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Enhancement of Heat Transfer in Crossed Flow Heat Exchangers with Aid of Oval Tubes and Multiple Delta Winglets

Mr. Lakpathi Kethavath[1], Ms.K.Chaitanya[2], Mr.Vadivel.J [3]
[1]*Dept. of Mechanical Engg. Assistant professor, Aurora engineering college, Bhongir Hyderabad Telangana India*
[2]*Dept. of Mechanical Engg. Assistant professor, Aurora engineering college, Bhongir Hyderabad Telangana India*
[3] *Dept. of Mechanical Engg. Assistant professor, Aurora engineering college, Bhongir Hyderabad Telangana India*
Corresponding Author: Mr. Lakpathi .Kethavath

Abstract: *A three-dimensional study of laminar flow and heat transfer in a channel with built-in oval tube and delta winglets is carried out through the solution of the complete Nervier–Stokes and energy equations using a body-fitted grid and a finite-volume method. The geometrical configuration represents an element of a gas–liquid fin–tube cross-flow heat exchanger. The size of such heat exchangers can be reduced through enhancement of transport coefficients on the air (gas) side, which are usually small compared to the liquid side. In a suggested strategy, oval tubes are used in place of circular tubes, and delta-winglet type vortex generators in various configurations are mounted on the fin-surface. Delta winglets were provided inside the oval tubes to create turbulence in the fluid which is flowing inside the tubes. These winglets acts like obstruction for the fluid and makes the fluid to change its direction and suddenly reduces the velocity over the place. Then a turbulence is created at the place and there will be an improvement in time required for the flow due to this the fluid will have enough time to touch the surface of the winglets. Due to which the fluid will transfer more heat to the winglets enhancing heat transfer. By changing the angles of the winglets various results can be obtained. An Evaluation of the strategy is attempted in this investigation. The investigation is carried out for different angles of attack of the winglets to the incoming flow for the case of two winglet pairs. The variation of axial location of the winglets is also considered for one pair of winglets mounted in common-flow-down configuration. The structures of the velocity field and the heat transfer characteristics have been presented. The results indicate that vortex generators in conjunction with the oval tube show definite promise for the improvement of fin–tube heat exchangers.*
Keywords*: Heat transfer enhancement, oval-tubes, Delta winglets, cross flow heat exchangers.*

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I. Introduction

A heat exchanger consists of heat transfer elements such as a core or matrix containing the heat transfer surface, and fluid distribution elements such as headers, manifolds, tanks, inlet and outlet nozzles or pipes, or seals. Usually, there are no moving parts in a heat exchanger; however, there are exceptions, such as a rotary regenerative exchanger (in which the matrix is mechanically driven to rotate at some design speed) or a scraped surface heat exchanger. The heat transfer surface is a surface of the exchanger core that is in direct contact with fluids and through which heat is transferred by conduction. That portion of the surface that is in direct contact with both the hot and cold fluids and transfers heat between them is referred to as the primary or direct surface. To increase the heat transfer area, appendages may be intimately connected to the primary surface to provide an extended, secondary, or indirect surface. These extended surface elements are referred to as fins. Thus, heat is conducted through the fin and convected (and/or radiated) from the fin (through the surface area) to the surrounding fluid, or vice versa, depending on whether the fin is being cooled or heated. As a result, the addition of fins to the primary surface reduces the thermal resistance on that side and thereby increases the total heat transfer from the surface for the same temperature difference. Fins may form flow passages for the individual fluids but do not separate the two (or more) fluids of the exchanger. These secondary surfaces or fins may also be introduced primarily for structural strength purposes or to provide thorough mixing of a highly viscous liquid. Not only are heat exchangers often used in the process, power, petroleum, transportation, air-conditioning, refrigeration, cryogenic, heat recovery, alternative fuel, and manufacturing industries, they also serve as key components of many industrial products available in the marketplace. These exchangers can be classified in many different ways. They are classified according to transfer processes, number of fluids, and heat transfer mechanisms. Conventional heat exchangers are further classified according to construction type and flow arrangements. Another arbitrary classification can be made, based on the heat transfer surface area/volume ratio, into compact and non-compact heat exchangers. This classification is made because the type of equipment, fields of applications, and design techniques generally differ.

