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Numerical Calculation of Heat Transfer in a Rectangular Channel with Triangular Vortex Generators in the Channel at Blade Angle 30° for Reynolds no. of 800, 1200

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Abstract: The vortex generators in this study are placed at angle of 30° in a rectangular channel. These vortices create the down wash flow towards the lower channel wall while the up wash flow is away from the wall which is found in the outside region of the vortices. Along the downstream direction, the secondary velocity vectors decreases while the distance between the vortex cores increases. Thinning of the thermal boundary layer thus occurs in between the two vortices.
The variation of span wise averaged Nusselt number was calculated at different values of x at Reynolds number of 800, 1200. We considered parametric studies between channel without winglet and with winglet pair. It shows that the span wise averaged Nusselt number increase with the increment of Reynolds no. at particular x. it is observed the behaviour at these Reynolds no with blade angle 30°. The increment in the span wise averaged Nusselt no. can be observed from these numerical calculations.
the effect of vortex generators in the spanwise averreged nusselt number. The comparison between the plots of spanwise avereged Nusselt number with and without vortex generators shows that there is a considerable increase in Nusselt number near the vortex generator region and this effect decreases as we move far from the vortex generator region in the stream wise direction.
Keywords: Vortex Generators, Reynolds Number, Nusselt Number, rectangular channel

I. INTRODUCTION

The compact heat exchanger is widely used in fields such as automobile industry, heating and air conditioning, power system, chemical engineering, electronic chip cooling and aerospace, etc. The subject of heat transfer enhancement is of significant interest in developing compact heat exchanger to meet the desire of high efficiency and low cost with the volume as small as possible and the weight as light as possible. The use of ribs/baffles placing in the cooling channels or channel heat exchangers is one of the commonly used passive heat transfer enhancement technique in single-phase internal flows. Periodically positioned ribs/baffles in the channels interrupt hydrodynamic and thermal boundary layers. Downstream of each rib/baffle the flow separates, recalculates, and impinges on the channel wall and these effects are the main reasons for heat transfer enhancement in such channels. The use of ribs/baffles increases not only the heat transfer rate but also substantial the pressure loss. The rib/baffle geometry and arrangement in the channel also alter the flow field resulting in different convective heat transfer distribution. In particular, the angled ribs, the rib cross-section, the rib-to-channel height ratio and the rib pitch-to-height ratio are all parameters that influence both the convective heat transfer coefficient and the overall thermal performance. It is, thus, difficult to realize the advantage of rib/baffle arrangements or geometry and the use of staggered ribs/ baffles with rib height and pitch spacing of 0.1 (0.5 for baffles) and 1 time of the channel height respectively is often recommended in most of the previous work.

II. LITERATURE REVIEW

Compact heat exchangers have wide applications in power, process, automotive and aerospace industries. Performance of these heat exchangers can be improved by adding protrusion type vortex generators such as fins, ribs, wings, winglets, etc. on the gas side of the core. When longitudinal vortex generators are placed near a heat transfer surface, they increase the heat transfer by transporting fluid from the wall into the free stream and vice versa. The effectiveness of a vortex generator in enhancing the heat transfer depends on the vortex strength generated per unit area of the vortex generator. The first use of longitudinal vortex was mentioned by Schuauer and Spangenberg [1960] and since then research on LVG's has increased. A delta winglets pair kept at an angle of attack is very effective as

III. COMPUTATIONAL FLUID DYNAMICS

Computational Fluid Dynamics also generally called CFD is an important branch of fluid mechanics and it uses numerical methods and algorithms to analyze and solve fluid flow problems. It has become popular since the previous methods, experimental and theoretical are either very expensive, time consuming, or involve too much labor. In CFD, computers are used to solve the algorithms that define and analyze the fluid flow. Due to the increase in the computational capabilities over time and better numerical solving methods, most experimental and theoretical work has been done using CFD. CFD is not only cost effective but it helps one analyze and simulate complex geometries, heat transfer, and shock waves in a fluid flow. It also helps solve partial differential equations (PDE) of any order in a fluid flow. CFD mainly helps analyze the internal or external fluid flow. The use of CFD has become increasingly popular in branches of engineering such as Aerospace to study the interaction of the propellers or rotors with aircraft fuselage, Mechanical to obtain temperature distribution of a mixing manifold, Bio-medical engineering to study the respiratory and circulatory systems. There are a few simple generic steps that must be followed for CFD analysis. The following scales and non-dimensional variables are used,

