CFD Simulation of Heat Transfer Enhancement by Plain and Curved Winglet Type Vertex Generators with Punched Holes

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ABSTRACT: CFD simulations were carried out to investigate the performance of plane and curved trapezoidal winglet type vortex generators (VGs). Effects of shape of the VGs on heat transfer enhancement were evaluated using dimensionless numbers - j/j_0, f/f_0 andh $=(j/j_0)/(f/f_0)$. The results showed that curved winglet type VGs have better heat transfer enhancement than plain winglet type in both laminar and turbulent flow regions. The punched holes really reduce the value of friction factor. The flow resistance is also lower in case of curved winglet type than corresponding plane winglet VGs.The best results for heat transfer enhancement is obtained at high Reynolds number values (Re > 10000) by using VGs.The processes in solving thesimulation consist of modeling and meshing the basicgeometry of rectangular channel with VGs using the package ANSYS ICEM CFD 14.0.Then the boundary condition will be set according to the experimental data available from the literature. Finally result has been examined inCFD-Post. This work presents a numerically study on themean Nusselt number, friction factor and heat enhancementcharacteristics in a rectangular channel having a pair of winglet type VGs under uniform heat flux of 416.67 W/m^2. The results indicate the advantages of using curved winglet VGs with punched holes for heat transfer enhancement

Keywords:Heat transfer enhancement, Vertex generators, Winglet types, punched holes Rectangular channel, Numerical investigation, CFD, Flow simulation

INTRODUCTION

Computational Fluid Dynamics (CFD) is a useful tool in solving and analyzing problems that involve fluid flows and heat transfer to the fluid. As a kind of passive heat-transfer enhancing devices, vortex generators (VGs) have been widely investigated to improve the convective heat transfer coefficient (usually the air-side) of plate fin or finned tube type heat exchangers. The basic principle of VGs is to induce secondary flow, particularly longitudinal vortices (LVs), which could disturb the thermal boundary layer developed along the wall and ensure the proper mixing of air throughout the channel by means of large-scale turbulence [1]. Among the various types of VGs, wings and winglets have attracted extensive attention since these VGs could be easily punched or mounted on the channel walls or fins and could effectively generate longitudinal vortices for high enhancement of convective heat transfer. However, the heat transfer enhancement (HTE) by LVs is usually accompanied with the increase of flow resistance. Experimental research by Feibig et al. [2] showed the average heat transfer in laminar channel-flow was enhanced by more than 50% and the corresponding increase of drag coefficient was up to 45% by delta and rectangular wings and winglets. Further experiment with double rows of delta winglets in transitional channel flow byTiggelbeck et al. [3] showed that the ratio of HTE and drag increasewas larger for higher Reynolds numbers. Feibig [4] also pointed outthat the winglets are more effective than wings, but winglet form isof minor importance.Recently, Tian et al. [5] performed three dimensional simulationson wavy fin-and-tube heat exchanger with punched deltawinglets in staggered and in-line arrangements and their resultsshowed that each delta winglet generates a downstream mainvortex and a corner vortex. For Re = 3000, compared with the wavyfin, the Colburn j-factor and friction f-factor of the wavy fin withdelta winglets in staggered and in-line arrays are increased by13.1%, 7.0% and 15.4%, 10.5%, respectively. Chu et al. [6] numericallyinvestigated the three row fin-and-oval-tube heat exchanger withdelta winglets for Re = 500-2500. They reported that, comparedwith the baseline case without LVGs, the average Nu with LVGs was increased by 13.6-32.9%