II. Literature Review

Motivation For Heat Transfer Enhancement

For well over a century, efforts have been made to produce more efficient heat exchangers by employing various methods of heat transfer enhancement. The study of enhanced heat transfer has gained serious momentum during recent years, however, due to increased demands by industry for heat exchange equipment that is less expensive to build and operate than standard heat exchange devices. Savings in materials and energy use also provide strong motivation for the development of improved methods of enhancement. When designing cooling systems for automobiles and spacecraft, it is imperative that the heat exchangers are especially compact and lightweight. Also, enhancement devices are necessary for the high heat duty exchangers found in power plants (i.e. air-cooled condensers, nuclear fuel rods). These applications, as well as numerous others, have led to the development of various enhanced heat transfer surfaces. In general, enhanced heat transfer surfaces can be used for three purposes:

(1) To make heat exchangers more compact in order to reduce their overall volume, and possibly their cost

(2)To reduce the pumping power required for a given heat transfer process

(3) To increase the overall UA value of the heat exchanger.

A higher UA value can be exploited in either of two ways:

(1) To obtain an increased heat exchange rate for fixed fluid inlet temperatures, or

(2) To reduce the mean temperature difference for the heat exchange; this increases the thermodynamic process efficiency, which can result in a saving of operating costs.

Enhancement techniques can be separated into two categories: passive and active. Passive methods require no direct application of external power. Instead, passive techniques employ special surface geometries or fluid additives which cause heat transfer enhancement. On the other hand, active schemes such as electromagnetic fields and surface vibration do require external power for operation.

The majority of commercially interesting enhancement techniques are passive ones. Active techniques have attracted little commercial interest because of the costs involved, and the problems that are associated with vibration or acoustic noise. This paper deals only with gas-side heat transfer enhancement using special surface geometries. Special surface geometries provide enhancement by establishing a higher hA per unit base surface area. Clearly, there are three basic ways of accomplishing this:

1. Increase the effective heat transfer surface area (A) per unit volume without appreciably changing the heat transfer coefficient (h). Plain fin surfaces enhance heat transfer in this manner.

2. Increase h without appreciably changing A. This is accomplished by using a special channel shape, such as a wavy or corrugated channel, which provides mixing due to secondary flows and boundary-layer separation within the channel. Vortex generators also increase h without a significant area increase by creating longitudinally spiraling vortices exchange fluid between the wall and core regions of the flow, resulting in increased heat transfer.

3. Increase both h and A. Interrupted fins (i. e. offset strip and louvered fins) acting this way. These surfaces increase the effective surface area, and enhance heat transfer through repeated growth and destruction of the boundary layers.

The performance of fin-and-tube heat exchangers are related to geometric parameters. Early experiment results achieved by Rich (1973). Lu et al. (2011) illustrated the effects of geometric parameters such as fin pitch, tube pitch, fin thickness and tube diameter in detail. The optimum value for $Q/\Delta P$ was found by numerical simulation. Tang et al. (2009) analysed the air-side heat transfer and friction characteristics of 5 types of fins. Besides, in order to enhance performance, different kinds of methods are used in finned tube heat exchangers. The effects of the attack angle of delta winglet pair were achieved by Wu et al. (2012) with numerical and experimental methods. It was turned out that the average Nusselt Number of winglets with attack angle of 60° was higher than that of winglets with attack angle of 45° by experiment and computational method. He et al. (2012) used winglet type of vortex generators to enhance air-side heat transfer performance. Another method was carried out by Tao et al. (2007) who used triangular wavy fins to make the performance better. It can be seen that vortex generators and wave fins are always made to enhance the heat transfer on the air-side. The hydrophobic prosperities are very important for chemistry applications. Wang et.al (2002) describes the air-side heat transfer performance of a hydrophilic coating on plain-fin surface. To enhance performance, different kinds of methods are used in finned tube heat exchangers. As is known to all, oval tubes have better performance than that of cylinder ones (Schulenb,1966). An experiment study was performed to compare the performance between cylinder tube and oval tube by James E.O (2004). The heat transfer and pressure drop characteristics of plate fin-and-oval-tube heat exchanger were analyzed numerically by Erek et al (2005). A three-dimensional numerical study of oval tube and the winglet pairs which can improve the heat transfer significantly was analyzed by Tiwalri et al (2003). The results indicated that the contribution of winglets pairs in heat transfer was more than 43.86 %, undoubtedly. The cooling delta angle on power plant natural draught dry cooling towers was analysed with porous media approach by Wang et al (2011).In general, heat exchangers are not placed horizontally but arranged with an angle to the ground in industrial application. In this paper, the inlet angle characteristics of a fin-and-oval-tube heat exchanger have been studied numerically. Because of the non-orthogonal layout of heat exchanger, we much concern about the thermal characteristics of fin-and-oval-tube heat exchanger with different inlet angles. So, 5 different inlet angles of fin-and-oval-tube heat exchanger unite are analysed by FLUENT in the present study.

