EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER AND PRESSURE DROP CHARACTERISTIC OF A NEW WINGLET TYPE VORTEX GENERATORS IN RECTANGULAR CHANNEL

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Abstract
In this study, the aim is to investigate heat transfer rate by using the punched trapezoidal longitudinal winglet type vortex generators. For this purpose, a punched trapezoidal vortex generators is developed. The trapezoidal vortex generators are directly punched on the longitudinal winglet at angles of 450. The various test parameters are, the position of the punched vortex generator on the longitudinal winglet, the geometry of vortex generators, and the angular position of punched vortex generators. The Reynolds numbers considered for the channel flow case (based on the hydraulic diameter) ranged from 7411 to 18645.The experiment is performed for trapezoidal longitudinal winglet type vortex generator with a hole 3.5mm at centroid and without hole at 450 angle of attack. The results of punched trapezoidal longitudinal vortex generators with and without hole were compared with the smooth plate surface. The vortex generators with hole at centroid show a more significant increase in heat transfer coefficient as compared to the vortex generators without hole for channel flows and flat plate. Results showed a 27–69% increase in heat transfer due to the use of vortex generators.

Key words: Heat transfer enhancement, punched vortex generators, pressure drop

I. INTRODUCTION

Increasing demands are placed on the performance of air fin tube heat exchanger used in air conditioning, power system, electronic chip cooling and aerospace, etc. for reasons of compactness, economy in manufacturing and operating costs, and energy conservation. A variety of techniques of enhancing air side heat transfer such as wavy fin, louver fin and slotted fin, etc. usually lead to a larger pressure loss penalty while enhancing heat transfer. One of the best techniques is to use vortex generators to produce longitudinal vortices inducing strong swirling motion that serves to bring about the enhancement of heat transfer at a modest cost of the additional pressure loss. The mechanism of heat transfer enhancement is based on flow separation and reattachment. In general, flow reattachment Introduces a strong shear flow on the surface behind each rib or winglet, resulting in an effective disruption of the thermal boundary layer and thus the improvement of the heat transfer.

Kai-ShingYang et al.[1] suggests that the asymmetric combination such as using loose vortex generator can be quite effective. The triangular attack vortex generator is regarded as the optimum enhancement design for it could reduce 12–15% surface area at a frontal velocity around 3–5 m/s. Wisam Abed Kattea et al. [2] was noticed that heat transfer is enhanced (36-56)% when Circular shapes of vortex generators are used and (39-51) % when Square shapes of vortex generators are used. Biswas et al. [3] investigate the effect of winglet thickness on convective heat transfer. As thickness of the winglets increases there is increase in the overall heat transfer of channel is 0.83%, 4.34%, 7.62%

and 12.49% for W/H 0.0622, 0.1244, 0.1866 and 0.2485 respectively. It is also find that as compared to a channel without winglets, the heat transfer is enhanced by 33% when single winglet is used and by 67% when a winglet pair is employed. W.Q.Tao et al. [4] Reported that the average Nusselt number on the surfaces of plate increases with the increase of the attack angle of delta winglet pair compared with that of plain plate. Jiangfeng et al. [5] experimentally investigated that the heat transfer characteristics of modified rectangular longitudinal vortex generator (LVG) obtained by cutting off the four corners of a rectangular wing is presented and compared with those of original rectangular LVG. Modified rectangular wing pairs (MRWPs) have better flow and heat transfer characteristics than those of rectangular wing pairs.QiulingYe et al. [6] were investigate the thermal characteristics of curved vortex generators as compared with traditional vortex generators rectangular winglet, trapezoidal winglet and delta winglet. Curved vortex generators have better flow and heat transfer characteristics than those of traditional vortex generators. Guobing Zhou, et al. [7] investigate the performance of plane and curved winglet (rectangular trapezoidal and delta) vortex generators (VGs) with and without punched holes. The vortex generators with punched holes have better flow and heat transfer characteristics but the optimal diameter of the holes needs to be matched with the VG face area.S.Caliskan et al.[8] investigate the effects of the attack angle and distance of both PTVGs and PRVGs from the channel bottom on the heat transfer and pressure drop characteristics. The PTVGs with less distance from the channel bottom have better convective heat transfer performance of PTVGs than those of PRVGs. Tigglebeck et al.[9] found that in a rectangular channel flow a pair of delta winglets performs slightly better heat transfer than a pair of rectangular winglets at higher attack angles and Reynolds Number. Tian et al. [10] numerically investigated the effects of RWP and DWP with two different configurations, such as common-flow-down and common-flow-up heat transfers along with fluid flow characteristics numbers.