NOMENCLATUR

A_c	cross sectional area of air channel (m^2)
A_i	heat transfer area of each small element on copperplate (m^2)
A_p	heat transfer area of copper plate in tested channel(m^2)
b	width of vortex generator (mm)
CRWP	curved rectangular winglet pair
CTWP	curved trapezoidal winglet pair
RWP	rectangular winglet pair
TWPH	trapezoidal winglet pair with holes
CTWPH	curved trapezoidal winglet pair with holes
HTE	heat transfer enhancement
C_p	specific heat (J /kg ^oC)
D	hydraulic diameter of the air channel (m)
f	Darcy friction factor
f_0	Darcy friction factor of smooth channel (i.e. withoutVG)
h	height of VG-trailing edge (mm)
h_c	convective heat transfer coefficient (W /$m^{2o}C$)
j	Colburn factor
j_0	Colburn factor of smooth channel (i.e. without VG)
l	length of vortex generator (mm)
L	length of tested channel along air flow direction (m)
LVG	longitudinal vortex generator
Nu	Nusselt number
p	pressure (Pa)
P	electric power (W)
Pr	Prandtl number
Q	heat transfer rate (W)
Re	Reynolds number
S_1	front edge pitch of a pair of vortex generators (m)
S_2	distance of vortex generator pair downstream
T	temperature (K)
U	velocity (m/s)
VG	vortex generator
K	thermal conductivity

Greek letters

α	inclination angle of VG (^{o})
β	attack angle (^{o})
ρ	density (kg m_3)
ΔP	pressure drop (Pa)
μ	dynamic viscosity (Pa s)
η	thermal enhancement factor

Subscripts

a	air
c	cross section or convective

e	effective
E	expanded
i	number of thermocouple or element
in	inlet
m	average
out	outlet
w	wall

and the corresponding pressure drop was increased by 29.2-40.6%, respectively.The above experimental and numerical results show that pressure drop penalty is comparative with the heat transfer enhancement caused by the LVGs. Under some conditions, the increase of pressure drop can be even 2-4 times higher than the heat transfer enhancement by LVGs [7-9], which weakens the advantages of LVGs. Chen et al. [10] pointed out that the form drag of the LVGs is predominant for the pressure drop, and the LVs themany additional pressure drop of the flow. As to the latter part i.e., the drag generated from the friction between the LVs and wallsurface, Wang et al. [11] reported that the transverse expansion ofLV takes the major part of the reason. After the flow separation atthe edge of VG, there exists a low-speed recirculation zone at theback of the LVG, which dissipates the kinetic energy. This is themain source of the form drag and increases with the increase ofattack angle against the flow. Therefore, the effort to diminish thelow-speed recirculation zone is important to decrease the formdrag and then the overall pressure drop caused by the LVG. Minet al. [12] developed a modified rectangular LVG obtained by cuttingoff the four corners of a rectangular wing. Their experimentalresults of this LVG mounted in rectangular channel suggested thatthe modified rectangular wing pairs (MRWPs) have better flow andheat transfer characteristics than those of rectangular wing pair(RWP).Xie and Ye [14] presented an injection flowmethod to reduce theform drag of the airship, i.e. an injection channel is made from theleading to the trailing of the airship. Part of the outer flow is conductedto flow through the injection channel. The simulation resultsshowed that the drag coefficient of the airship is reduced sharply. Ifthe radius of the injection channel is 1/15 of themaximum thicknessof the airship, the drag coefficient is reduced to 32.7% of the originalvalue. Tang et al. [15] numerically investigated the fluid flowand heattransfer by trapezoidal tab with and without clearance using theRealizable k-ε model. Their results showed that the overall performanceof trapezoidal tab with clearance is higher than that withoutclearance and the formdrag is reduced.Habchi et al. [16] numericallyinvestigated the performance of trapezoidal wing with excavation atthe bottom. The results showed that the excavation really reduced theflow resistance, but on the other side, the cavity also reduces thecontact surface between the heatedwall and the vortex generator andthus reduces the conduction heat flux through the vortex generator;as a result, convective heat transfer between the vortex generator andthe surrounding fluid is decreased. Therefore, the size of the cavityshould be optimized to maximize the effect of heat transferenhancement and flow resistance reductionNumerical study by Biswas and Chattopadhyay [17] on deltawing with punched hole in base wall showed that heat transferenhancement and friction factor (f × Re) at the exit are bothrelatively lower than those of the case without any punchedhole. Wu and Tao [18] also did numerical study on thermalhydraulicperformance of rectangular winglets with punchedholes at the channel wall and found that the case withpunched holes has slightly higher average Nu number (about1.1%) and slightly lower average friction factor (about 1.2%)compared with the case without punched holes. Both theabove two papers dealt with punched holes just located infront of the folding line (baseline) of wing or winglet VG.Whereas the low-speed recirculation zone is just behind theVG where the heat transfer and flow drag are slightly influencedby the front holes.To address the heat transfer enhancement in recirculation zoneas well as flow drag reduction, the present paper attempt to punchholes within the plane winglets as well as recently developedcurved winglets and experiments were performed to examine theeffect of these kinds of VGs on air-side heat transfer enhancementand flow resistance in channel flow. Then, the average convectiveheat transfer coefficient was measured and dimensionless numberse j/j0, f/f0 and thermohydraulic performance factor (j/j0)/(f/f0)were used for performance evaluation. The effect of size and positionof the holes on the performance of these VGs were thenevaluated. In our previous work the simulation of heat transfer enhancement was carried out using RWP & CRWP without punched holes.

