Numerical Simulation On Fin And Oval Tube Heat Exchanger With Longitudnal Vortex Generators

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ABSTRACT- Fin-and-tube heat exchangers are widely used in industries such as air conditioning, refrigeration, etc. The heat transfer coefficient on the air side is normally very low due to the thermo physical property of air. Thus the air-side thermal resistance is the dominant part of total thermal resistance of a fin-and-tube heat exchanger. The temperature distribution reflects the thermal resistance of a heat exchanger. The velocity field can be modified through flow manipulation and so we can improve the heat transfer performance by flow manipulator. Vortex generator is one promising flow manipulator, which introduce vortices into the flow passage causing heat transfer enhancement. Three-dimensional numerical study will be performed using ANSYS Fluent 13.0 for heat transfer characteristics of fin-and-oval-tube heat exchangers with longitudinal vortex generators (LVGs).

1. INTRODUCTION

A heat exchanger is a complex device that provides the transfer of thermal energy between two or more fluids, which are at different temperatures and are in thermal contact with each other. Heat exchangers are used either individually or as components of a large thermal system, in a wide variety of commercial, industrial and household applications, e.g. power generation, refrigeration, ventilating and air-conditioning systems, aerospace industries, electronic chip cooling as well as in environmental engineering. The improvements in the performance of the heat exchangers have attracted many researchers for a long time as they are of great technical, economical, and not the least, ecological importance. Performance improvement becomes essential particularly in heat exchangers with gases because the thermal resistance of gases can be 10 to 50 times as large as that of liquids which requires large heat transfer surface area per unit volume on gas side.

The traditional methods of reducing the air-side thermal resistance are by increasing the surface area of the heat exchanger, or by reducing the thermal boundary layer thickness on the surface of the heat exchanger.. One of the methods to reduce boundary layer thickness is by the generation of passive vortices. In this technique the flow field is altered by an obstacle to generate a vortex oriented in the direction of the flow. The resulting change in the flow due to an obstacle alters the local thermal boundary layer. The net effect of this manipulation is an average increase in the heat transfer for the affected area.

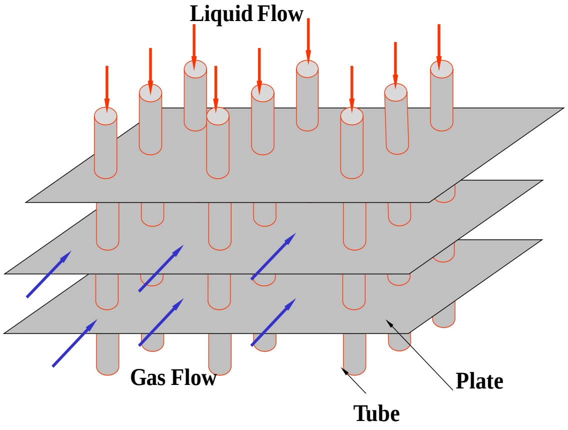


Fig.1. Schematic diagram of core region of a fin-and-oval-tube heat exchanger

2. MODEL DESCRIPTION

2.1.Physical Model

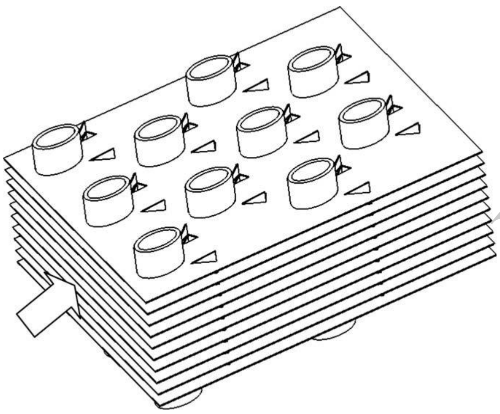


FIg.2.Schematic diagram of core region of a fin-and-oval-tube heat exchanger with LVGs

The schematic diagram of a fin & oval tube heat exchanger with LVGs is shown in fig 2.There are four basic LVGs – delta wing, rectangular wing, delta and rectangular winglet pairs (see fig 3).In this present study, we adopt the delta wing vortex generator due to recommendation that wings produce the same heat transfer enhancement for less pressure penalty in comparison with winglets. A pair of delta wing is punched out of the fin surface symmetrically behind each oval tube. The height of the winglets is equal to the channel height (H), so that the delta winglets can also serve as pitch holders for the fins.