$$x^* = \frac{x}{L},$$

$$u^* = \frac{u}{V},$$

$$P^* = \frac{p}{\rho V^2},$$

$$\frac{\delta}{\delta x^*} = \frac{1}{L}\frac{\delta}{\delta x^*},$$

$$Re = \frac{VL}{\Theta} = \frac{V^2/L}{\Theta V/L^2},$$

$$T^* = \frac{T - T_{in}}{T_w - T_{in}}$$

The continuity equation in a non-dimensional form for a steady state flow is given by,

$$\frac{\delta u^*}{\delta x^*} + \frac{\delta v^*}{\delta y^*} + \frac{\delta w^*}{\delta z^*} = 0$$

The momentum equation in a non-dimensional form for a steady state flow is given by,

$$u^*\frac{\delta u^*}{\delta x^*} + v^*\frac{\delta u^*}{\delta y^*} + w^*\frac{\delta u^*}{\delta z^*} + \frac{\delta P^*}{\delta x^*} = \frac{1}{Re}\left(\frac{\delta^2 u^*}{\delta x^{*2}} + \frac{\delta^2 u^*}{\delta y^{*2}} + \frac{\delta^2 u^*}{\delta z^{*2}}\right)$$

$$u^*\frac{\delta v^*}{\delta x^*} + v^*\frac{\delta v^*}{\delta y^*} + w^*\frac{\delta v^*}{\delta z^*} + \frac{\delta P^*}{\delta y^*} = \frac{1}{Re}\left(\frac{\delta^2 v^*}{\delta x^{*2}} + \frac{\delta^2 v^*}{\delta y^{*2}} + \frac{\delta^2 v^*}{\delta z^{*2}}\right)$$

$$u^*\frac{\delta w^*}{\delta x^*} + v^*\frac{\delta w^*}{\delta y^*} + w^*\frac{\delta w^*}{\delta z^*} + \frac{\delta P^*}{\delta z^*} = \frac{1}{Re}\left(\frac{\delta^2 w^*}{\delta x^{*2}} + \frac{\delta^2 w^*}{\delta y^{*2}} + \frac{\delta^2 w^*}{\delta z^{*2}}\right)$$

The energy equation in a non-dimensional form for a steady state flow is given by,

$$\frac{\delta(u^*T^*)}{\delta x^*} + \frac{\delta(v^*T^*)}{\delta y^*} + \frac{\delta(w^*T^*)}{\delta z^*} + \frac{\delta P^*}{\delta z^*} = \frac{1}{Re\ Pr}\left(\frac{\delta^2 T^*}{\delta x^{*2}} + \frac{\delta^2 T^*}{\delta y^{*2}} + \frac{\delta^2 T^*}{\delta z^{*2}}\right)$$

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IV. GEOMETRIC MODEL

The rectangular channel has cross section dimensions of 25 cm x 10 cm. Length of the channel is 200 cm. The vortex generators have the dimensions length x breadth x height of 1cm x .4 cm x 1 cm. The blade angle for the trangular vortex generators is taken 30°. The bottom wall material is taken as Aluminium and the fluid flowing is water. The flow in the channel is with 3 % turbulent intensity and enters the channel with uniform constant velocity across the cross section. The backflow turbulent intensity at outlet is also taken as 3 %. The Reynolds numbers are chosen as 800, 1200, 1600 and 2000. Shown in Figure 4.2 is the isometric view of the channel along with vortex generators.

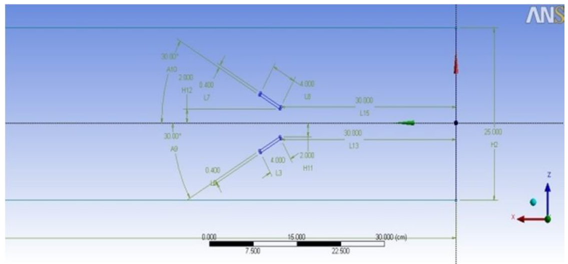


Fig. 4.3 Geometric model with angle b/w vortex generators 30° showing the dimensions (in cm)