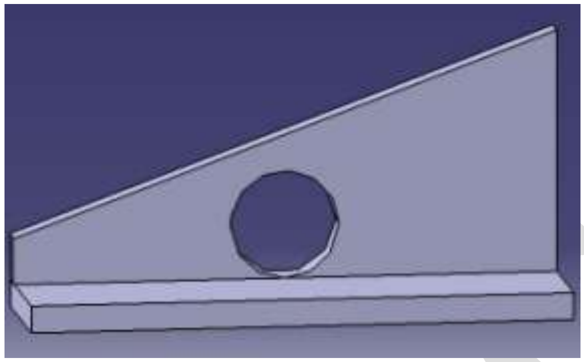

Fig: 1.1 Trapezoidal Winglet with hole (TWH)

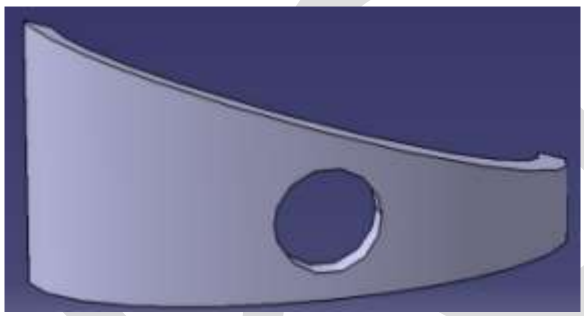

Fig: 1.2 Curved Trapezoidal Winglet with hole (CTWH)

2. NUMERICAL SIMULATION

2.1 Physical Model. The numerical simulations were carried out using FLUENT V6 Software that uses the finite-volume method to solve the governing equations. Geometry was created in CATIA Design tools for air flowing through an electrically heated rectangular channel with copper plate at bottom of 1000mm×300mm and the dimension of the channel is 1000mm×240mm×40mm. Meshing has been created in ICEM CFD 14.0 with tetrahedral shapes (Fig. 2). In this study Reynolds number varies from 750 to 21000.

2.2 Numerical Method. For turbulent, steady andincompressible air flow with constant properties .Wefollow the three-dimensional equations of continuity,momentum and energy, in the fluid region.

