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Experimental Investigation of Heat Transfer Enhancement by Winglet Type Vortex Generator in Compact Plain Fin Tube Heat Exchanger

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ABSTRACT

a Heat exchanger performance for reasons of compactness, economy in manufacturing and operating costs, energy conservation and even for ecological reasons. The importance of these issues continues to motivate the study of enhancement techniques. The impact of vortex-generation (VG) technique for air side heat transfer improvement is experimentally investigated on plain fin round tube heat exchanger. A new 2VG array deployed in a heat exchanger at the leading edge and at the middle of the fins . "delta winglet" is proposed in the present work aiming to create a constructive interface between vortices. The array is composed of 2-VG in a line array placed at the entrance at an angle β=30° with the horizontal. In this method, protuberances such as delta-wings are used to generate stream wise vortices that are carried through the heat exchanger by the main flow and induce bulk fluid mixing and a reduced thickness of the thermal boundary layer. This enhancement method has the important advantage of low cost and ease of implementation, with a usually modest pressure drop penalty.

In this study, three different arrangements of heat exchanger are tested. In first case the heat exchanger without vortex generator. In second case the heat exchanger with delta winglet type vortex generator and third case with punched hole delta winglet type vortex generator. The heat transfer coefficient (h), Reynolds number has been discussed. The results, discussion about the heat transfer coefficient shows increment in case of delta winglet type vortex generator as compared to heat transfer coefficient without delta winglet type vortex generator. Whereas heat transfer coefficient (h) for heat exchange with delta winglet type vortex generator to heat exchanger without delta winglet type vortex generator are 25-30% & 10-15% for with VG, results for Q_{avg} are for punched hole VG 10-15% and for with VG 8-12% . In summary, this study strongly suggests heat transfer coefficient with delta winglet type vortex generator applied to the fin and tube heat exchanger is found to be more as compared to the heat exchanger without delta winglet type vortex generator.

Keywords— Heat Transfer Enhancement, Fin Tube Heat Exchanger, Winglet-Type Vortex Generation, Winglet array.

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I. INTRODUCTION

Fin and tube heat exchangers are used in automobile, air conditioning, power system, chemical engineering, electronic chip cooling and aerospace, etc. Air-side convective resistance to heat transfer is dominant, at 75% or more of the total thermal resistance. Enhancement of air-side heat transfer using passive vortex generators is a promising technique in a range of applications. Increasing demands are being placed on a heat exchanger performance for reasons of compactness, economy in manufacturing and operating costs, energy conservation and even for ecological