The first literature reporting the enhancement of heat transfer of using surface protrusion vortex generators is by Edwards and Alker [1]. They noted a maximum increase in the local Nusselt number of 40%. Russell et al. [2] presented the first study on the air-side heat transfer enhancement using vortex generators for the heat exchanger. The numerical studies by Biswas et al. [3] and Jahromi et al. [4] showed that the heat transfer in the wake region can be enhanced significantly in the presence of winglet type longitudinal vortex generators behind the tubes. Extensive studies have been done on heat transfer characteristics and flow structure for heat exchangers with longitudinal vortex generators (LVGs). In recent years, the application of vortex generators in compact heat exchangers has received more and more attention.

An experimental study was conducted by Torii et al. [5] to obtain heat transfer and pressure loss in a fin-and-tube heat exchanger with in-line or staggered tube banks with delta winglet vortex generators of various configurations. The winglets were placed in a special orientation to augment heat transfer and reduce form drag. They showed that in case of staggered tube banks, the heat transfer was augmented by 30–10%, and the corresponding pressure loss was reduced by 55–34% for the Reynolds number ranging from 350 to 2100. Gentry and Jacobi [6] experimentally explored the heat transfer enhancement by delta-wing-generated tip vortices in flat-plate and developing channel flows. They reported that on the complete channel surface the largest spatially averaged heat transfer enhancement was 55% accompanied by a 100% increase in the pressure drop relative to the same channel flow with no delta-wing vortex generator. Leu et al. [7] numerically and experimentally studied the heat transfer and flow in the plate-fin and tube heat exchangers with inclined block shape vortex generators mounted behind the tubes. They pointed out that the proposed heat transfer enhancement technique is able to generate longitudinal vortices and to improve the heat transfer performance in the wake regions.Sommers and Jacobi[8] experimentally investigated the air-side heat transfer enhancement of a refrigerator evaporator using vortex generation. They noted that for air-side Reynolds numbers between 500 and 1300,the air-side thermal resistance was reduced by 35–42% when vortex generation was used.

Pesteeiet al. [9] experimentally studied the effect of winglet location on heat transfer enhancement and pressure drop in fin-tube heat exchangers. They found that the winglet pairs were most effective to enhance the heat transfer coefficients when these were placed in the downstream side.

Hiravennavar et al. [10] numerically studied the flow structure and heat transfer enhancement by a winglet pair of nonzero thickness. They observed that in comparison with a channel without winglets, the heat transfer was enhanced by 33% when single winglet is used and by 67% when awinglet pair was employed. Joardar and Jacobi [11] numerically investigated the flow and heat transfer enhancement using an array of delta-winglet vortex generators in a fin-and-tube heat exchanger. They adopted ‘‘common-flow-up” arrangement for vortex generators in three different configurations. The 3VG-inline-array configuration achieves enhancements up to 32%in total heat flux and 74% in j factor over the baseline case, with an associated pressure drop increase of about 41%. Wu and Tao [12] investigated the laminar heat transfer in aligned three row fin-and-tube heat exchangers with longitudinal vortex generators from the viewpoint of field synergy principle. There are few reports of implementation of longitudinal vortex generators in fin-and-oval-tube heat exchangers in open literature [18].