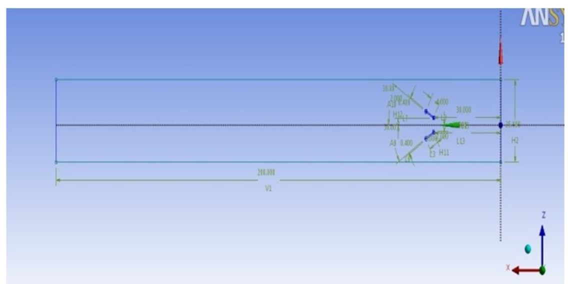


Fig. 4.4 Geometric model with angle b/w vortex generators 30° showing the dimensions (in cm

A. Grid Generation

In this study ANSYS Workbench was used as a meshing tool. The structured non uniform grid was generated. The fig. 4.5 shows the fine meshing near the vortex generators. Fine meshing is needed near the wall region to capture the boundary layer on the wall. The whole fluid domain has all the elements as hexa elements. Mapped Face Meshing option from the ANSYS Workbench meshing tool was used to generate this kind of structured non uniform meshing. Bias factor of 10 was used to make the grids near the wall and the vortex generators fine to capture the boundary layer effects

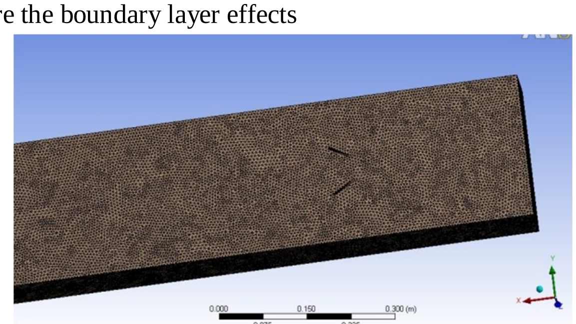


Fig. 4.7 Tetrahedral messing view of the fluid domain meshing.

B. FLUENT Setup

FLUENT 14.0 was used for CFD analysis in this study. Importing the mesh files created in ANSYS Workbench, the model is setup to allow energy equation in a Realizable k-Ɛ model with enhanced wall treatment. The fluid in this study is water with a constant density of 998.2 kg/m^3, dynamic viscosity of 0.001003 kg/m.s, the constant pressure specific heat is 4182 J/Kg-K, and thermal conductivity is 0.6 W/m.K. The operating condition on the interior of the channel is fluid while the before mentioned boundary conditions are applied on the rectangular channel including the inlet and outlet channel.

A temperature of 373K is applied on the surface of the heated section which is the bottom surface of the rectangular channel and all the surface of the vortex generator. The inlet has been given an inlet temperature of 300 K and a specific velocity based on the Reynolds number corresponding to the chosen Reynolds number. The outlet has zero pressure thus implying ambient condition. The walls of the whole channel, as well as surfaces of the vortex generator, have been given the no slip boundary condition. A second order upwind discretization method has been used for energy and momentum. Convergence is based on the absolute criteria of continuity, x velocity, y velocity and z velocity equal to 10^{-4} and energy equal to 10^{-8}. This means the solution will converge once the residuals reach the above mentioned mark. The model is computed from the inlet surface and 1000 iterations were given for the solution to converge.

C. Grid Independency Validation

Different sets of grids were considered for different geometries. The grid independence was conducted by simulating the channel with or without vortex generators at Re = 800, 1200.. The element numbers for the geometries is shown in Table 4-2.

Table 4-2

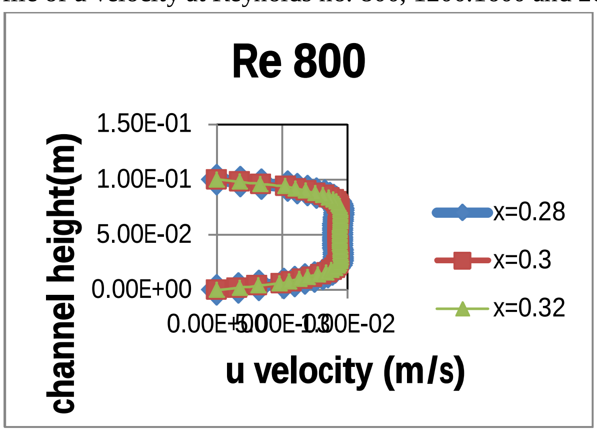
Geometry	Element Numbers
β= 30°	1731773

V. RESULTS AND DISCUSSION

Numerical simulation of the flow in the channel with heated bottom wall has been carried out. In this chapter feature of the flow at a blade angle 30° at different Re are discussed in detail.