Nomenclature

A	convection heat transfer area of channel (m^2)	K	thermal conductivity of air (W/m K)
AR	Aspect ratio of channel	L	length of test channel (m)
b	distance of punched winglet from the channel bottom (m)	g	spacing between punched winglet (m)
c	punched winglet length (m)	H	channel height (m)
D_h	hydraulic diameter (m)	Nu	Nusselt number
e	height of winglet (m)		**Greek Symbols**
f	friction factor	α	attack angle of punched winglet
S	spacing between longitudinal winglet (m)	ρ	density of the fluid (kg/m^{3})
ΔP	pressure drop (Pa)	η	thermal enhancement factor
Pr	Prandtl number	μ	Kinematic viscosity (m^2/s)
PT VGs	punched trapezoidal vortex generators		
Re	Reynolds number		**Subscripts**
Q	heat transfer (W)	a	Augmented
T	temperature (K)	avg	Average

t	thickness of longitudinal winglet (m)	b	Bulk
U	mean velocity (m/s)	0	channel without vortex generator
V	voltage (V)	pp	pumping power
W	width of channel (m)	s	Surface
h	heat transfer coefficient (W/m^2 K)	o	Outlet
I	current (A)	i	Inlet

II.EXPERIMENTAL SETUP

The experimental setup is designed and constructed at PES MCOE, Pune. The heat transfer experiments were conducted in a rectangular channel as shown in Fig. 1. The experimental setup consisted of a test section, a centrifugal blower, vortex generators, and devices for measuring flow velocity, temperature and pressure difference. Air was drawn in by a variable speed and passed through the rectangular duct as test section of the channel. The duct inner cross section dimensions were 150 mm (wide) and 100mm (height) and 2100mm long. The channel was constructed with 5 mm thick acrylic sheet. A square slot is cut at the bottom of the duct. The slot is made to insert the plates and heater assembly in the duct. Area of the slot provided is 150mm x 150mm. A MS sheet is used to make a nozzle which provides passage of air from the blower to the duct. The length of the nozzle is 300mm. The nozzle has a rectangular cross section 150mm x 100mm at the duct side and circular cross section whose diameter is 40mm at the blower side. The plate heater is made of nichrome coil as heating element of dimension 150mm x 150mm is used to heat the aluminium plate at the desired temperatures. A heating plate of dimension 150mm x 150mm was made of aluminium of thickness 10mm and creates a slot on it of 20mm spacing. In order to confirm the uniform temperatures at the bottom and upper side of the 10mm thick aluminium plate, each surface was equipped with K-type thermocouples, All thermocouples were separately calibrated. The trapezoidal longitudinal winglets were mounted on top of aluminium plate to enhance the convective heat transfer. The averaged heat transfer coefficient on the plate surface was measured for various rates of airflow through the duct.

The longitudinal winglets were made of high conductivity aluminium material. Each of the longitudinal winglets was fabricated from 1mm thick aluminium plates, and 150mm long (L). The longitudinal winglets were fit into the slot which created on the aluminium plates. The thermal contact resistance due to the slot created on the heating plate introduced a minor conservative preference to the reported results. The minimum distance between the two longitudinal winglets was 20mm (S).The punched length of the trapezoidal winglet was 13.5mm (c).while the distance between the two punched winglets was 20mm (g).The air velocity was measured by the Kimo LV107-type anemometer connected to the output of the blower and a pressure transducer was used to determine the pressure drop between the air inlet and outlet at the test section. The inlet and outlet temperatures of the channel air were measured in different locations of the channel by using a K-type thermocouple. All of these thermocouples were connected to a digital control system. To avoid the heat loss from bottom side of heater the plate is covered by the asbestos sheet of size 150mm x 150mm x 5mm.

