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Experimental Investigation of Heat Transfer Enhancement by Vortex Generators in Heat Exchanger

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ABSTRACT

Air -side thermal resistance is an important factor in designing a compact heat exchanger. In fact, the purpose of a compact heat exchanger is to present a large heat transfer surface area to compensate for the poor convection coefficient of air Vortex generation is a new and innovative strategy of enhancing air-side heat transfer. Vortex generators such as wings and winglets can introduce vortices into the flow field causing heat transfer enhancement. This study presents the experimental investigation of thermal and flow characteristics of different types of vortex generators. Four types of vortex generators are examined. The performance of a pair of new vortex generators that is curved trapezoidal winglet (CTW) is experimentally investigated and compared with traditional vortex generators rectangular winglet, trapezoidal winglet and delta winglet using dimensionless factors- j/j0, f/f0 and R = (j/j0)/(f/f0). To compare the thermal and flow performance of these VGs, the tests are conducted under the attack angle ($\beta = 45^0$), front edge pitch (S_1 = 20 mm) and placement (S_2 = 170 mm).

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I. INTRODUCTION

The development of advanced Nano fluids, which have better conduction and convection thermal properties, has presented a new opportunity to design a high energy efficient, light-weight automobile radiator. A high-efficiency compact heat exchanger is effective to achieve such goals as improving energy efficiency and reduction of CO_2 emission. The total thermal resistance for such kind of heat exchangers is comprised of three parts: the air-side convective thermal resistance, the wall conductive thermal resistance and the liquid-side (often two-phase heat transfer) convective thermal resistance. The heat transfer coefficient on the air-side is typically low due to the thermo-physical properties of air. Thus the air-side thermal resistance is the dominant part of the overall heat transfer process and efforts to improve the performance of these heat exchangers should focus on the air-side surfaces. Vortex generation is a new and innovative strategy of enhancing air-side heat transfer. Vortex generators such as wings and winglets can introduce vortices into the flow field causing heat transfer enhancement. Vortex can be divided into two categories based on the axes of these vortices: transverse vortices and longitudinal vortices.

Longitudinal vortices have their axes parallel to the main flow direction and transverse vortices have their axes perpendicular to the main flow direction. Vortex generators can be punched, mounted or embossed on a heat transfer surface. When fluid flows through vortex generators, vortices are generated due to the friction and separation on the edge of the vortex generator. Longitudinal vortex generators generate higher heat transfer enhancement for the same pressure penalty than transverse vortex generators.

In traditional point of view, there are three mechanisms for passive heat transfer enhancement, developing boundary layers, swirl and flow destabilization. Longitudinal vortex generators can generate all the three mechanisms for heat transfer enhancement. Vortex generators are a kind of passive heat transfer enhancing device which are attached to the duct walls or fin surfaces and protrude into the flow at an angle of attack to the flow direction.

Air-side thermal resistance is an important factor in designing a compact heat exchanger. In fact, the purpose of a compact heat exchanger is to present a large heat transfer surface area to compensate for the poor convection coefficient of air.

II. METHODOLOGY

A. a Vortex Generators

Four kinds of vortex generators (VGs) are selected for comparative test, which are rectangular winglet (RW), trapezoidal winglet (TW), delta winglet (DW) and curved trapezoidal winglet (CTW).

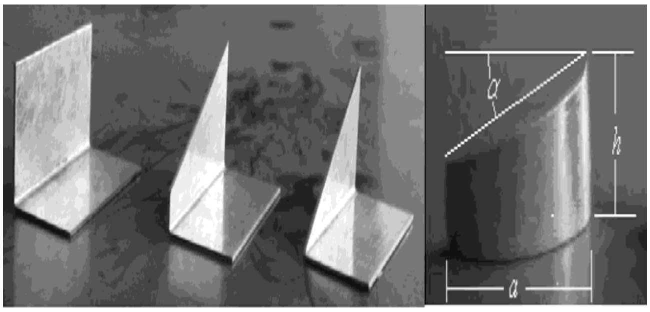

Fig.1.Vortex Generators

A pair of above vortex generators is fixed vertically on the copper plate (using silica gel) with a common flow-down configuration.

To compare the thermal and flow performance of these VGs, the tests are conducted under the attack angle ($\beta = 45^o$), front edge pitch (S_1 =20 mm) and placement (S_2 = 170 mm downstream the inlet of the test section).