reasons. The importance of these issues continues to motivate the study of enhancement techniques.Enhancement of air-side heat transfer using passive vortex generators is a promising technique in a range of applications. In this method, protuberances such as delta-wings are used to generate stream wise vortices that are carried through the heat exchanger by the main flow and induce bulk fluid mixing and a reduced thickness of the thermal boundary layer. This enhancement method has the important advantage of low cost and ease of implementation, with a usually modest pressure drop penalty. However, strategies for vortex-generator design and appropriate placement have not been described in the open literature for highly compact heat exchangers which are essential to fully exploit the method. Mao Yu Wen et al. [1] In this study, three different fins (plate fin, wavy fin, and compounded fin) were investigated in a wind tunnel. The heat transfer coefficient and the pressure drop of the air side, against air velocity (1–3 m/s) and Reynolds number (600–2000) have been discussed. The results of the wavy fin to the flat fin show that the heat transfer coefficient increase about 11.8–24.0%. In addition, the results of the compounded fin compared to the flat fin show that the heat transfer coefficient increase about 27.0–45.5%. Y. L. He et al. [2] The arrays are composed of two delta winglet pairs with two layout modes of continuous and discontinuous winglets. A Reynolds number ranging from 600 to 2600 was used for experimentation. Arrays with discontinuous winglet shows 33.8 to 70.6% enhancement in heat transfer coefficient accompanied by a pressure drop penalty of 43.4 to 97.2%. Whereas in case of front continuous winglet arrays heat transfer improvement of 36.7 to 81.2% and a pressure drop penalty of 60.7 to 135.6%. In case of the winglet type of vortex generator its orientation also affects the heat transfer coefficient. Winglet type vortex generator (VG) arrays for air-side heat transfer enhancement is experimentally evaluated by **A.** Joardar et al. [3] In a full-scale wind-tunnel testing of a compact plain-fin-and-tube heat exchanger. Reynolds number range based on hydraulic diameter is from 220 to 960. The enhancement in heat transfer coefficient increases with Reynolds number from 29.9% to 68.8% with a pressure drop penalty from 26% at Re 960 to 87.5% at Re 220.Jiong le et al. **[**4] Performed simulation on heat transfer and flow characteristics of a slit fin and tube heat exchanger with a longitudinal vortex generator. A Reynolds number ranging from 250 to 2500 was used for experimentation. The general categorization of the fin and tube heat exchanger is the staggered tube arrangement and inline tube arrangement. The vortex generators can also effect, according to the orientation of the winglets as staggered and online.Y. Chen et al. [5] In his study investigated Punched longitudinal vortex generators in the form of winglets in staggered arrangements to enhance heat transfers in high performance finned oval tube heat exchanger elements. For Re = 300 and I = 500, the ratios of heat transfer enhancement to flow loss penalty (j/j0) /(f/f0) were 1.151 and 1.097 for a finned oval tube with two and four staggered winglets, respectively.K. Torri et al. [6] In this paper proposes a novel technique that can augment heat transfer, but nevertheless can reduce pressure-loss in a fin-tube heat exchanger with circular tubes in a relatively low Reynolds number flow, by deploying delta winglet-type vortex generators. In case of staggered tube banks, the heat transfer was augmented by 30% to 10%, and yet the pressure loss was reduced by 55% to 34% for the Reynolds number (based on two times channel height) rang is 350 to 2100, when the present winglets are added. In this case of in-line tube banks, these were found to be 20% to 10% augmentation, and 15% to 8% reduction, respectively. Even though the above design modification shows the heat transfer enhancement the exact reason for this enhancement of the behavior of the fluid is not being briefly studied. Some papers try to learn and observe the fluid flow pattern. Jin-Sheng Leu et al. [7] Carried out experiments by an infrared thermo vision and a water tunnel systems respectively, to observe the temperature distribution and local flow structure. Numerical and experimental analyses were carried out to study the heat transfer and flow in the plate-fin and tube heat exchangers with inclined block shape vortex generators mounted behind the tubes. One more such effort for Flow visualization is carried by Chi-Chuan Wang et al. [8] Via dye-injection technique. Their Study presents flow visualization and frictional results of enlarged fin and tube heat exchangers with and without the presence of vortex generators. Many modifications are seen in the fin of the fin and tube heat exchanger, but research is also being done on the enhancement of heat transfer by modifying tube surface. J.M. Wua et al. [9] In his paper to achieve heat transfer enhancement and lower pressure loss penalty, even pressure loss reduction, presented the technique of two novel fin-tube surfaces with two rows of tubes in different diameters. The fin-tube surface with first row tubes in smaller size and second row tubes in larger size can lead to an increase of heat transfer and decrease of pressure drop in comparison with the traditional fin-tube surface with two rows of tubes in the same size. C.B. Allison et al. [10] Experimentally analyze the effect of delta-winglet vortex generators on the performance of a fin and tube radiator is presented. It was found that the winglet surface had 87% of the heat transfer capacity, but only 53% of the pressure drop of the louver fin surface.

II. EXPERIMENTAL PROCEDURE

Heat exchanger used for experimentation has following dimensional details. It has 18 copper tubes with inline arrangement & 32 fins made of aluminium.