III.EXPERIMENTAL PROCEDURE

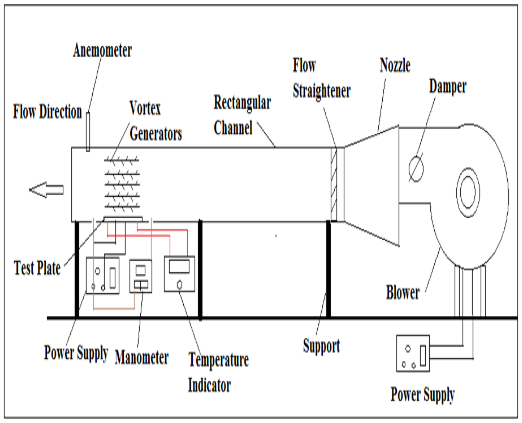

Fig.1 Experimental setup block diagram

Fig.2 Actual test setup image.

Fig.3 A channel with trapezoidal vortex generators

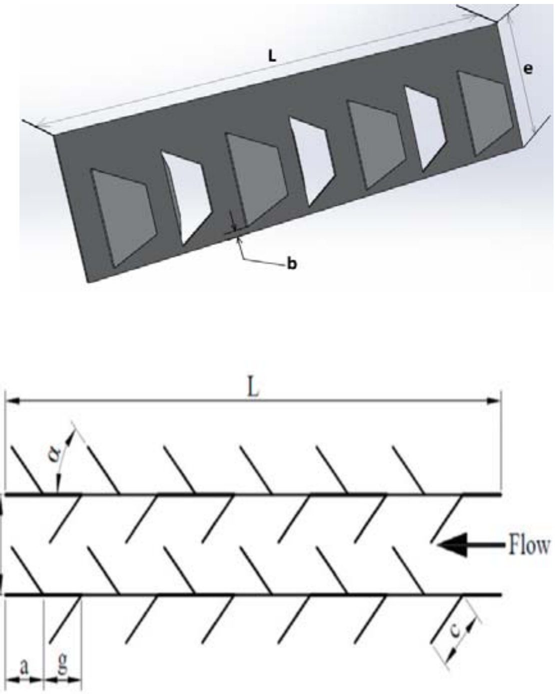

Fig.4 Schematic view of vortex generators

A constant power input was supplied and a constant velocity was adjusted. A temperature of air at inlet and exit of test section and at different locations on a test plate were recorded with temperature indicator. An initial period of 25-35 minutes required to reach a steady state condition (which was considered to be attained, when the temperatures indicated by the thermocouples did not vary with more than $\pm 1^0$C within a period of about 3-4 Mins.) After collecting a set of data at a steady state conditions, a new set of data was collected with a same velocity for different arrangements. The velocity of air then increased to get a new set of data for corresponding value of Re number. A new set of data was collected for the varying heat flux condition when a steady state conditions were reached again.

IV.FORMULAE USED FOR CALCULATION

The convection heat transfer can be written as:

$$Q_{conv} = h \times A \times (T_S - T_b)$$

In which

$$T_b = (T_o + T_i)/2$$

And

$$T_s = \sum (T_{s1} + T_{s2} + T_{s3} \cdots + T_{s14})/14$$

Where T_{s1-14} the local surface temperatures and The average surface temperature is calculated from 14 stations, The Nusselt number (Nu) are estimated as follows:

$$Nu = (h \times D_h)/k$$

The Reynolds number based on the channel hydraulic diameter was given by

$$Re = (\rho \times u \times D_h)/\mu$$

Where $D_h = 2WH/(W+H)$ was the channel hydraulic diameter. Friction factor, f can be written as

$$f = (\Delta P) \times \frac{D_h}{L} \times (\frac{2}{\rho \mu^2})$$

Where ΔP was pressure drop across the length of the channel, L.