Chen et al. [13] numerically explored the conjugate heat transfer of a finned-oval tube with a Punched longitudinal vortex generator in form of a delta winglet. They mainly focused on one element of the heat exchanger (only one oval tube involved). Based on their work, they further investigated the heat transfer enhancement of finned oval tube with inline longitudinal vortex generators [14] as well as with staggered longitudinal vortex generators [15], and they obtained some useful optimized results. Tiwari et al. [16] numerically studied the laminar flow and heat transfer in a channel with built-in oval tube and delta winglet vortex generators. They revealed that combinations of oval tube and the winglet pairs improved the heat transfer significantly.O’Brien et al. [17] experimentally investigated the forced convection heat transfer in a narrow Rectangular duct fitted with an elliptical tube and one or two delta-winglet pairs. They found that the addition of the single winglet pair to the oval-tube geometry yielded significant heat transfer enhancement, averaging 38% higher than the oval-tube, no-winglet case. The corresponding increase in friction factor associated with the addition of the single winglet pair to the oval-tube geometry was moderate.

The foregoing literature review shows that only limited numerical analyses on fin-and-oval tube heat exchangers with LVGs have been published. The previous investigations were mainly focused on parametric study, and heat transfer enhancement was only explored from the traditional perspective. In fact, that is insufficient to uncover the fundamental causes of heat transfer enhancement. Moreover, most of the previous studies of fin-and-oval-tube heat exchangers with LVGs are focused on single heat transfer element (only one tube involved) [18]. Therefore, P. Chu et al. [18] numerically investigated the heat transfer characteristics and fluid flow structure of fin-and-oval-tube heat exchangers with longitudinal vortex generators (LVGs)for ranges from 500 to 2500. They found that the average *Nu* for the three-row fin-and-oval tube heat exchanger with LVGs increased by 13.6–32.9% over the baseline case and the corresponding pressure loss increased by 29.2–40.6%. The study carried out with three geometrical parameters -placement of LVGs (upstream and downstream), angles of attack (α =15o, 30o, 45o and 60o) and tube-row number (n = 2, 3, 4 and 5). The results show that the LVGswith placement of downstream, angle of attack of 30o and minimum tube-row number provide the best heat transfer performance. Continually with the last reference, this paper presents a two-dimensional numerical investigation of laminar flow and heat transfer characteristics at constant wall heat flux condition, over a three rows oval-tube bank in staggered arrangement with rectangular vortex generators, for Reynolds number ranging from 250 to 1500. The study focuses on the effects of position and angle of attack of VGs on heat transfer and fluid flow characteristics of oval-tube heat exchanger. Behind each oval-tube, a pair of rectangular winglet VG is situated with placement ofdownstream. Six different positions (3 in *x*-axis and 2 in *y*-axis) and two angles of attack *α* (30and 45 degrees) are investigated. The Influence of the different parameters on heat transfer enhancement(average Nusselt number of tubes), and skin friction coefficient characteristic are studied numerically by the aid of the computational fluid dynamics (CFD) commercial code of FLUENT version 6.3.