1. Height of Heat exchanger (He) = 242mm
2. Width of Heat exchanger (We) = 210mm
3. Copper tubes inner diameter (Di) = 8.1mm
4. Copper tubes outer diameter (Do) = 9.65mm
5. Aluminium fin length (Lf) = (He) = 240mm
6. Aluminium fin width (Wf) = 65mm
7. Aluminium fin thickness (Tf) = 0.25mm
8. Pitch of fin=6.2 mm

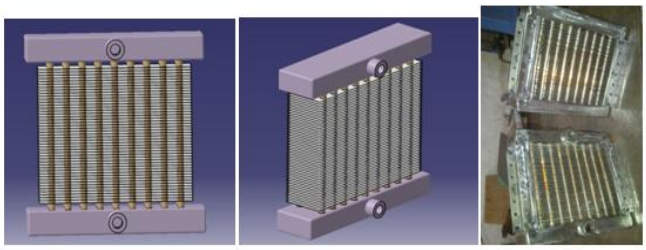
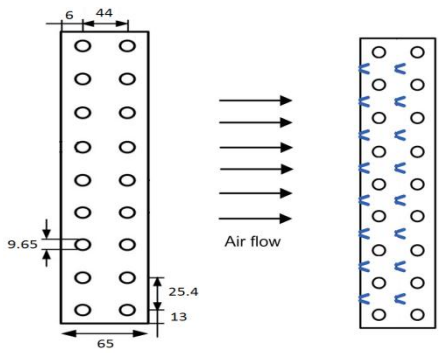


Fig 1 heat exchanger without Vortex Generator (2-Qty)



(a) (b)

Fig.2 (a)Schematic of heat-exchanger fin configuration;all dimensions in mm (b) 2VG array

The tube fin heat exchanger's fin is shown in the fig:1.In this heat exchanger it has a collared fin & tube configuration withan inline tube arrangement.The fin spacing & thickness 6.2mm & 0.25 respectively. With reference to fig:2 the proposed VG array is composed of two winglet pairs arrangement in two rows out of which one is placed at the leading edge and second at the centre of the fin (at 31.2mm) from leading edge.

The proposed VG array is composed of two delta winglet pairs placed at the leading edge of the fins with an attack angle 30° .The proposed VG's are of two types

i) Delta Winglet Vortex Generator
ii) Delta Winglet With Punched hole

For each winglet height and base are 5mm & 9mm with these combination heat transfer enhancement is compared with the plain tube fin type heat exchanger without VG. The total no of VG pairs are 16 per fin at attack angle β=30° where as single pair shown in the in the figure referred to as one VG were deployed in the heat exchanger resulting in less than 2.5% increase in the area .

In open loop wind tunnel air flow was driven by blower ad passed through oven where air is heated by the burner in oven then it passes through square duct of size 200mm x 200mm.A water was supplied by a pump to cool the heat exchanger during experiment. K –type thermocouple with (±1°C) used to detect the heat exchanger surface temperature at inlet and outlet respectively. Also for K-type thermocouples with (±1°C) were positioned at the inlet and outlet of water side. Air side pressure drop across the heat exchanger was measured by the manometer.

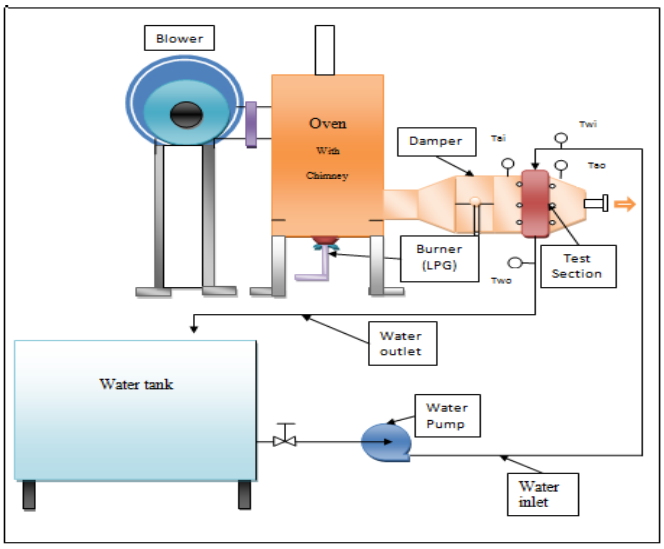
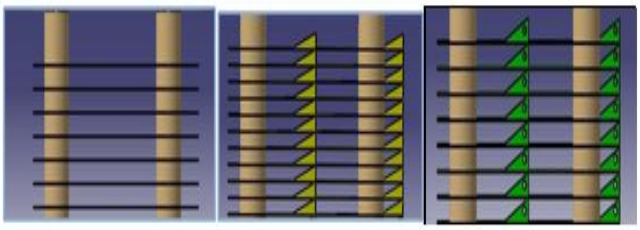


Fig.3 Block diagram of experimental set up

Experiments were conducted with the heat exchanger

i) Without Vortex Generator
ii) With Vortex Generator
iii) With Punched Hole Vortex Generator



(a) Without VG (b) With VG (c) With PHVG

Fig 4 Test section side views for three cases.