V. VALIDATION OF EXPERIMENTAL SET UP

The experimental data for the forced convection heat transfer and friction factor in a rectangular duct with punched trapezoidal vortex generators (PTVGs) was examined under a turbulent flow regime. The basic purpose of validation is to serve as a basis of comparison of experimental result and theoretical result of the flat plate. The present experimental results of a flat plate were first validated in terms of the Nusselt number and the friction factor. The Nusselt number and the friction factor obtained from the present Flat plate were compared with the correlations of Dittus-Boelter and Blasius found in the open literature

for turbulent flow in ducts.

Correlation of Dittus-Boelter,

$Nu = 0.023\, Re^{0.8}\, Pr^{0.4}$ For heating

Correlation of Blasius,

$f = 0.316 Re^{-0.25}$ For $3000 \leq Re \geq 20000$

Fig.5a and b for 10w, Fig 6a and b for 20w shows a comparison of the Nusselt number and the friction factor obtained from the above correlations. In the figures, the present results agree within the ±11% deviation for both the friction factor and Nusselt number correlations.

The thermal enhancement factor, η, was defined as the ratio of the heat transfer coefficient of an augmented surface, h, to that of a smooth surface, h_0, at a constant pumping power:

$$\eta = h/h_0 = Nu/Nu_0$$

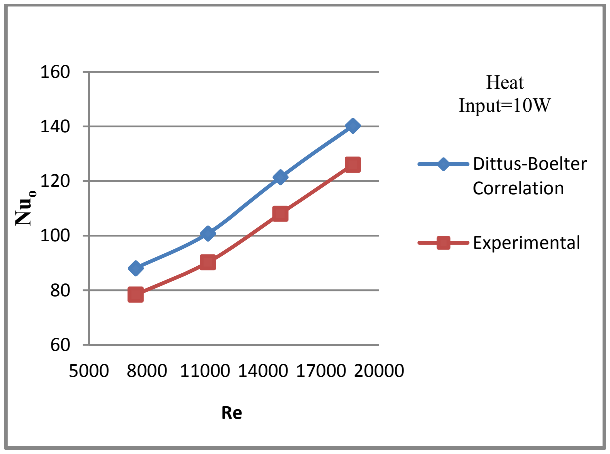

Fig.5a Verification of: Nusselt number for smooth channel.

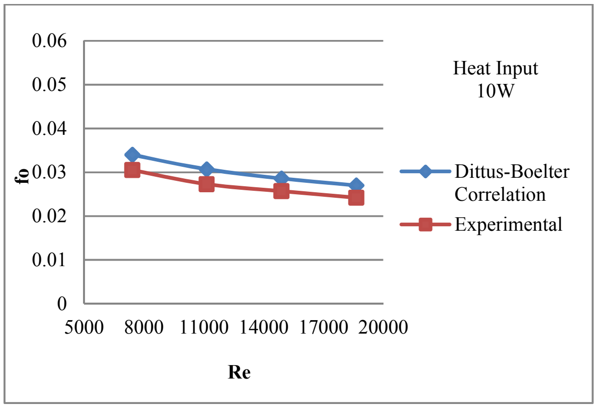

Fig.5b Verification of friction factor for smooth channel

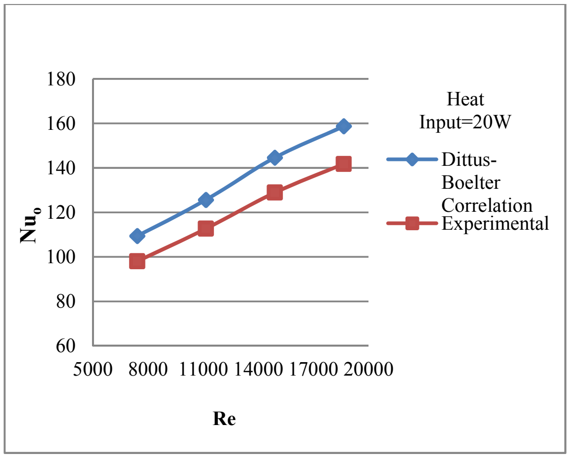

Fig.6a Verification of: Nusselt number for smooth channel.