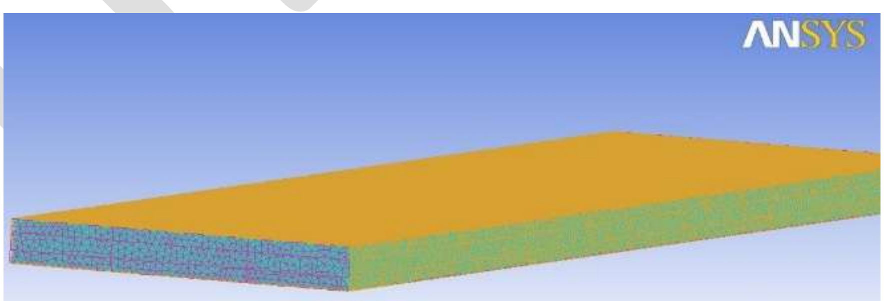

Fig. 2.1 Meshing in ICEM CFD 14.0

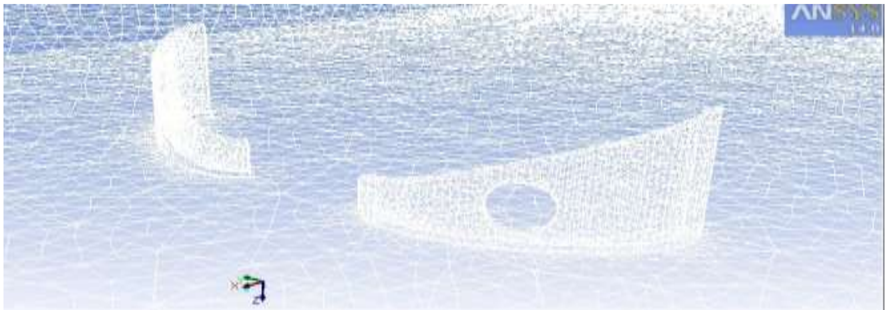

Fig. 2.2 Meshing of CTWPH

These equations are below:

Continuity equation:

$$\frac{\partial p}{\partial t} + \nabla.(\rho \mathrm{V}) = 0 \ldots\ldots. (1)$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_j}{\partial x_i}\right) - \frac{\partial p}{\partial x_j}. \ldots\ldots (2)$$

Energy equation:

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

Table 1.1 Properties of air at 25₀C

Properties	value
Specific heat capacity, C_p	1006 J/kg K
Density,	ρ 1.225 kg/m_3
Thermal conductivity, k	0.0242W/m K

Velocity and pressure linkage was solved by SIMPLEalgorithm. For validating the accuracy of numericalsolutions, the grid independent test has been performedfor the physical model. The tetrahedral grid is highlyconcentrated near the wall regions and also near the VGs

Table 1.2 Nodes and Element in geometry are below.

Geometry type	Nodes	Elements
Smooth	196412	884899
TWPH	199461	994569
CTWPH	203820	997590

Table 1.2 shows that CTWPHhave maximum nodes and element in comparison of smooth and TWPH. In addition, a convergencecriterion of $10^{-7}$was used for energy and 10^{-3} for the massconservation of the calculated parameters. The air inlet

temperature was specified as 293 K and three assumptionswere made in model: (1) the uniform heat flux was along the length of rectangular channel. (2)Wall of the channel will be perfectly insulated. (3) Steady and incompressible flow.In fluent, inlet was taken as velocity-inlet and outlet was taken as pressure-outlet.

3. Data Reduction. Three important parameters wereconsidered-friction factor, Nusselt number and thermalperformance, which determined the friction loss, heattransfer rate and the effectiveness of heat transfer enhancement in the rectangular channel, respectively.

The friction factor (f) is investigated from pressure drop, ΔP across the length of rectangular channel (L) using thefollowing equation:

$$f = 2\Delta pD / (LU^2\rho) \qquad ..... \ (4)$$

The Nusselt number is defined as

$$Nu_L = \frac{hL}{k} = \frac{convective\ heat\ transfer}{conductive\ heat\ trasnfer} \ .. (5)$$

The Nusselt number and the Reynolds number were basedon the average of the channel wall temperature and the outlettemperature, the pressure drop across the test section, andthe air flow velocity were measured for heat transfer of the heated wall with different kind of VGs. Theaverage Nusselt numbers and friction factors wereobtained and all fluids properties were found at the overallbulk mean temperature.

Thermal performance factor was given by:

$$\eta = (j/j_0)/(f/f_0) \qquad ...(6)$$

Where $j_0$and j and f and f_0 were the Colburn factor and friction factors for the smooth channel and channel with VGs respectively.