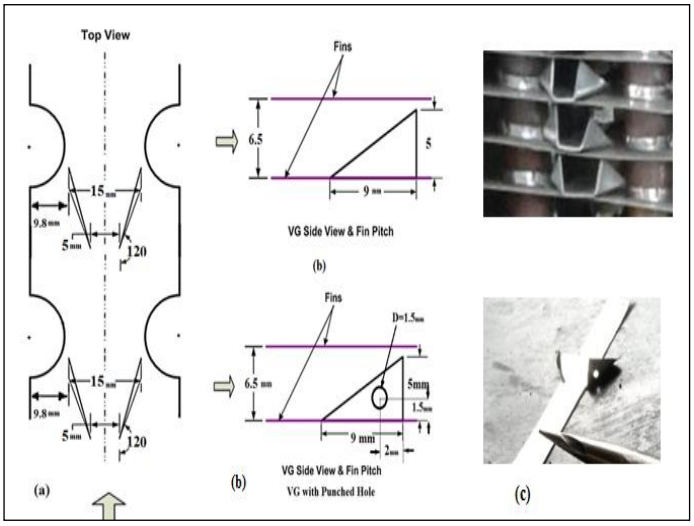


Fig.5 Geometrical details of VG

Methodology of Experiment

1. Fill the tank with water and start ignition of burner and wait until the temperature of air rise to a desired temperature.

2. Start the blower few minutes before to get the stable flow of the fluid.
3. Note down the temperature of inlet and outlet of the water.
4. Also note down the temperature of tube walls of heat exchanger on air side at various uniform location.
5. Note down the temperature of inlet and outlet of the air.
6. Note down the water column height difference of U tube manometer across the orifice meter.
7. Note down the water column height difference of U tube manometer across the heat exchanger or test section.

Start heating to the air is started by turning burner flame on in the gas oven after getting desired temperature we start water pump to circulate water from water tank in closed system then waiting for steady state (~15 min), a steady state was considered when all temperature varied by less than at least 3min.Ones an experiment reached the steady state and readings of thermocouple as well as all conditions remained constant then data is noted down for each experiment

The temperature and flow rate data sampled during experiment were first used to determine the airside heat-transfer coefficient. For the hot air and water flows, the heat-transfer rates at each side were calculated based on mass flow rate and temperature change

A. From energy balance equation

$$Q_a = \dot{m} \times C_{p,a} \times (T_{a,i} - T_{a,o}) \quad (1)$$

$$Q_w = \dot{m} \times C_{p,w} \times (T_{w,o} - T_{w,i}) \quad (2)$$

B. Q_{ave} was estimated as the average of the air side and water side heat-transfer rates using the following:

$$Q_{ave} = \frac{Q_a + Q_w}{2} \text{ W} \quad (3)$$

C. A log-mean-temperature–difference (LMTD) method was employed to analyse the heat-transfer performance

$$UA_{tot} = \frac{Q_{ave}}{F \times LMTD} \quad (F=0.86) \quad (4)$$

In this equation, U is he overall heat transfer coefficient and A_{tot} denotes the total air-side heat transfer area (fin and tube).LMTD for counter flow was given by

$$LMTD = \frac{(T_{a,i} - T_{c,o}) - (T_{a,o} - T_{c,i})}{ln\left(\frac{T_{a,i} - T_{c,o}}{T_{a,o} - T_{c,i}}\right)} \quad (5)$$

and the correction factor F was very close to unity it is taken as 0.86 after calculation.