Table no. 1: Geometric sizes of vortex generator pairs and test condition

Type	α (degree)	B (degree)	a (mm)	b (mm)	h (mm)	S_1 (mm)	S_2 (mm)
RWP	-	45	40	0	20	20	170
TWP	12 & 20	45	40	0	20	20	170
DWP	-	45	40	0	20	20	170
CTWP	12 & 20	0 to 90	40	10 & 20	20	10,20 &30	170

b. . Experimental setup

As shown in Fig.1, the experiment is conducted in an air channel which is heated by hot water with counter-flow design. The air channel consists of inlet contraction section, stabilization section, test section, flow metering section, blower. The hot water is from a boiler, pumped into the water channel, cooled by the air and then returns to the boiler. A sandwich configuration is used for the heat transfer section (test section) where the air and water are separated by a copper plate.

The test section and the water loop are well insulated to minimize the heat loss. The apparatus is setup in an air-conditioned room to keep the inlet air temperature at constant of 20^0C. Air is drawn in by a blower, passes through the vortex generator embedded in the test section (VG is attached on the copper plate) and then is exhausted.

For measuring the water flow rate rotameter is used. Inclined U tube manometer is used for measuring pressure difference between inlet and outlet of air. Whereas 8 number of thermocouples and temperature indicators are utilized for measuring the wall temperature of plate

.

Fig.2. Experimental Setup

c. Test Procedure

The pump, boiler and blower are switch on after filling the water loop. Zero point is verified for air flow meter, manometer and water flow rate meter before each test run. The water temperature, water flow rate and air velocity are regulated to the required values for test. The inlet and outlet temperatures of water and air are measured by temperature indicator.

Once the steady state occurs, the observed parameters such as inlet and outlet temperature of water and air, wall temperatures of copper plate, water flow rate, air velocity and pressure loss are recorded for 2 minutes of intervals. The results are notified for different air velocities and the set of all types of VG pairs.

d. Mathematical modeling

The average convective heat transfer coefficient, $h_{c,m}$ and corresponding parameters use for finding the average heat transfer coefficient are determined as follows

$$Re = \rho UD/\mu \qquad (1)$$

Where U and D are the average velocity and hydraulic diameter of the air channel, respectively; ρ and μ.are the density and dynamic viscosity of air

$$h_{c,m} = Q/Ap\Delta t_{lm} \qquad (2)$$

Where Q is the air-side heat transfer rate; Ap is the heat transfer area of copper plate and Δt_{lm} is the logarithmic mean temperature difference between the wall of copper plate and the air.

Q is calculated as

$$Q = Cp\rho UA_c(t_{aout}-t_{ain}) \qquad (3)$$

Where U is the average velocity of air flow; A_c is the cross sectional area of air channel, t_{ain} and t_{aout} are the cross sectional average temperature of air at the inlet and outlet of test section, respectively

The average Nusselt number, Colburn factor and friction factor are employed to describe the thermal and flow characteristics of air channel and are defined as

$$Num=h_{c,m}D/\lambda \qquad (4)$$

$$j = (h_{c,m}/\rho UCp) \times (Cp\mu/k)^{2/3} \qquad (5)$$

$$f = 2\Delta PD/\rho U^2L \qquad (6)$$

The results from the above calculations are then tabulated and presented in the form of graphs. Experimental results are validated after comparing it with various researchers.

For the validation of the setup and measurement, the obtained Nusselt number and friction factor values of the smooth channel is compared with the empirical correlations of Dittuse-Boelter (Num), Gnielinski (Num), Petukhov (f),and Blasius (f) for turbulent flow regions, respectively. These correlations are cited from references

Dittus-Boelter correlation:

$$Num = 0.023\ Re^{0.8}Pr^{0.4} \qquad (7)$$

Gnielinski correlation:

$$Num=0.0214(Re^{0.8}–100)\ Pr^{0.4} \qquad (8)$$

Petukhov correlation:

$$f= (1.82\log10Re–1.64)^{-2} \qquad (9)$$

Blasius correlation:

$$f=0.3164Re^{-0.25} \qquad (10)$$

Where Re is based on the hydraulic diameter D of the channel

III.RESULT & DISCUSSION

a. a Validation of smooth channel

The comparisons shown in Fig. 4 found that Nusselt numbers of the present smooth channel are in reasonable conformity with the Dittuse Boelter and Gnielinski correlation in turbulent region. A graph is drawn for the experimental and theoretical values of N_{um} for different values of Reynold's number. The X-axis in the graph is for the Reynold's number and Y-axis is for the values of N_{um}.

Table no. 2: theoretical and experimental values of N_{um} for the given Reynold's number

Reynold's no.	Experimental values of N_{um}	dittus and boelter equation (N_{um})	gnielinski eqvation (N_{um})
10785.9	31.77	37.27	32.62
12769.51	44.83	42.66	37.63
14381.2	59.75	46.79	41.59
16054.87	68.75	51.24	45.61

The experimental values of Num for the smooth channel (without vortex generators) is calculated for the different Reynolds number and the theoretical values of the Num for these different Reynolds number are calculated from the two correlations which are selected from the references. By comparing the experimental and theoretical values of Num it is found that the experimental values are in reasonable conformity with the theoretical values.