A. Analysis of Performance of Vortex Generator with 30° blade angle Reynolds no. 800, 1200.

Fig. 5.1 to 5.16 shows the velocity profile of u velocity at Reynolds no. 800, 1200.1600 and 2000. at β= 30



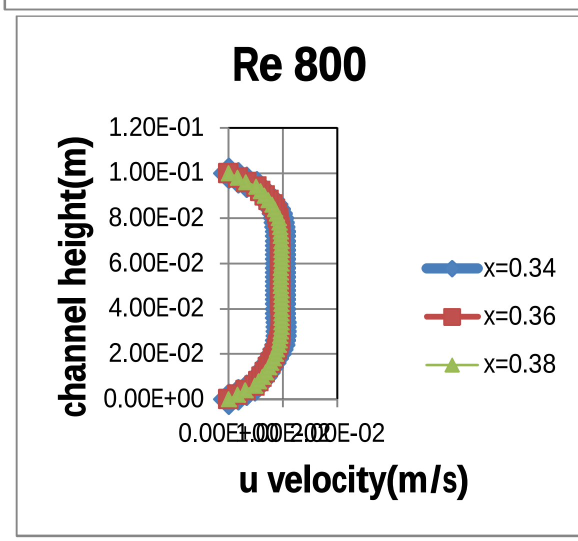


Figure 5.2: Velocity Profile at different planes with x varying from 0.34m to 0.38m for Re=800

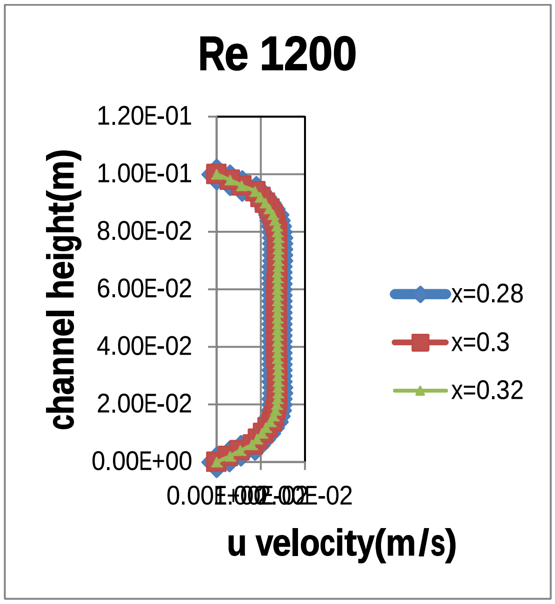


Figure 5.4: Velocity Profile at different planes with x varying from 0.28m to 0.32m for Re=1200

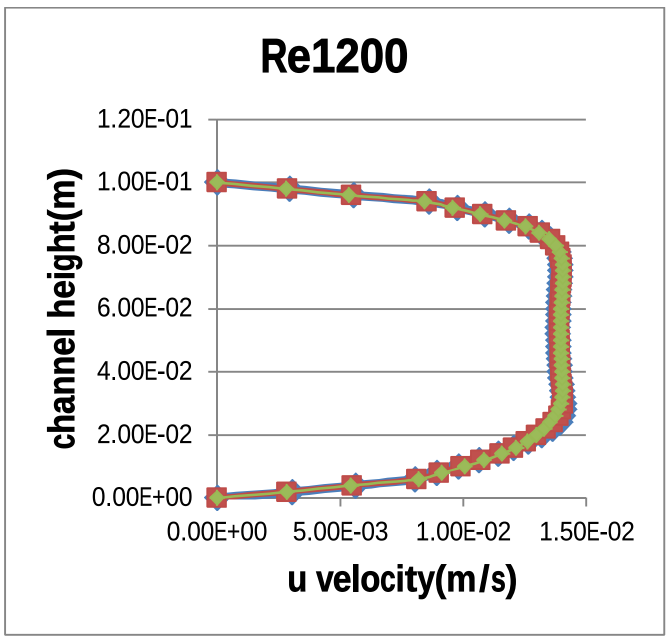


Figure 5.3: Velocity Profile at different planes with x varying from 0.34m to 0.38m for Re=1200