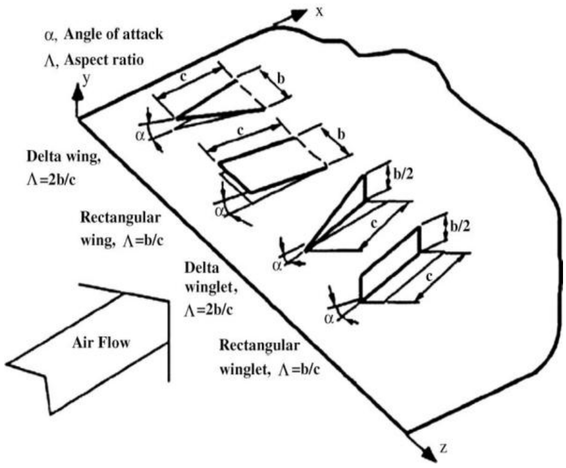
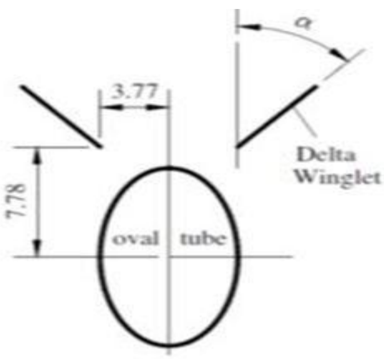


Fig.3. Four basic LVGs forms and the associated geometrical definitions

Fig.4 shows the dimensions of delta winglets and their placement with respect to the oval tube. The aspect ratio K(2b/c) for delta winglet equals 2. The delta winglet pairs are placed in ‘‘common flow-down” arrangement. Due to the symmetric arrangement, the region occupied by dashed lines in Fig. 5 is selected as the computational domain. Fig. 5 gives a top view of the computation domain of a 3-tube-row fin-and-oval-tube heat exchanger with LVGs.

a



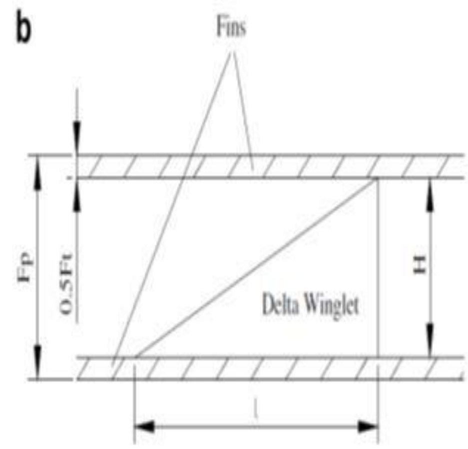


Fig.4. Winglet vortex generator dimensions and the placement with respect to the oval tubes: (a) top view of LVGs and oval tube; (b) side view of LVGs

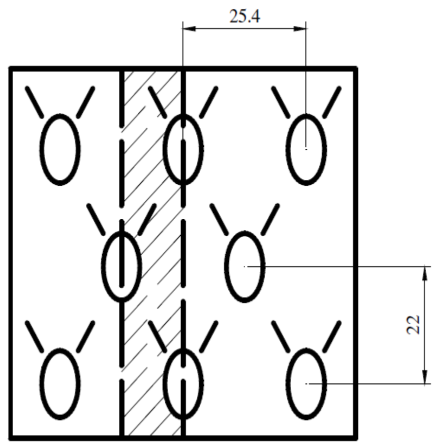


Fig.5. Schematic diagram of computational domain

Two neighboring fins form a channel of height H = 3.2 mm, width B = 12.7 mm, and length L = 64.4 mm. The first oval tube, of semi-major diameter Ra = 6.28 mm. and semi-minor diameter Rb = 3.77 mm, is located at Y = 17.92 mm from the inlet of the channel. The longitudinal tube pitch Pl is 22 mm and the span wise tube pitch Ps is 25.4 mm. The tube rows are arranged in a staggered design. The fin material is aluminum and fin thickness Ft = 0.33 mm. The actual computation domain is extended by 10H at the inlet to maintain the inlet velocity uniformity and the domain is extended by 30H at the exit to ensure a recirculation-free flow there. Owing to the large parameter space for LVGs, our following investigations are performed with a fixed aspect ratio A= 2.

