NUMERICAL INVESTIGATION ON TURBULENCE EFFECTS OF DELTA WING VORTEX GENERATORS ON THERMAL PERFORMANCE OF OVAL FIN HEAT EXCHANGER

[1]Naveen Kumar, [2]Santosh A Bellad, [3]Sartaj Patel, [4]K. Ganesh
[1,4]Associate Professor, [2,3]Assistant Professor
Department of Mechanical Engineering
Kuppam Engineering College, Kuppam, Chittoor dist (AP), India

Abstract: In this paperwork, numerical analysis of heat transfer flow characteristics using Delta wing vortex generator in fin and tube oval heat exchanger has been presented. Heat transfer 3D numerical model has been developed and successfully carried out. The fin-and-oval heat exchanger with punched delta wings for a range of $500 \leq Re \leq 2500$. The influences of the punched angle 450, flow directions, wing tips pointing downstream and upstream of pitch ratio 2 is investigated. The numerical results showed that in the range of the present study, the variation of these parameters can result in the increase of heat transfer. The study focuses on the Influence of the different parameters of Vortex Generators on heat transfer and fluid flow characteristics. The characteristics of the average Nu number and friction coefficient are studied numerically by the use of the computational fluid dynamics (CFD) commercial code of FLUENT ANSYS 14. The results show that the use of the punched delta wings in the fin-and-oval-tube heat exchanger leads to an enhancement in the heat transfer and friction as compared to the plain fin for all cases (Nu/Nu0 and f/f0 higher than 1) with the increasing of Re number. It has been observed that the overall Nu number of oval tubes increasing by 20.33%, 30.78%, 38.89%, 46.26% and 54.92% with an angle of 45^0 and Reynolds numbers 500, 1000, 1500, 2000 and 2500.

***IndexTerms* - Fin oval tube Heat Exchanger, CFD, Nusselt Number, Heat transfer etc.**

I. INTRODUCTION

Fin and Tube heat exchanger is widely used in industries such as air conditioning, refrigeration, automobile, petrochemical, electronic clip cooling, and chemical engineering [1]. The heat transfer coefficient on the air side is normally very low due to the thermophysical properties of air. A lot of investigations were mainly focused on circular-tube-fin and oval-tube-fin with mounted and punched longitudinal vortex generators to enhance the heat transfer [2–10]. Then heat transfer enhancement on the air side of fin and tube heat exchanger is necessary. Over the past several decades, much attention has been devoted to the heat transfer enhancement on-air side of fin and tube heat exchangers. The temperature distribution reflects the thermal resistance of a heat exchanger; moreover, it is relevant to the velocity field. The velocity field can be modified through flow manipulation and can improve the heat transfer performance by flow manipulator. 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. Local heat transfer coefficient was measured on the fin-tube heat exchanger with winglet using a single heater of two-inch diameter and five different positions of winglet type vertex generators. The measurements were made at Reynolds number about 2250 [11]. The T-W formed on the tape was used as vortex generators for enhancing the heat transfer coefficient by the breakdown of the thermal boundary layer and by the mixing of fluid flow in tubes [12].

II. CONSTRUCTION

A. GEOMETRY AND COMPUTATIONAL DOMAINS

Basic geometry for this configuration is taken from literature and dimensions of the configuration shown below. Inlet Domain length is considered as is 10B for fully developed flow and Outlet Domain length is considered as 20B to avoid any recirculation effects.