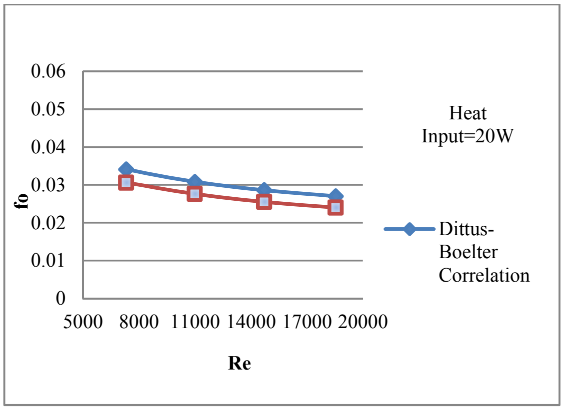

Fig.6b Verification of friction factor for smooth channel

VI. RESULT AND DISSCUSSION

The experiment is performed for trapezoidal longitudinal winglet type vortex generator with a hole 3.5 mm at centroid and without hole at 45^0 angle of attack. The results of punched trapezoidal longitudinal vortex generators with and without hole were compared with the flat plate. The experiment is conducted with different Reynolds Number and with 2 sets of power input (10W and 20W). The results on PTVGs with different parameter are presented in this paper, the Nusselt numbers obtained under turbulent flow conditions for trapezoidal-type vortex generators, with different Reynolds Number (7411 to 18645) and distances of punched winglet from the channel bottom (b/e = 0.06), and attack angle α=45°are presented in Fig.7-8

As shown in Fig.7-8, the use of vortex generators with hole lead to considerable heat transfer enhancements and lower pressure drop in comparison with the vortex generators without hole. The Nusselt number values, increase with the rise of the Reynolds number. It is also found that the Nusselt number decreases with the rise of the distance between the punched winglet and the channel bottom [8].This can be explained by strong turbulence intensity in the presence of vortex generators, leading to a rapid mixing between the core and wall flow, especially at a lower b/e ratio. As can be seen, from all the figures, the PTVGs with hole provide a higher heat transfer of coefficients than the PTVGs without hole and flat plate for all Reynolds number values. This can be attributed to the higher flow blockage, which creates a stronger reverse/recirculation flow from the triangular vortex generators, leading to better mixing between the core and the wall flows. Furthermore, a close examination reveals that for PTVGs with hole and for 20w heat input, the heat transfer augmentation with α= 45° is higher than that with 10w, The maximum Nusselt number were obtained at b/e = 0.06, α= 45°, Re = 18,645 and heat input 20w for PTVGs. The maximum difference of the Nusselt number between flat plate and PTVGs without hole, with hole is found to occur at b/e = 0.06, α = 45°, and Re = 18,645, with a value equal to 47.72% and 69.32%.

As shown in Fig.7-8,the Nusselt numbers between Flat plate, PTVGs without hole and with hole depending on b/e and for Re = 7411 and Re = 18,645 varies between 27.15-47.72% and 46.92–69.32%, respectively. From the results mentioned above, it is clear that the distances of punched winglets from the bottom of the channel, attack angle and vortex generator shapes have a significant effect on the heat transfer.