3. LITERATURE SURVEY

O'Brien [1] investigated a combined experimental and numerical heat transfer enhancement techniques that may be applicable to large-scale air-cooled condensers such as those used in geothermal power applications. The research is focused on whether air-side heat transfer can be improved through the use of finsurface vortex generators (winglets,) while maintaining low heat exchanger pressure drop. **HERPE.et.al[2]** analyzed the entropy production rate induced by longitudinal vortex generators punched on the fins of a compact heat exchanger. The flow is assumed to be three-dimensional, unsteady and laminar. The entropy production rate due to heat fluxes and drag forces is studied by means of finite volume methodology. The heat conduction in the fin is also taken into account and hence the induced thermal entropy production is also evaluated in the fin. The effects of the fin efficiency (factor Fi) and of the angle β of the vortex generator are evaluated for the volumetric entropy production rate in the fin and in the flow field. **Russell et al. (1982)** reported the first use of longitudinal vortices to enhance finned tube heat exchanger performance. They conducted experiments using a transient melt line method to assess the effectiveness of several vortex generator configurations and, after considering delta and winglets, it was found that a rectangular winglet in two staggered rows was most promising. They implemented this geometry in a full-scale finned flat tube heat exchanger with the winglets at angle of attack β=20° and the holes (due to punching the winglet) downstream of the winglet. Measured data were compared with plain-tube correlations, and at a Reynolds number of 500 the j factor was enhanced by about 47% while the f factor increased by 30%. At a Reynolds number of 1000, the j factor increased by about 50% and f factor increased by only 20%. **Biswas et al. (1994b)** numerically investigated the flow structure and heat transfer in a three-row fin-tube heat exchanger with built-in delta winglet pairs. The staggered array of tube rows , and a punched-out delta winglet pair with an aspect ratio of 2 and an attack angle of 45° was located behind each tube with $\Delta x/D = 0.5$ and $\Delta y/D = 1.0$. At a Reynolds number of 500, the local heat transfer was found to increase by more than 240% at a location about 12 times the channel height downstream of the inlet. The spanwise average Nusselt number at Re=646 compared favorably with the experimental results from the same geometry for most of the stream wise locations. **Kwak et al. (2003)** performed experiments to find the heat transfer and pressure loss penalty for 2,3,4,5 transverse rows in staggered fin tube bundle with a single row of winglet pairs besides the front row of the tube bundles. The pairs of winglets were placed with common-flow-up configuration. The heat transfer enhancement of 30-40% and pressure loss reduction by 55 to 34% was recorded with an increase in Reynolds number (350 to 2100), with 3 rows of tube bundles and a single transverse row of winglets beside the front row of the tubes. The reduction of the pressure loss penalty for three rows of tube bundles was found to be largest in comparison with the other numbers of rows. **Allison and Dally (2007)** experimentally found the effects of delta-winglet vortex generators on the performance of a fin and tube radiator. The winglets were arranged in flow-up configuration, and placed directly upstream of the tube. The study included dye visualization and full scale heat transfer performance measurements. The results were compared with a standard louver fin surface. 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

4. OBJECTIVES OF THE WORK

1) To study the heat transfer and pressure drop encountered in a heat exchanger with a delta wings mounted on fins. **2)** To study the performance of the delta wing at various angles of attack using ANSYS. **3)** To compare the heat transfer characteristics with the experimental results. **4)** To study the CFD simulation on fin and oval tube heat exchanger without delta wing. **5)** Perform CFD simulation on fin and oval tube heat exchanger with delta wing and compare the effects in terms of Nusselt number and HTC values using ANSYS Fluent.