3. RESULTS AND DISCUSSION

3.1 Validation of setup. The CFD numerical result of the smooth channel without any VG beenvalidated with the experimental data as shown in Figures3.1 and 3.2. These results are within ±8% deviation for heat transfer (Nu) and ±3% the friction factor (f) witheach-other. In low Reynolds number the deviation becomesmall in experimental and CFD results but when Reynoldsnumber become more then these deviation slightly higherin experimental and CFD results, respectively.

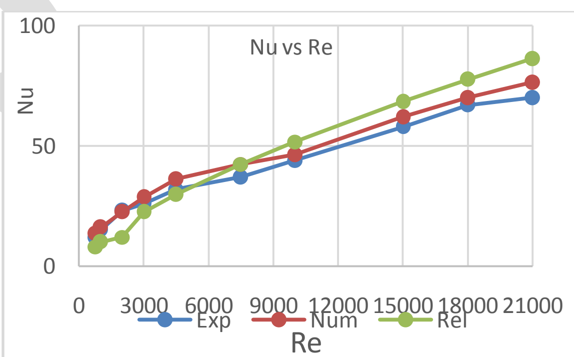

Fig: 3.1 Nusselts number Vs Reynolds number

3.2 Heat Transfer.Effect of the VGs of TWPH & CTWPH types on the heat transfer rate is presented in Figure- 3.3.The results for the dimensionless number (j/j_0) of channel with TWPH & CTWPH Vs Reynolds number is shown.All the Reynolds numbers used due to the induction of high reverse flows and disruption of boundary layers. We clearly seen that as the Reynolds number goes on increasing, the heat transfer coefficient also goes on increasing or Colburn factor. The channel with TWPH & CTWPH increase the heat transfer rate by average 53% & 50% respectively than the smooth channel.

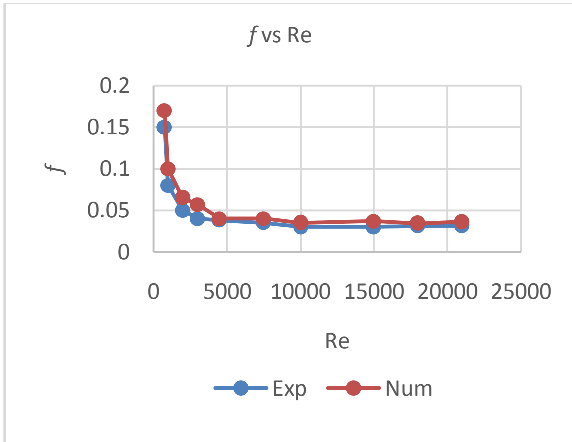

Fig: 3.2 Friction factor Vs Reynolds number

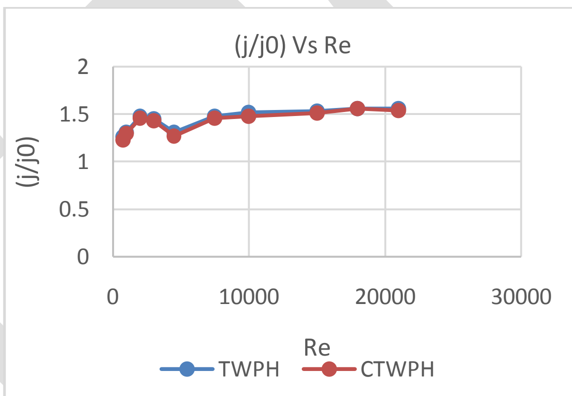

Fig: 3.3 (j/j_0)Vs Reynolds number

3.3 Friction Factor. The variation of the pressure drop is presented (eq.4) in terms of friction as Figure 3.4.It showsthe friction factor Vs the Reynolds number, for TWPH& CTWPHin rectangular channel. It is seen that frictionfactor decreases with an increases in Reynolds number.It was found that the pressure drop for the TWPH & CTWPH was average 51% & 45% respectively.The curved shape winglet pair have lower friction in comparison with the plain type winglet pair. The punched holes helps in reducing the flow resistance.