Physical modeling

The plan-view representation of the computational domain is shown in Fig. 3.1the dimensions used are those of a proposed design being reported elsewhere by the authors. Two neighboring fins form a channel of height H,width B ¼ 11:25H and length L ¼ 13:75H. The built-in oval tube, of semi-major diameter a ¼ 4:40Hand semi-minor diameter b ¼ 1:465H, is located at a distance L1 ¼ 6:87H from inlet of the channel. The tube center is located at X ¼ 6:87H, Y ¼ 5:625H.The winglets are thin triangular devices (shown on top right of Fig. 3) placed vertically on the fin surface, with their horizontal axis from the tip either angled outward or inward from the centerline. The position of the winglets is shown as W1 and W2. If it is angled outward (see W1 in Fig. 3), it is called common-flow down(CFD) configuration and if angled inwards (W2 in same figure), it is the common-flow-up (CFU) configuration. The axial distance (X11) between the leading edge of the first winglet pair in common-flow-down configuration and the channel inlet is 1.63H. The transverse distance (Y11) between the channel centerline and the leading edge of either winglet is 2.23H. The axial distance (X12) between the trailing edge of the either winglet and the channel inlet is 3.38H. The transverse distance (Y12) between the channel centerline and the trailing edge of either winglet is 3.69H. The other winglet of the first winglet pair is placed symmetrically about the channel centerline. The axial distance (X21) of the leading edge of second pair of winglets in common flow-up configuration from the inlet of channel is 3.96Hand transverse distance (Y21) from the centerline of the channel is 5.33H. The axial distance (X22) between trailing edge of the second pair of winglet and channel inlet is 5.71H and the distance (Y22) of it from the channel centerline is 3.88H. The length of all the winglets is 2.27H and their height is h ¼ 0:5H. Fig. 3.1 shows a layout of various configurations in which winglet pairs are mounted in the present study. Computations are performed for each of these configurations. Air has been considered as the working fluid, hence the Prandtl number is taken as 0.7. The winglets and oval tube areassumed to be at the temperature of the channel wall.

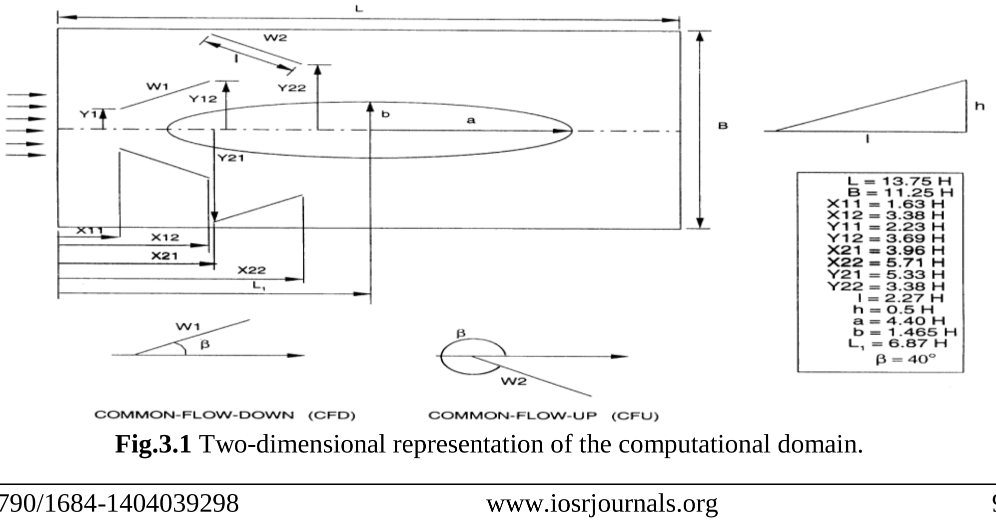

Fig.3.1 Two-dimensional representation of the computational domain.