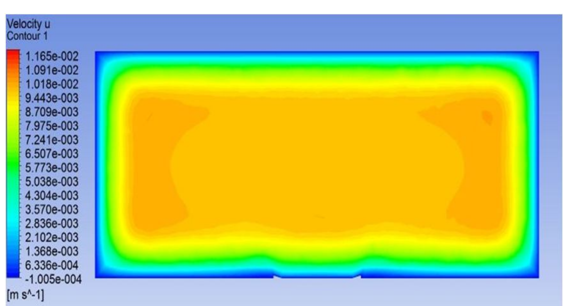


Fig. 5.8 Contour of u velocity, Re 800, β = 30° (yz plane, x=30cm)

B. Flow Structure

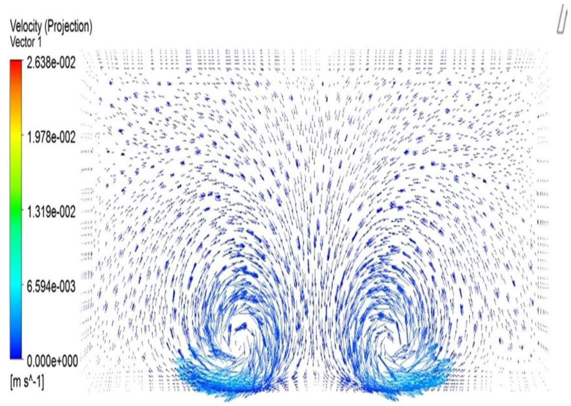
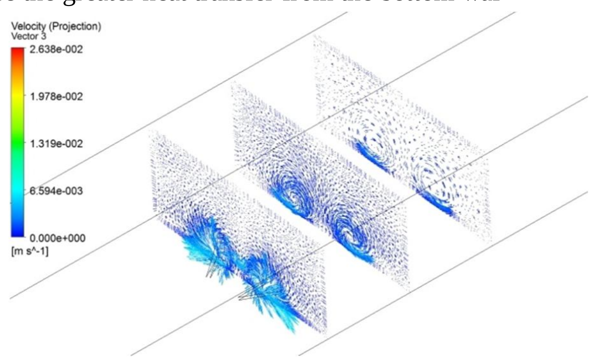


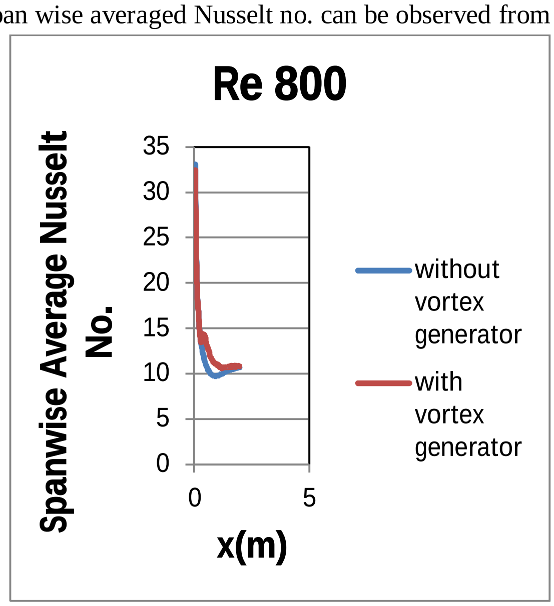
Fig. a Formation vortices on yz plane at x=42cm,R 1200,β=30°

Fig.b shows that vortex strength is maximum near the vortex generators and decrease continuously in stream wise direction. So the mixing near the vortex generators cause the greater heat transfer from the bottom wal



VI. HEAT TRANSFER

The variation of span wise averaged Nusselt number was calculated at different values of x at Reynolds number of 800, 1200. We considered parametric studies between channel without winglet and with winglet pair. It shows that the span wise averaged Nusselt number increase with the increment of Reynolds no. at particular x. Fig. 5.a to 5.d show the behavior at these Reynolds no with blade angle 30°. The increment in the span wise averaged Nusselt no. can be observed from these figures.