3.4 Thermal Performance Factor. From Figure 3.5, it hasbeen observed that the thermal performance factor is high for CTWPH in comparison with TWPH. It was also observed that the thermal enhancement factor is increase as the Reynolds number increases. The maximum enhancement factor was observed at Reynolds number upper limit consideration I,e Re = 21000for the present study.

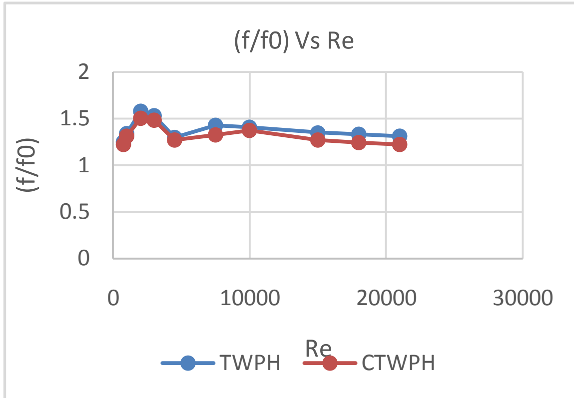

Fig: 3.4 *(f/f_0)* Vs Reynolds number

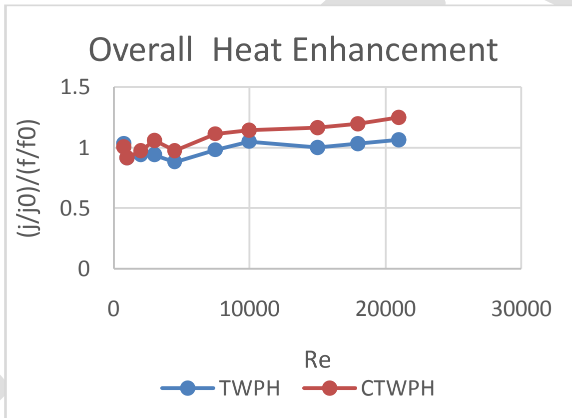

Fig: 3.5 *(j/j_0)/(f/f_0)* Vs Reynolds number

The figure 4.1 to 4.3 shows the temperature distribution for TWPH at different flow zones I,e laminar, transient and turbulent zone and figure 4.4 to 4.6 shows the pressure distribution for the same. The figure 4.7 to 4.9 shows the temperature distribution for CTWPH at different flow zones and figure 4.10 to 4.12 shows the pressure distribution for CTWPH.