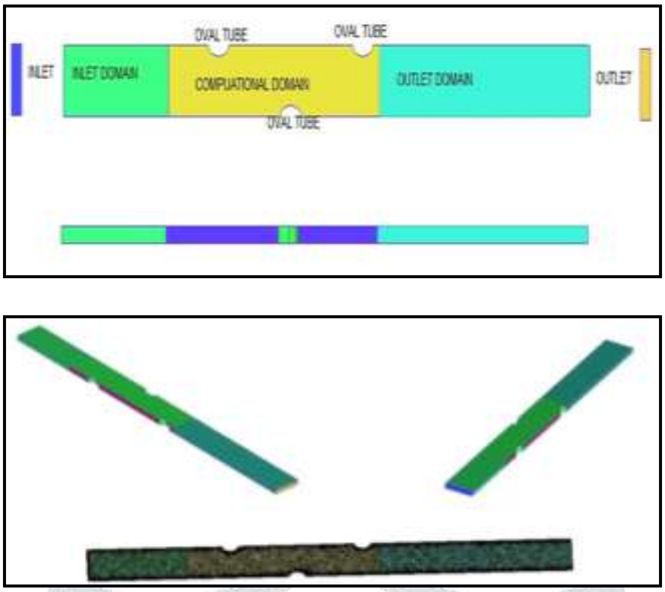

Fig.1. Basic Geometry for Experimental Validation

B. GEOMETRY WITH WINGLETS

Basic Geometry with winglets of pitch ratio 2, 4 and 6 has been considered here. Winglets have been placed on both the bottom and the top face of geometry with angle 45^0. Here, winglet has been considered as an equilateral triangle of side 3.6 mm. The distance between the two winglets is 3.6, 7.2 and 10.8 mm for pitch ratios 2, 4 and 6 respectively.

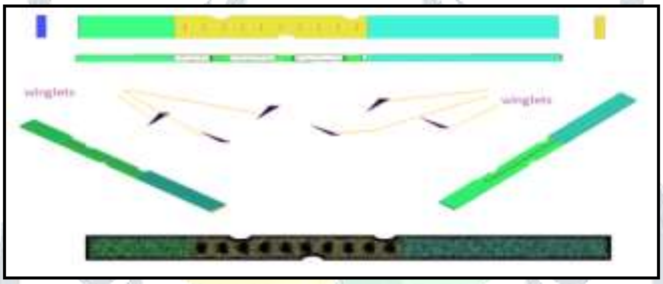

Fig.2. Geometry with winglets

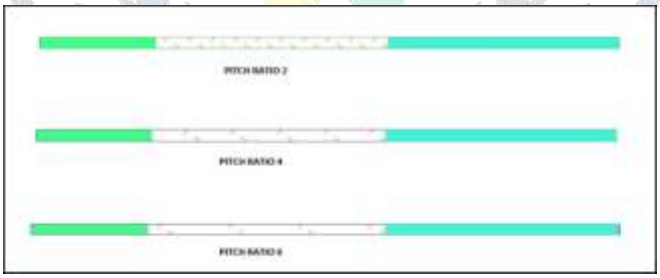

Fig.3. Comparison pitch ratios of winglets

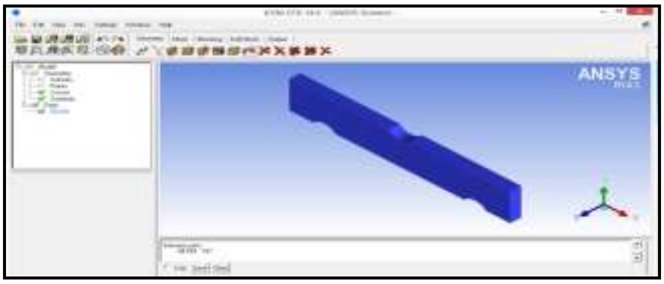

Fig.4.Modeling rectangular duct with oval type pin fins

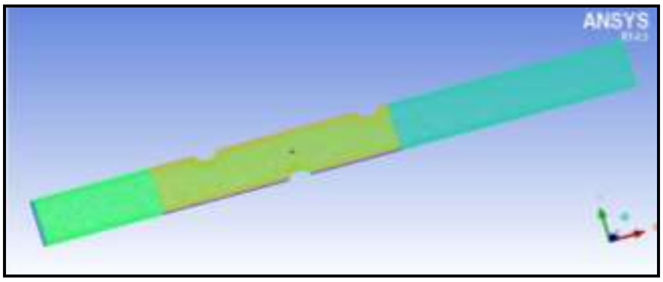

Fig.5. Three-dimensional meshing file

A meshing of this entire configuration has been done using ANSYS ICEMCFD commercial software.