5.BASELINE RESULTS AND DISCUSSIONS

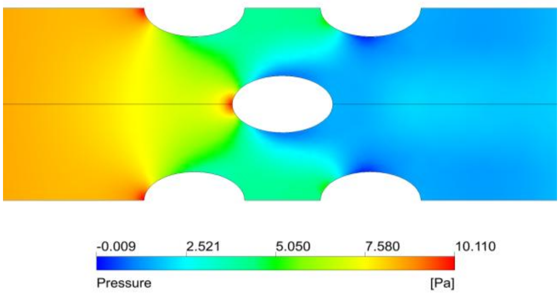


Fig.5. 1 Pressure profiles on the middle cross-section for without delta wing Re-500

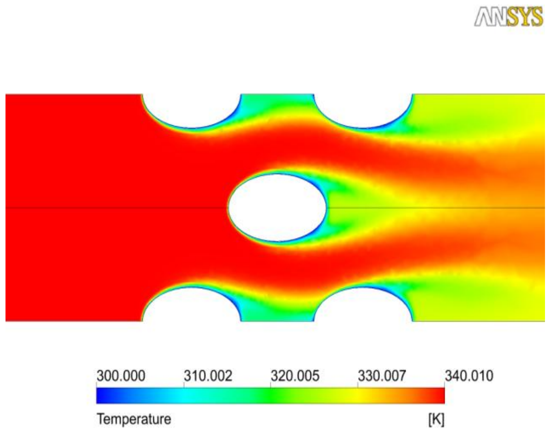


Fig.5. 2 Temperature profiles on the middle cross-section for without delta wing Re-500

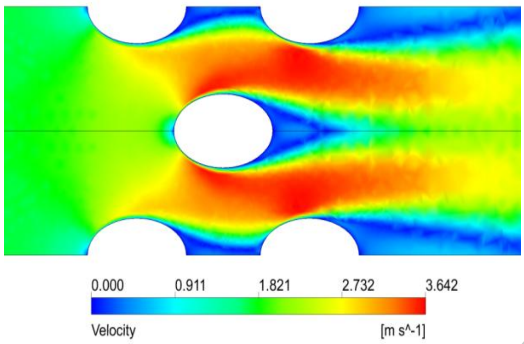


Fig.5. 3 Velocity profiles on the middle cross-section for without delta wing Re-500

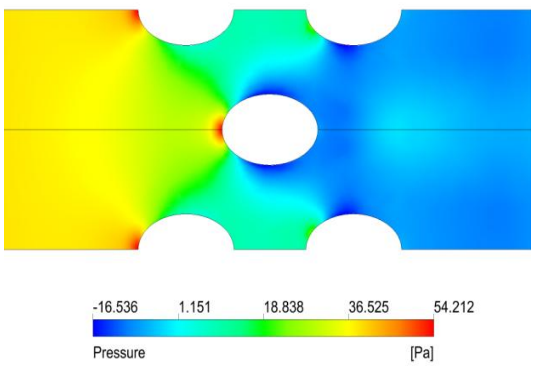


Fig.5. 4 Pressure profiles on the middle cross-section for without delta wing Re-1500

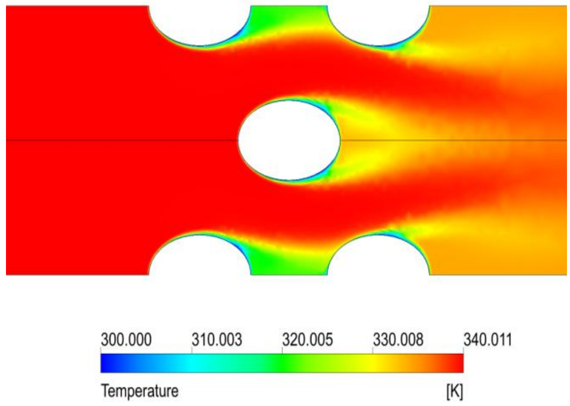


Fig.5. 5 Temperature profiles on the middle cross-section for without delta wing Re-1500

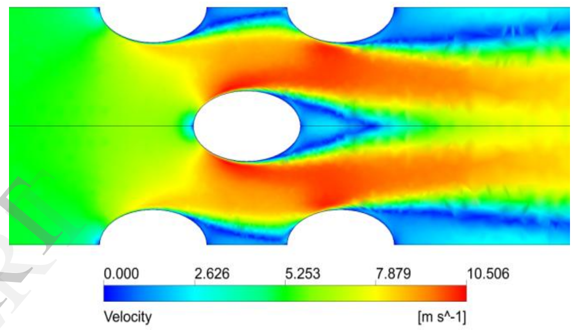


Fig.5. 6 Velocity profiles on the middle cross-section for without delta wing Re-1500