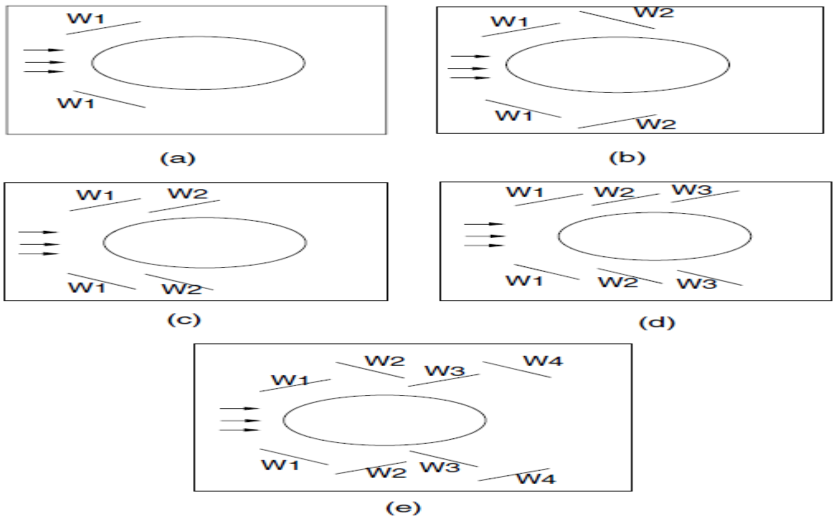

Fig.3.2 Various configurations of the winglet pairs. (a) One winglet pair (W1) in CFD. (b) Two winglet pairs (W1 in CFDand W2 in CFU). (c) Two winglet pairs (W1 and W2 both inCFD). (d) Three winglet pairs (W1, W2 and W3 in CFD). (e)Four winglet pairs (W1 and W3 in CFD; W2 and W4 in CFU)

Numerical modeling

As stated in the foregoing paragraph, the entropy production terms are determined from the temperature and velocity fields. As a consequence, the accuracy of the entropy production field calculation is directly linked to the quality of the temperature and velocity fields obtained by the CFD analysis. Several previous works [20–22,25] have enforced the use of laminar three-dimensional numerical simulations for finned oval tube heat exchangers at low Reynolds numbers. In particular hen and co-workers have performed accurate comparison between their numerical results and experimental data (see references in [20] for more details) in order to validate their CFDinvestigations.Our computational results, obtained by using the numerical method described in following paragraphs, have been successfully compared to Chen's numerical data. Indeed numerical/experimental comparisons of the temperature distribution on the fin and span-averaged pressure distribution across the finned oval tube element have shown perfect agreement.

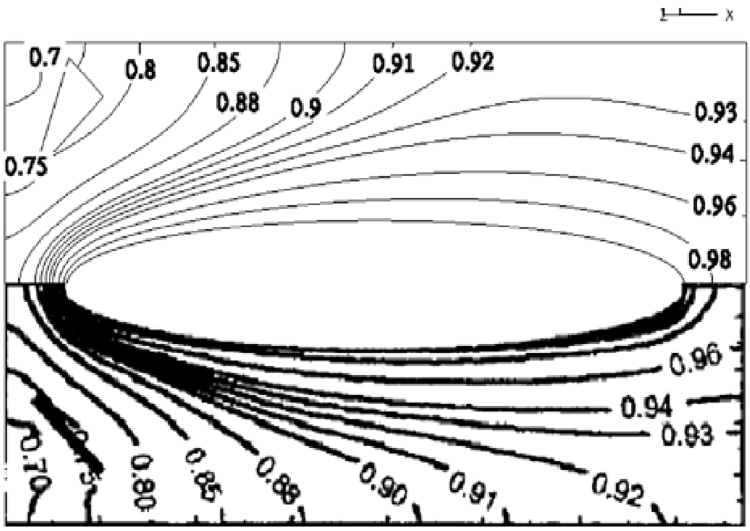

Fig. 3.3 Dimensionless temperature distribution on the fin, *(T –T*in)/(*T*tube –*T*in): present study (upper), then results [20] (lower)

Numerical procedure

The conservative equations are solved using the Fluent commercial solver [31]. The volumetric entropy production rate obtained by post-processing is calculated with specifically developed subroutines. The equations have been linearized in an implicit form and solved sequentially. The SIMPLEC algorithm has been used for the pressure-velocity coupling. The pressure interpolation scheme has been chosen to be a second order upwind scheme. The momentum equation has been discredited with a power law scheme and the energy equation with a second orderscheme.In order to calculate the convection and diffusion terms constituted by a derivative function, the Green Gauss theorem has been used. This face value is calculated as the arithmetic average of the values at the neighboring cell centers.