In the Fig.9 and Fig.10, the relation of Nu_a/Nu_o for 10w and 20w is shown, for the Reynolds number, the distances of punched winglet from the channel bottom (b/e = 0.06) and also constant attack angle α=45° in the PTVGs with and without hole respectively. It can be seen that Nu_a/Nu_o of the channel with PTVGs with hole and without hole, increases as the Reynolds number values increased. The shorter distance of the punched winglet from the channel bottom makes disturbance at the boundary layer more effectively and provides a better air flow mix. Hence, the heat transfer is enhanced. The heat transfer enhancement from the trapezoidal vortex generator with hole was more significant than that of the without hole vortex generator. For the PTVGs without hole, The Nusselt number ratio varied from 1.27 to 1.46 (with Re increasing from 7411 to 18645). For the PTVGs with hole, the highest Nu_a/Nu_o values from 1.47 to 1.83, (as Re is increased from 7411 to 18645) are achieved for heat input 10w and For the PTVGs without hole the Nusselt number ratio varied from 1.38 to 1.74 (with Re increasing from 7411 to 18645). For the PTVGs with hole, the highest Nu_a/Nu_o values from 1.65 to 2.01, (as Re is increased from 7411 to 18645) are achieved for heat input 20w.It is worth noting that as the Reynolds number increases from 7411 to 18645 pressure drop increases correspondingly, it is also observed in Fig.11 that PTVGs with 3.5 mm hole in centroid have low pressure drop and maximum Nusselt number as compare to PTVGs without hole but it does not much effect on pressure drop when the heat input change from 10w to 20w as shown in Fig.11-12.This indicated that the use of PTVGs with hole in centroid leads to the advantage over that of PTVGs without hole.

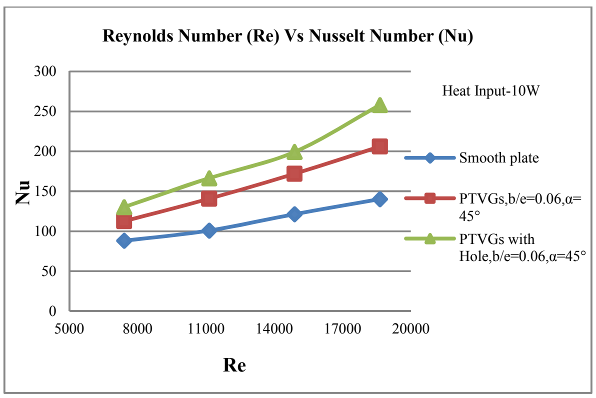

Fig.7. Nusselt Number for varying Reynolds Number and Heat input=10W

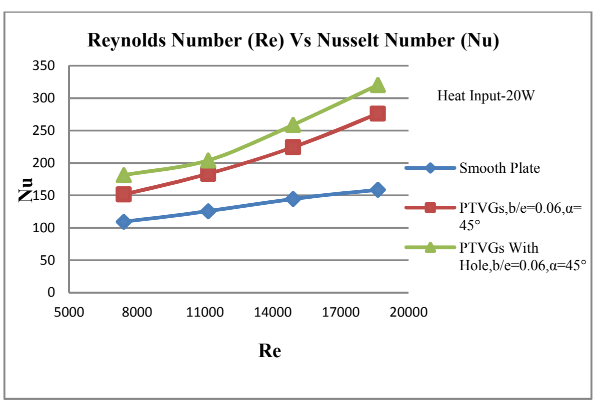

Fig. 8. Nusselt Number for varying Reynolds Number and Heat Input=20W

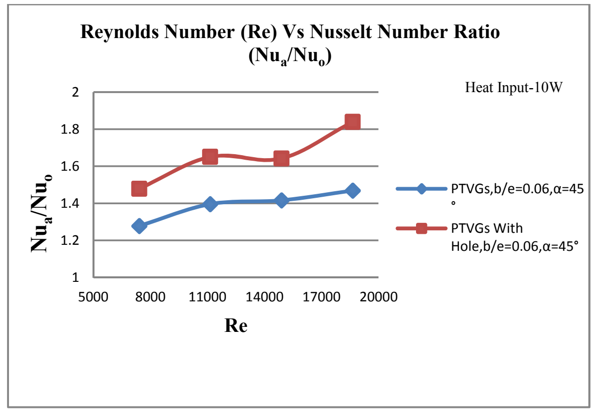

Fig. 9 Nusselt Number ratio Nu_a/Nu_o for varying Reynolds Number and Heat Input=20W