5.1 EXPERIMENTAL–NUMERICAL COMPARISON OF NU AND DP FOR MODEL VALIDATION

The inlet air velocity ranges from 1.3 m/s to 5.5 m/s and the corresponding Re number ranges from 500 to 2000. The predicted results are compared with the experimental results from Kays and London [22]. The average Nu number and the overall pressure loss penalty DP are shown in Figs. 5.7 and 5.8, respectively. As we can see from the figures, the average discrepancy between the predicted Nu number and the experimental values is less than 10% and the average discrepancy between the predicted pressure loss and the experimental values is less than 2%. The good agreement between the predicted and experimental results indicates that the numerical model is reliable to predict heat transfer characteristics and flow structure in compact heat exchangers.

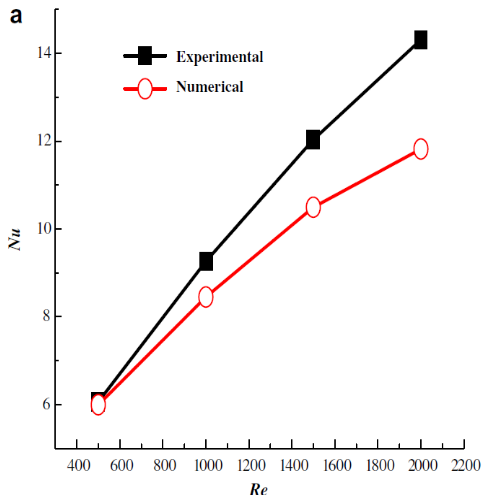


Fig.5. 7 Experimental–numerical comparison effect of Re on Nu number

Velocity m/s	Heat Flux W/m2	Pressure Drop Pa	Outlet Temperature K
1.36	2239.77	9.98455	334.54
2.72	3397.46	25.4231	336.003
4.08	4328.1	45.6187	336.63
5.44	5062.06	69.7646	337.053