D. The overall thermal conductance (UA_{tot}) of the heat exchanger was then available from equation (4),and it could be related to total thermal resistance through the following expression (where the contact resistance and fouling are assumed to be negligible):

$$\frac{1}{UA_{tot}} = R_w + R_{cond} + R_a \quad (6)$$

E. Where R_w, $Rcond$, R_a are the water-side convective resistance, tube wall conductance resistance, and air side heat transfer resistance, respectively,

$$R_W = \frac{D_i}{N_{uc} K_c A_c} \quad (7)$$

$$R_{cond} = \frac{ln\left(\frac{D_o}{D_i}\right)}{2K_{Cu}A_w} \quad (8)$$

$$R_a = \frac{1}{\eta_o h_a A_{tot}} \quad (9)$$

The conduction resistance across the tube wall was calculated assuming steady,1D heat conduction as described in equation (8),In this study *Rcond* was found to be l% of the total resistance. The conduction resistance of collar is also negligible.

F. From the Dittus bolter equation we can find out Nusselt numer:

$$Nu = 0.023 \times Re^{0.8} \times Pr^{0.4} \quad (10)$$

G. The non dimensional friction factor of the heat exchanger f is expressed as (excluding the negligible entrance and exit losses)

$$f = \left[\frac{(\Delta P \times 2 \times \rho_a)}{(G^2)} \cdot \left(\frac{A_{min}}{A_{tot}}\right)\right] - \left[(1+\sigma^2) \cdot \left(\frac{\rho_{ai}}{\rho_{ao}}\right) \cdot \left(\frac{A_{min}}{A_{tot}}\right) \cdot \left(\frac{\rho_a}{\rho_{ao}}\right)\right] \quad (11)$$

$$\sigma = \frac{A_{min}}{A_{front}} \quad (11a)$$

$$\rho_a = \frac{\rho_{ai} + \rho_{ao}}{2} \quad (11b)$$

III.RESULT & DISCUSSION

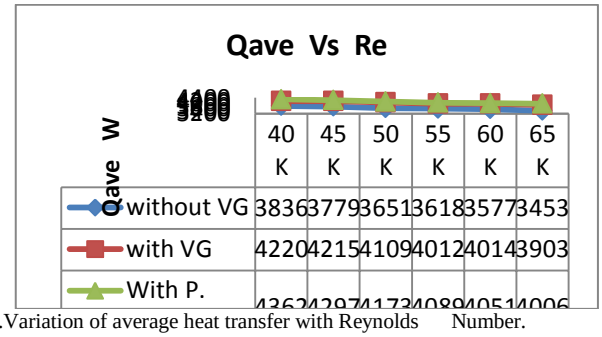


Fig.6.Variation of average heat transfer with Reynolds Number.

Fig.5 Shows the variation of average heat transfer with Reynolds Number for the heat exchanger .In this graph average heat transfer through the exchanger after attaching vortex generator is compared with the without vortex generator (VG).it was found that the average heat transfer Q_{avg} for punched hole vg 10-15% and for with vg 8-12% when compared with without VG.

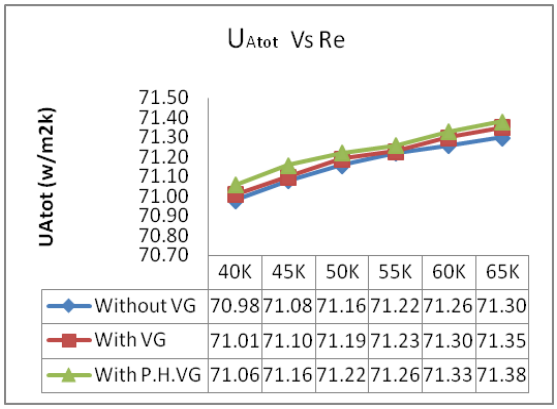


Fig.7. Variations of overall heat transfer coefficient with Reynolds Number.

