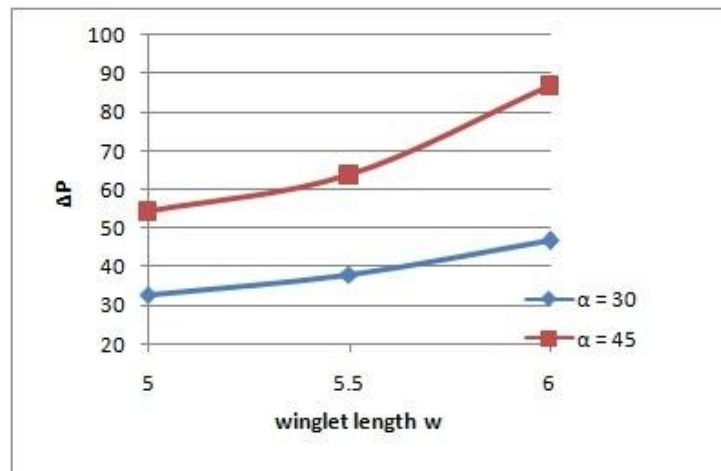


Fig.10. Effect of winglet size and angle of attack on Pressure drop.

V. Conclusions

The fluid flow and heat transfer over a four rows tube bank in staggered arrangement with variation of parameters of VGs, for Reynolds number 2908, were studied numerically. The main results include:

1. The VGs enhance the heat transfer on the tube bank. Scanty difference in enhancement of heat transfer obtained by varying the winglet size and angle of attacks of VGs, where the study shows that heat transfer (Nu) and pressure drop (ΔP) increases with increase in winglet size and angle of attack.
2. For four-rows of tube bank with VGs having three different length (5mm, 5.5mm and 6mm), the overall Nu number is augmented by 18.68-46.38% with angle of attack 30° and by 38-55.86% with angle of attack 45° and the pressure drop is increased by 9.26-56.35% for angle of attack 30° and by 38-190.30% for angle of attack 45° in comparison with baseline case.
3. The best results are obtained with winglet length of 5.5mm and angle of attack 30° .

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