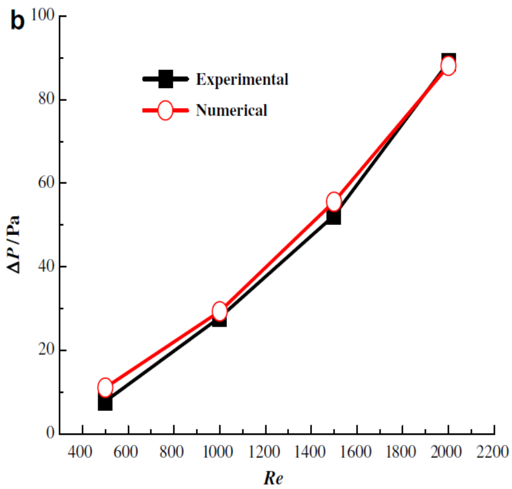


Fig.5. 2 Experimental–numerical comparison effect of effect of Re on DP

6.OPTIMIZATION AND CFD ANALYSIS OF ANGLE OF ATTACK WITH LVGS

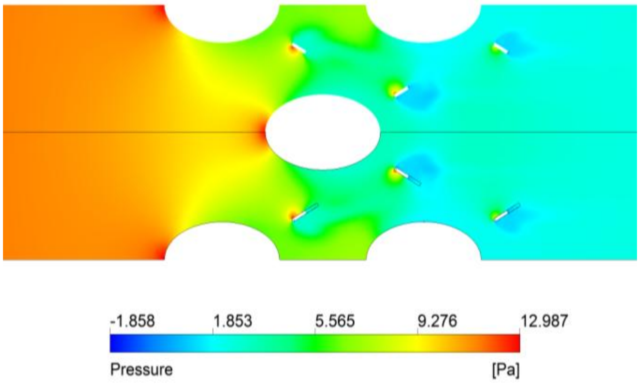


Fig.6. 1 Pressure profiles with delta wing angle of attack = 30^0 and Re-500

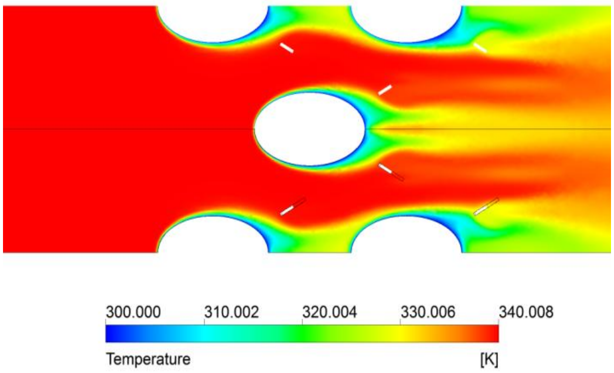


Fig.6. 2Temperature profiles with delta wing angle of attack = 30^0 and Re-500

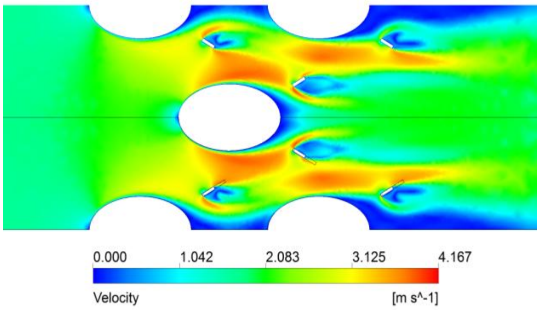


Fig.6. 3 Velocity profiles with delta wing angle of attack = 30^0 and Re-500

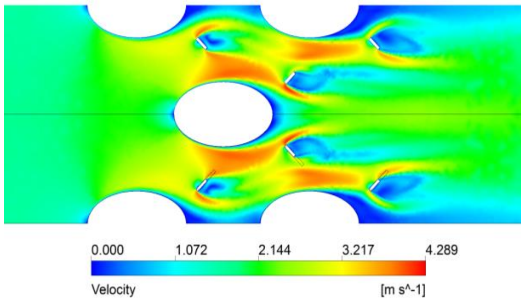


Fig.6. 6 Velocity profiles with delta wing angle of attack = 45^0 and Re-500

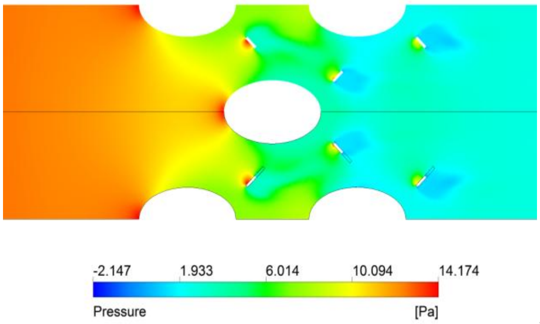


Fig.6. 4 Pressure profiles with delta wing angle of attack = 45^0 and Re-500

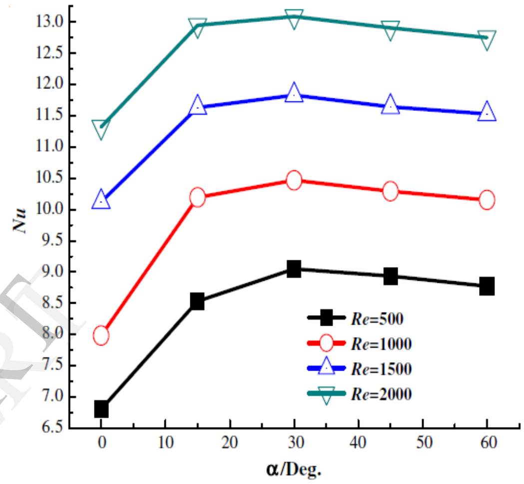


Fig.6. 7 Influence of angle of attack α on Nu

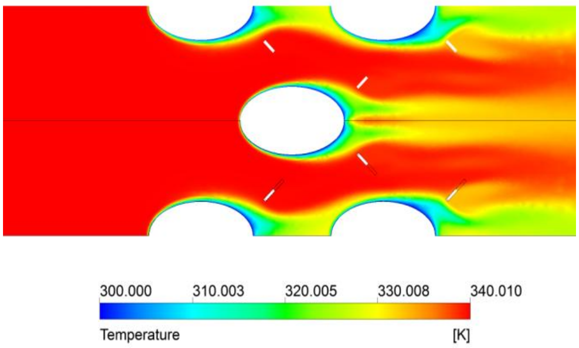


Fig.6. 5 Temperature profiles with delta wing angle of attack = 45^0 and Re-500

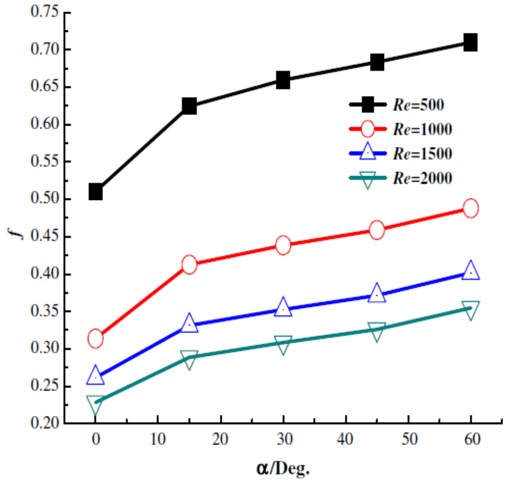


Fig.6. 8 Influence of angle of attack α on friction factor

Fig.6.7 shows the relation of the average Nu number and the angle of attack α. From the figure we can see that first the average Nu number increases with the increasing angle of attack α, then the average Nu number reaches the maximum at the attack-angle α = 30 degree, and finally the average Nu number decreases with the increase of attack-angle α when α > 30. The change trend of the average Nu numbers is almost the same for different Re number which ranges from 500 to 2000. It should be noted that it is impossible to generate pure longitudinal vortices and transverse vortices are always accompanied. Obviously, the magnitude of transverse vortices increases with the increasing angle of attack. In steady flow, transverse vortices do not improve convective heat transfer from the walls to the main flow, because the transverse vortices with their steady two dimensional flow have no convective mechanism for heat transfer augmentation. However, in steady flow, longitudinal vortices can increase heat transfer significantly by generating a three dimensional spiraling motion. In addition, the vortex may break down when the angle of attack is too large. There always exists an optimum attack-angle for the heat transfer augmentation. With the increase of angle of attack ($\alpha < 30^0$), the strength of the generated LVs become increasingly stronger, so the average Nu number increases with the increasing attack-angle α. However, for a large angle of attack (α > 30) the combined influence of transverse vortices and vortex breakdown may vitiate heat transfer enhancement and lead to a decrease on the average Nu number.