Computational domain and boundary conditions

The computational domain is shown in Fig. 2. The *x*-axis is aligned with the main flow direction. The *y*-axis is the cross direction, and the *z*-axis corresponds to the direction perpendicular to the fins. The upstream numerical domain has an $L/2$length and the downstream numerical domain has a $2L$ length.

The boundary conditions are the following:

- At the inlet, the temperature is set to 293 K and the velocity is specified such as it corresponds to a Reynolds number equal to 300.
- At the outlet the atmospheric pressure is fixed.
- The punched holes in the fins are assumed to be periodic boundary conditions in the *z*-direction.
- A heat flux equal to zero is fixed on the outer side of the fins.
- On the left and right sides of the finned oval tube element symmetric boundary conditions are set.
- The thickness of the wall tube is not taken into account.
- The boundary conditions are set according to the input conditions defined.
- The flow unsteadiness is responsible of temperature and velocity fluctuations, hence of entropy production fluctuations

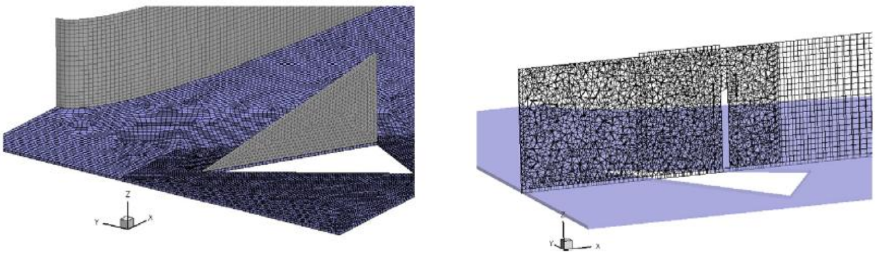

Fig.3. 4Detailed view of the surface mesh and the volume mesh around a delta winglet.

As we focus only on the air-side performances, the convective heat transfer in the tube, the conduction in the tube wall and the contact resistance at the fin tube junction are not considered and the tube outer surface is assumed to beisothermal.In order to optimize the mesh size and quality, the mesh domainis divided in three zones. The first one is localized around the delta winglets (Fig. 3.4). It is made of tetrahedral elements and is refined near the wall. The second zone takes into account the rest of the air side. Hexahedral elements have been used. The third one corresponds to the fin. The material of the fin corresponding to $Fi = 500$ is aluminum. In this case the Biotnumber is small with respect to unity. Thus the heat conductionis assumed to be two-dimensional in the fin. Therefore the meshing the fin contains only a hexahedral cell in the thickness. The mesh quality as required in [31] is such as the equianglesize factor and aspect ratio factor of the cells are respectively below 0.85 and 5. The results grid independency has by comparing three meshes with 1.6, 2.8 and 4.5million of cells. Skin friction coefficient values, Nusselt numbers, local and global entropy production rates have been shown to converge with the number of mesh elements (see Table 1). Finally the finest mesh has been chosen for the present numerical study.

Temporal discretization

Because of the vortex shedding downstream the tube and the unsteadiness of the longitudinal vortices (Fig. 5 (a) and (b)),an unsteady numerical formulation is necessary. A first order implicit scheme is chosen. The monitoring of the flow velocity shows that the wake downstream the tube and the secondary flow generated by the vortex generators are unsteady periodic structures. The necessary time step considered to capture the velocity and temperature fluctuations is8×10^{-5} s, corresponding to 30 samples per period. The initial conditions are such that dynamic and thermal fields are uniform on the computational domain at the initial time step. Equations for continuity, momentum and energy have been solved until the establishment of the periodic oscillations and also the convergence of the residual criteria is reached [31] for each time step. The residual criteria are set to 10^{-6} for continuity equation and 10^{-8} for momentum and energy. The flow unsteadiness is responsible of temperature and velocity fluctuations, hence of entropy production fluctuations. As the outlet and the inlet domain are not considered for the evaluation of $\dot{S}$, the periodic evolution of $\dot{S}$ in the process of time is supposed to correspond only to the unsteadiness of the longitudinal vortex confined between the fins. The arithmetic time average of the entropy production rates, and of all the other physical values considered here, is based on the two last calculated periods of the established periodic phenomena observed. Anyway, the time fluctuations of the entropy production are negligible with regards to the variation of entropy production due to a parametric modification. In fact, the variations of $\dot{S}$ around the averaged value are below 0.01% for all the configurations studied, which is significantly smaller than the parametric variations studied in the following sections.