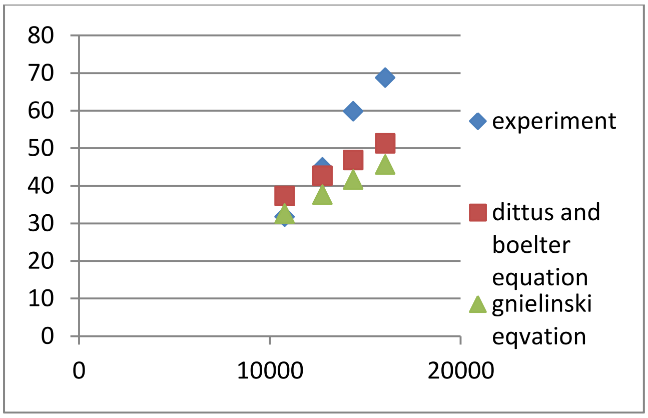

Fig.3 Verification of average Nusselt number for smooth channel

Similarly the experimental values of f factor for the smooth channel (without vortex generators) is calculated for the different Reynolds number and the theoretical values of the f factor for these different Reynolds number are calculated from the two correlations which are selected from the references. By comparing the experimental and theoretical values of f factor it is found that the experimental values are in reasonable conformity with the theoretical values

b. Comparison between different vortex generators

The comparison of different vortex generators shown in fig. 4 found that the rectangular winglet vortex generator has the best heat transfer enhancement followed by curved trapezoidal winglet, delta winglet and trapezoidal winglet. This is due to the fact that RWP has the largest area facing the air flow inducing the strongest longitudinal vortices

Table no. 3: experimental values of j factor

Reynold's number	J_0	j_{RW}	J_{DW}	J_{TW}	J_{CTW}
10785.9	0.0312	0.0495	0.0384	0.0384	0.0454
12769.51	0.037	0.06234	0.05275	0.0452	0.06075
14381.2	0.044	0.07017	0.05776	0.05052	0.0678
16054.87	0.045	0.08597	0.06501	0.05798	0.0738

These are the values of the j factor obtained from the calculations; here j_0 is value of j factor for the smooth channel and other are the values for the four types of vortex generator. A graph is plotted for understanding the comparison of vortex generators. A dimensionless factor is introduced j/j0 which is nothing but the value of ratio of value of j factor for the given vortex generator and the value of j factor for the smooth channelThe Y-axis in the graph is for the value of j/j_0 for the different vortex generator and X-axis is for the values of Reynold's number.

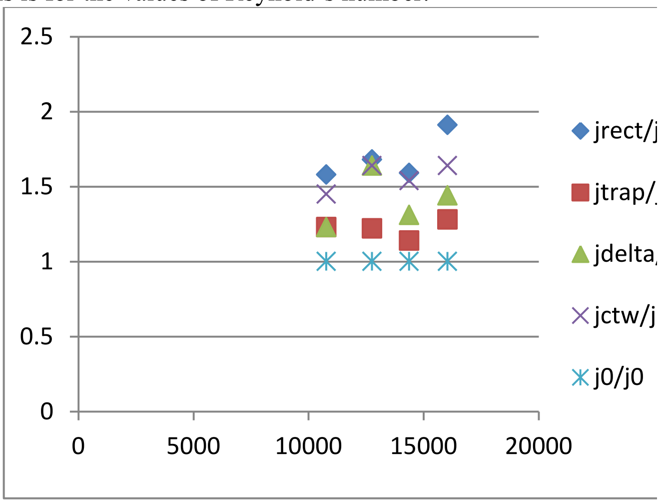

Fig.4 Comparison of performances of the four kinds of VG pairs
(a) j/j_0 vs. Re

Similarly by finding the values of f factor for the four Reynolds numbers the comparison of values of f factor for all the four types of vortex generators with the f_0 factor obtained for the smooth channel can be done and also by finding out the values of R (thermohydraulic performance factor), $(j/j_0)/(f/f_0)$ for each type of vortex generator the value of heat transfer enhancement can be determined.

IV.CONCLUSION

a The performance of a pair of new vortex generators CTW has been investigated experimentally and compared with traditional vortex generators rectangular winglet, trapezoidal winglet and delta winglet. DWP is the best in transitional flow region, while CTWP has the best thermohydraulic performance in fully turbulent region due to the streamlined configuration and then the low pressure drop, which indicates the advantages of using this kind of vortex generators for heat transfer enhancement.

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