The effect of angle of attack a on the friction factor f is presented in Fig. 6.8. It can be seen that with the increase of angle of attack α, the friction factor f will always increase. The form drag of LVGs increases with the increasing attack-angle α, which leads to the increase of flow loss penalty in the channel, and so the friction factor f increases. The change trend of the friction factor is almost the same for different Re number.

7. CONCLUSIONS

In this thesis, three-dimensional numerical simulations are employed to investigate the heat transfer characteristics and flow structure in full-scale fin-and-oval-tube heat exchangers with LVGs. The heat conduction in fins and the thickness of LVGs are considered. The geometrical parameters with respect to fin-and oval- tube heat exchangers with LVGs are also optimized. In order to explore the fundamental mechanism for heat transfer augmentation, the numerical results are analyzed from the point view of the field synergy principle. The results obtained in this work can be well explained through the field synergy principle and it demonstrates that the reduction of the average intersection angle h between the velocity field and the temperature field is one dominant reason for heat transfer enhancement. The major conclusions are drawn Owing to the existence of complex stream wise vortices system in the flow passage, the temperature field in the passage is distorted and the heat transfer between the fluid and its neighboring surfaces is significantly enhanced. The stream wise vortices also removes the zone of poor heat transfer from the recirculation region behind oval tube. For a three-row fin-and-oval-tube heat exchanger with LVGs, the average Nu number is augmented by 13.6–32.9% and the corresponding friction factor f increases by 29.2–40.6%, in comparison with fin-and-oval-tube heat exchanger without LVGs, for the Reynolds number ranging from 500 to 2500.

Further investigations are needed to optimize longitudinal vortex generators for different heat transfer situations.

8. REFERENCES

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