III. Results And Discussions

The streamlines in the mid-plane of the channel of the study area for different inlet angle at *Re* = 7,000. The flow with inlet angle past the oval tube and it can be clearly found out that the wall in the direction of tube arrangement has periodic boundary condition which is plotted in Figure 5. It can be seenthat the flow is separated by the oval tubes and clearly exhibits the phenomenon of vortex near the rear ofthe tube. Meanwhile, the location of stationary point is shifted upward as well as the inlet angle decreases and the velocity gradient is also changed in the channel. Such behavior can be attributed to the periodicalflow condition in vertical direction which results that the downstream flow is affected by upstream flow. Furthermore, it can be seen that the region of vortex enlarges obviously with the decrease of inlet angle. Besides, the sequence of pictures illustrates that the average flow velocity has been increased because of
the enhancement of vortex.

IV. Conclusions

In this project, a two-row heat exchanger unit model has been established and the inlet angle effects of plain fin-and-oval-tube heat exchanger have been investigated by FLUENT software. Some major conclusions are drawn as follows:

1. With the variation of inlet angle, the streamline has being changed remarkably. The layout of Vortex effected by inlet angle obviously and the hydrodynamics determine the distribution of the local Nusselt number.
2. The Nusselt number increases 16.7 % averagely at most for large *Re* comparing θ=30° with θ=90°. Meanwhile, the pressure drop increases about 57.8 % at the same time. Because of the excellent capability of heat transfer, the arrangement of θ=30° is frequently used for industrial applications as the pressure loss is acceptable.
3. The Nusselt number increases 18.5 % averagely at most for large *Re* comparing θ=25° with θ=90°. Meanwhile, the pressure drop increases about 62.9 % at the same time. Because of the excellent capability of heat transfer, the arrangement of θ=25° is frequently used for industrial applications as the pressure loss is acceptable.
4. Comparing the overall performance of different θ reflected by the ratio of *j* factor and *f* factor, the trend shows that 45° have an excellent performance. Meanwhile, the advantages are more obvious with the increase of *Re*.

A three-dimensional computational study of forced convection heat transfer has been accomplished to determine the flow structure and heat transfer in a rectangular channel with a built-in oval tube and delta winglet type vortex generators in various configurations. The duct was designed to simulate a passage, formed by two neighbouring fins in a fin–tube heat exchanger. The present study reveals that combinations of oval tube and the winglet pairs improve the heat transfer significantly, especially in the dead water zone. The mean span-averaged Nusselt number for the case of four winglet pairs, each two in sequence having a staggered configuration (Inner pair in common-flow-down and outer pair in common-flow-up arrangement) is about 100% higher as compared to no-winglet case at a Reynolds number of 1000. The enhancement in heat transfer, on the basis of finned oval tube as the base line case, is 43.86% for the case of two winglet pairs in staggered mode. A comparison of heat transfer for the cases of one, two and three winglet pairs (all in common-flow-down configuration) Confirms that the addition of each extra winglet pair causes further enhancement of heat transfer. The enhancement of heat transfer is marked even at far downstream locations. The winglets, at their moderate angle of attack, have quite streamlined like behaviour and so, are not expected to contribute much towards pressure losses. On the other hand, the contribution towards enhancement in heat transfer due to the winglet pairs is undoubtedly significant

V. Scope For The Future

With the growing technology and demand for the heat transfer equipments this would be a fortune effect for the enhancement of heat transfer in the field of thermodynamics. This will be a beginning for the development of new materials for obtaining higher efficiencies in all the fields of science which are related to the transfer of energy. It is proved from the above project that the heat transfer enhancement is not only specified for some fields it also differentiates so many problems in the field of heat transfer.

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