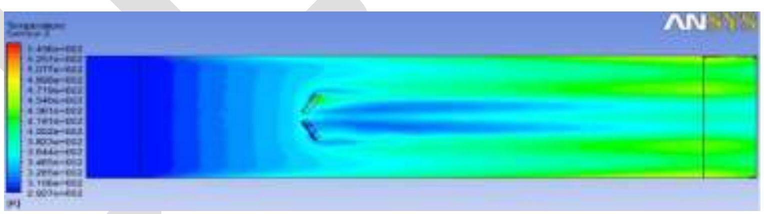

Fig: 4.1 Temperature distribution at velocity 0.165 m/s

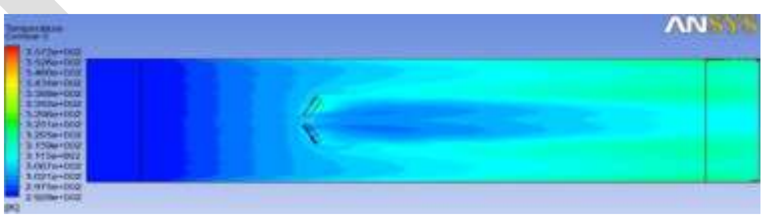

Fig: 4.2 Temperature distribution at velocity 0.662 m/s

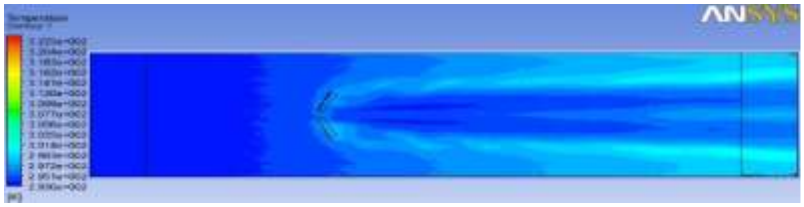

Fig: 4.3 Temperature distribution at velocity 3.31 m/s

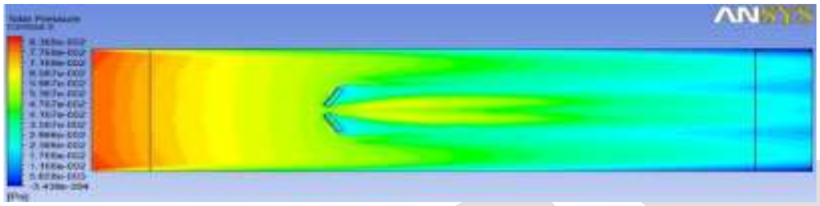

Fig: 4.4 Pressure distribution at velocity 0.165 m/s

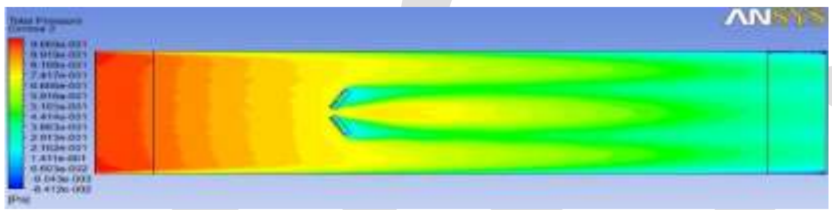

Fig: 4.5 Pressure distribution at velocity 0.662 m/s

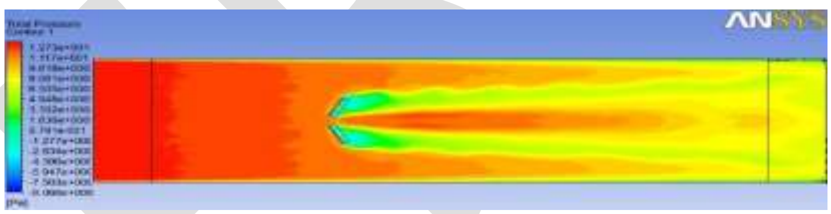

Fig: 4.6 Pressure distribution at velocity 3.31 m/s

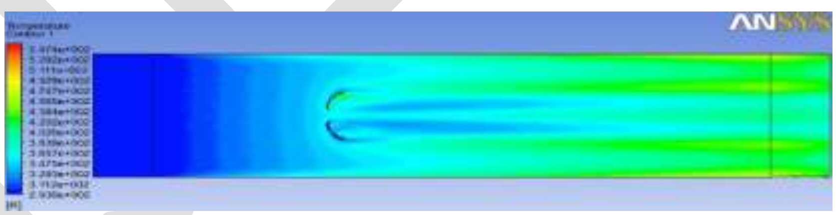

Fig: 4.7 Temperature distribution at velocity 0.165 m/s

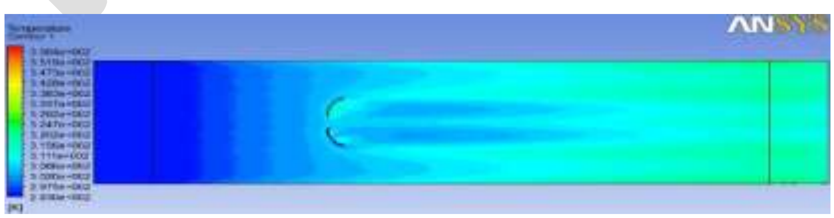

Fig: 4.8 Temperature distribution at velocity 0.662 m/s

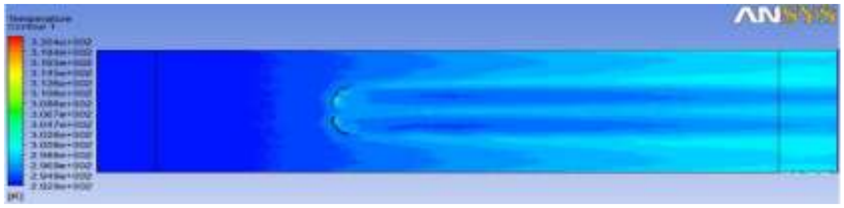

Fig: 4.9 Temperature distribution at velocity 3.31 m/s

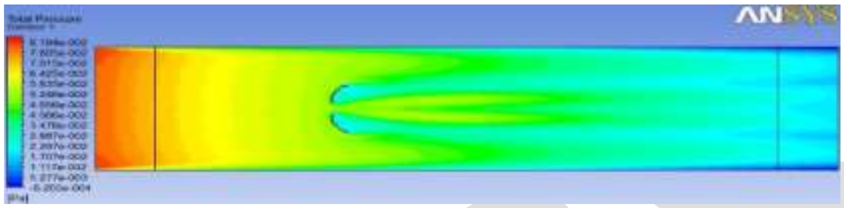

Fig: 4.10 Pressure distribution at velocity 0.165 m/s

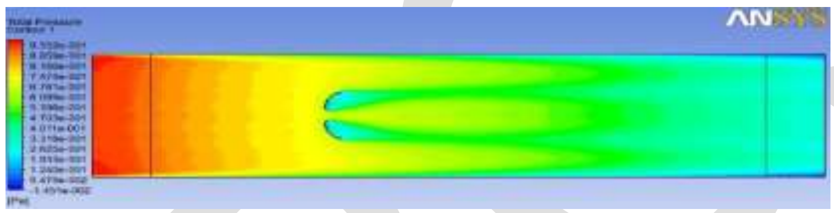

Fig: 4.11 Pressure distribution at velocity 0.332 m/s

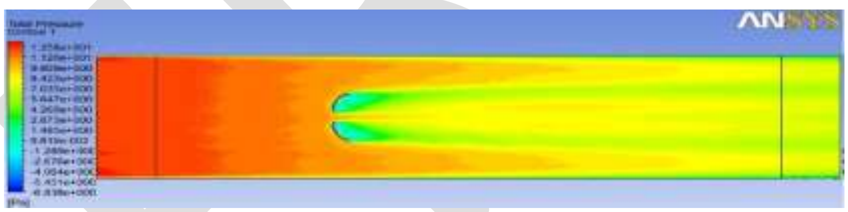

Fig: 4.12 Pressure distribution at velocity 3.31 m/s

4. CONCLUSION

When the winglet type of VGs (TWPH & CTWPH) are fitted in a rectangular channel, the effect on the heat transfer (Nu) or Colburn factor (j), friction factor (f) and thermal performance factor (η) have been investigated numerically by using ANSYS-14 software. The following conclusions are below:

1. We clearly seen that as the Reynolds number goes onincreasing, the heat transfer coefficient also goes on increasing .The TWPH in rectangular channel increase heat transfer by average 53% and CTWPH is by average 50% more than smooth channel.

2. Pressure drop for the TWPH configuration are 51% more than smooth channel and for the CTWPH configuration is 45% more than the smooth channel.

3. The punched holes in the winglet pairs really improve the heat transfer by reducing the flow resistance.

4. It has been observed that the thermal enhancement factor tends to decreases at low values of Reynolds number and it increases at high values of Reynolds number.

5. Overall it is concluded that the use of winglet type VGs enhance the heat transfer and the curved type of winglet pairs with punched holes are more effective in heat enhancement than plain winglet type VGs with or without holes.

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