Figure 4.Shows the variation of overall heat transfer with Reynolds Number for the heat exchanger .In this graph overall heat transfer through the exchanger after attaching Vortex Generator is compared with the without Vortex Generator (VG). Overall heat transfer coefficient when compared with without VG results shows 0.10-0.15% enhancement for punched VG and 0.5-0.6% with VG arrangement. It was found that the overall heat transfer with punched hole VG is higher, without VG and with VG has mean overall heat transfer coefficient at 60000 below this UA_{tot} is decreasing and after this UA_{tot} is increasing.

As shown in Fig.5 the maximum enhancement can be seen to occur in the low RE region. Therefore, the heat transfer improvement for VG fins can be mainly due to longitudinal vortices. Through enhancing bulk mixing, modifying the boundary layer, and potentially causing flow destabilization, the vortices results in net effect of decreased air-side thermal resistance (i.e. increased convective heat transfer coefficient).

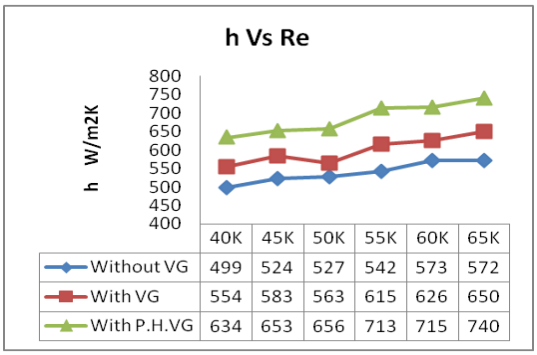


Fig.8.Air side heat transfer coefficient with Reynolds Number.

heat transfer coefficient (h) for heat exchange with delta winglet type vortex generator to heat exchanger without delta winglet type vortex generator are 25-30% & 10-15%.

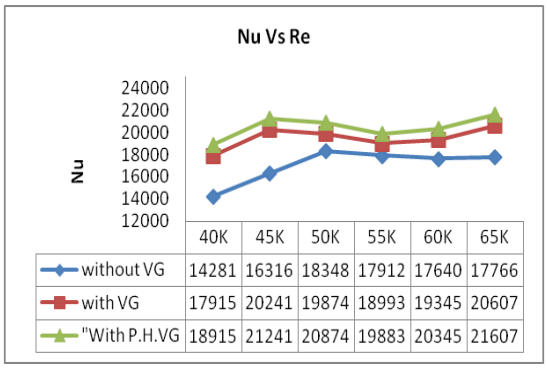


Fig.9. Nusselt Number with Reynolds Number.

Nusselt Number for heat exchange with delta winglet type vortex generator to heat exchanger without delta winglet type vortex generator are 25-30% & 5-10%.

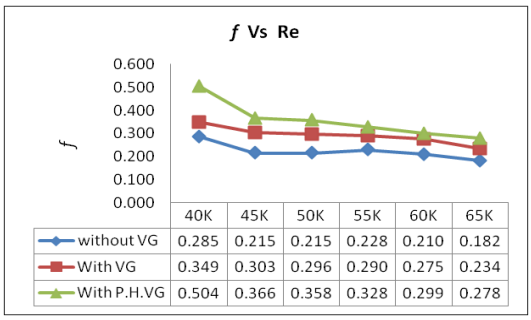


Fig.10. Friction Factor with Reynolds Number.

Friction Factor for heat exchange with delta winglet type vortex generator to heat exchanger without delta winglet type vortex generator are 40-45% & 15-20%.

IV.CONCLUSION

The thermal and hydraulic performance of a prototype plain-fin round tube heat exchanger was experimentally evaluated before and after implemention of two types of VGs. Deployed at an angle 30°.these VGs are of shape one with delta winglet type and other punched hole delta winglet type.the experimental results shows air side heat transfer resistance is higher than other two cases and hence heat transfer improvement is achived.the VG array is found more effective at comparatively low Reynolds Number which is representative of many HVAC&R applications and heat-exchanger design.The two row winglet configuration was employed for test specimen.Based on the heat transfer and perssure drop results,it is conjectured that the vortex produced by the leading winglet reduces the wake zone of the trailing winglet and mitigates the pressure differance between its fore and after sides.The overall performance of the heat exchanger is appreciably enhanced.Finally it may be anticipated that in the practical design the winglets will likely be punched out of the fins.

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