- Unstructured meshing has been used for grid generation
- In unstructured mesh the nodes equal to half of the nodes.
- The shape of the element is triangles or tetrahedral in an unstructured mesh.
- In the structured mesh, the element shape is hexahedra or quadrilateral
- The Governing equations and boundary conditions are solved by using computational fluid dynamics toll called Fluent.

C. COMPUTATIONAL DOMAIN AND ANALYSIS

A meshing of this entire configuration has been done using Ansys ICEMCFD commercial software. Unstructured meshing has been used for grid generation. The Governing equations and boundary conditions are solved by using computational fluid dynamics toll called Fluent. The convective term and diffusive term in governing equations for momentum and energy equations are solved by using a second-order upwind scheme. Ansys Fluent a cell-centered based Finite volume solver. The inlet velocities are 1.3, 2.6, 3.9, 5.2 & 6.5 m/s have been considered for Reynolds numbers are 500, 100, 1500, 2000 and 2500 respectively.

III. RESULT AND DISCUSSION

Analysis has been carried out without winglet and with winglet of pitch ratios 2, 4 and 6. The various figures has been drawn to compare Nusselt Number, Friction factor, Nusselt Number ratio, Friction factor, and Thermal Enhancement Factor to find out the effect of winglets on thermal performance of oval tube heat exchanger.

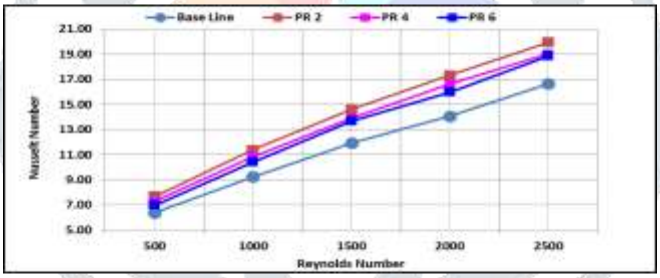

Fig.6.Nusselt Number variation with Reynolds Number

From the above figure, it can be observed that Nusselt number is higher for a configuration with winglet pitch ratio 2 and lower for baseline case. Nusselt number is inversely proportional to pitch ratio. Higher the pitch ratio, turbulence mixing is less and lowers the pitch ratio higher the turbulent mixing.

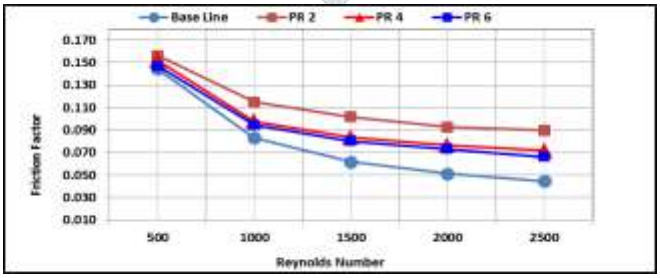

Fig.7. Friction factor variation with Reynolds Number

From the above charts, it can be observed that friction factor is higher for a configuration with winglet pitch ratio 2 and lower for baseline case. The friction factor is inversely proportional to pitch ratio. Higher the pitch ratio, turbulence effect is less and lowers the pitch ratio higher the turbulent effect i.e. higher turbulence effect causes for the increase in friction factor.