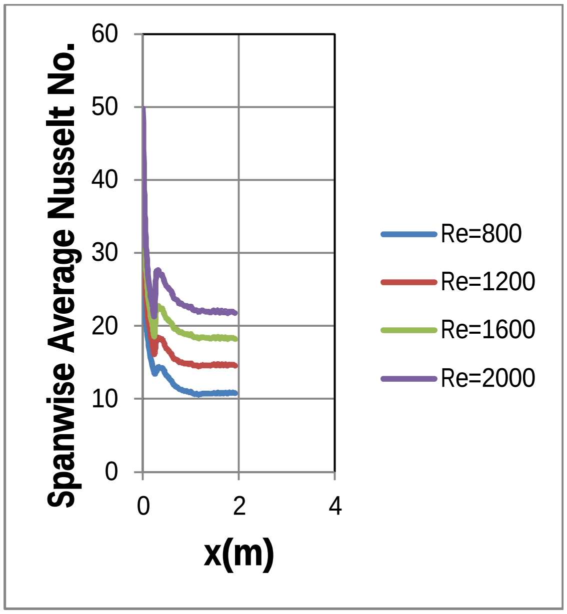
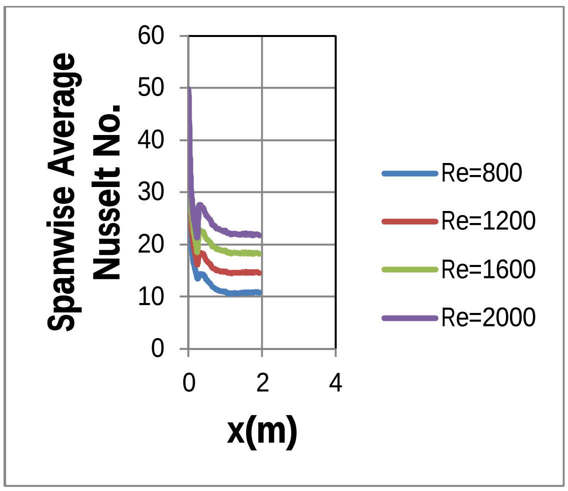


Fig.5.e shows the combined plot for all the cases dicussed. The plot shows that with increase in Reynolds number the span wise avereged Nusselt number increase for the particular x. It is also observed that with increase in the Reynolds number the rate of increase in span wise avereged Nusselt number near the vortex genreator region high,means the vortex generators are more effective at higher reynolds number and causes greater heat transfer from the bottomwall



Above figures show the comparison between the previous work (S.R. Hiravennavar, E.G. Tulapurkara, G. Biswasb 2019) and the present work. The curve behavior is observed same. The difference between the values of Nusselt number is due to difference in the channel and vortex generators dimensions**.**

Table below shows the improvement of heat flux (in percentage) from the bottom wall by the use of vortex generators

Table 5-1

Reynolds No.	β = 30°
800	8.841
1200	7.262

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VII. CONCLUSIONS

In this study of augmentation of heat transfer in a rectangular channel with triangular vortex generators was evaluated. The span wise averaged Nusselt no., mean temperature and total heat flux were compared with and without vortex generators in the channel at blade angle 30° for Reynolds no. of 800, 1200 respectively.

1) The use of vortex generators increases the span wise averaged Nusselt no. than the case of without vortex generators considerably.
2) At particular angle, by increasing the Reynolds no. the overall performance increases and span wise averaged Nusselt no. was found to be greater at particular location for larger Reynolds number.
3) The mean temperature of the bulk fluid at particular location in the flow direction was increased by the use of vortex generators.
4) The total heat flux from the bottom wall with vortex generators was found greater than the case of without vortex generators [table 5-1]. Also this difference increase with increase in Reynolds number.

REFERENCES

[1] The first use of longitudinal vortex was mentioned by Schuauer and Spangenberg [2012]
[2] A delta winglets pair kept at an angle of attack is very effective [2015].
[3] Laminar channel flow (Fiebig et al., 2018). In the case of the heat exchangers the flow on gas side is usually laminar
[4] Edwards and Alker (1974) find that the delta winglets provided a higher overall heat transfer enhancement, com-pared to cubes placed on a flat plate.