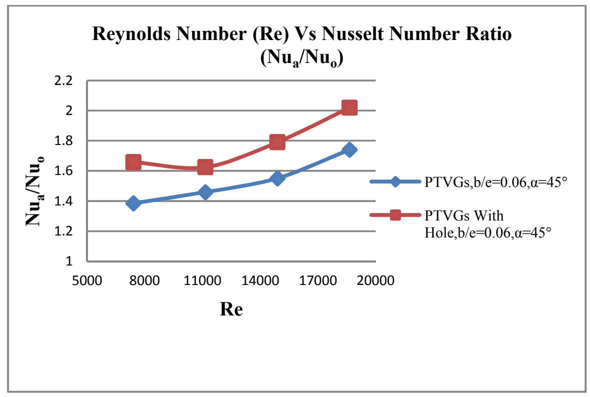

Fig. 10 Nusselt Number ratio for varying Reynolds Number and Heat Input=20W

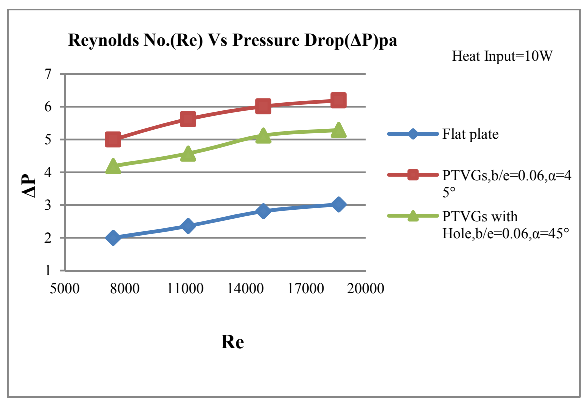

Fig.11 Pressure drop for varying Reynolds Number and Heat Input=10W

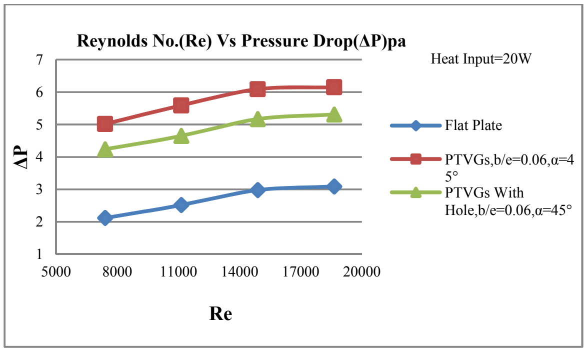

Fig.12 Pressure drop for varying Reynolds Number and Heat Input=20W

VII.CONCLUSION

An experimental investigation of heat transfer characteristics in a rectangular duct with and without hole in PTVGs under varying heat flux conditions have been performed.
The following conclusions have been drawn:

- The PTVGs with and without Hole arrangements had significantly enhanced the heat transfer rate, in comparison to a smooth plate surface. The heat transferred from surfaces with Hole in PTVGs was higher than that of the PTVGs without hole.
- The VGs which have smaller face area, smaller holes tend to give better thermal performance such as d = 3.5 mm for the varying Reynolds numbers.
- The hole in trapezoidal vortex generators decreases the flow resistance and improves the performance of VGs but the optimal diameter of the Hole Need to be matched with the VG face area.
- The PTVGs with Hole and the distance of the punched winglet from the channel bottom (b/e) of 0.06 with attack angle 45^0, provided the more effective heat transfer enhancement, as compared to the PTVGs without hole and smooth plate surface.
- The PTVGs with Hole and heat input=20w provided the more effective heat transfer enhancement, as compared to the heat input =10w.
- The PTVGs with 3.5 mm hole in centroid have low pressure drop and maximum Nusselt number as compare to PTVGs without hole but it does not much effect on pressure drop when the heat input change from 10w to 20w.

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