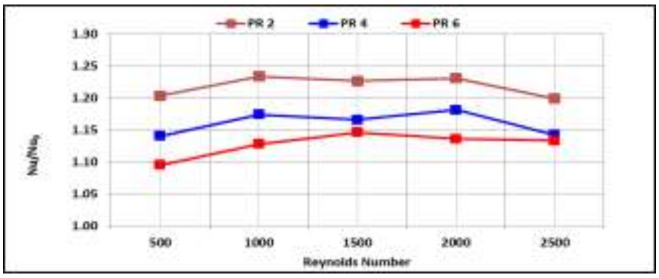

Fig.8.Nusselt Number ratio variation with Reynolds Number

From the above charts, it can be observed that the Nusselt number ratio is higher at Reynolds number 1000 for pitch ratio 2. For pitch ratio 4 Nusselt number is high at Reynolds number 2000 and for pitch ratio 6, A Nusselt number high at Reynolds number 1500.

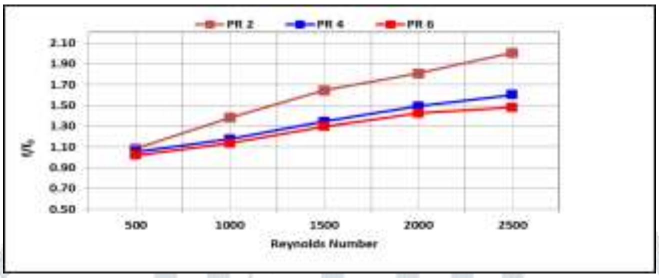

Fig.9. Friction factor ratio variation with Reynolds Number

From the above charts, it can be observed that Friction facto is increases with a decrease in pitch ratio for each Reynolds number. The difference in an increase of friction factor increases with Reynolds number. Due to the increase in velocities, frictional losses are high.

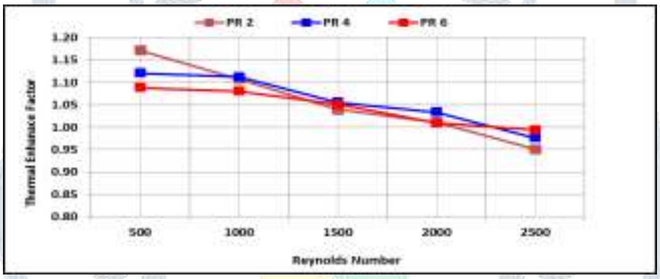

Fig.10. TEF ratio variation with Reynolds Number

The above figure shows that Thermal enhancement factor is maximum for Reynolds number 500 for pitch ratio 2. Thermal enhancement factor decreases with increase in Reynolds numbers increases in friction factor compared to baseline case is more compared to increase in Nusselt number compared to the baseline case. From the above chart it can be concluded that optimized configuration is winglet with pitch ratio 2 and for Reynolds number 500, winglet with pitch ratio 4 for Reynolds number 1000. For Reynolds number 1500, winglet with pitch ratio 4 is optimized configuration. For Reynolds number 2000, winglet with pitch ratio 4 is optimized. For Reynolds number 2500, Winglet with pitch ratio 6 is optimized. Based on Inlet velocity of flow, pitch ratio can be chosen to maximize Thermal enhancement factor. Without winglet configuration has been validated with experimental and it shows good agreement with experimental results. The same analysis has been used in the same configuration with winglets of pitch ratio 2, 4 and 6 for Reynolds numbers 500, 1000, 1500, 2000 and 2500.

IV. CONCLUSION

Analysis has been carried out without winglet and winglet with pitch ratios 2, 4 and 6. By doing this we can conclude that thermal enhancement factor decreases with increase in Reynolds number increases in friction factor compared to baseline case is more compared to increase in Nusselt number compared to the baseline case. Optimized configuration is winglet with pitch ratio 2 and for Reynolds number 500, winglet with pitch ratio 4 for Reynolds number 1000. For Reynolds number 1500, winglet with pitch ratio 4 is optimized configuration. For Reynolds number 2000, winglet with pitch ratio 4 is optimized. For Reynolds number 2500, Winglet with pitch ratio 6 is optimized. Based on Inlet velocity of flow, pitch ratio can be chosen to maximize Thermal enhancement factor.

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