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Numerical Analysis of Winglet Type Fin-and-Tube Heat Exchanger

Shailesh Kumar Sarangi*, Dipti Prasad Mishra

Department of Mechanical Engineering, Birla institute of technology, Mesra, Ranchi-835215, India.

Abstract: Three dimensional conservation equations of mass, momentum and energy are solved numerically to determine the heat transfer and fluid flow characteristics of a rectangular winglet type fin-and-tube heat exchanger. The flow is assumed to be laminar and the surface of the tube is maintained at a constant temperature. The winglets are placed in "common flow up" configuration. This configuration results considerable separation delay and improves heat transfer in the near wake of the tubes. A moderate attack angle (10°) of the winglet has been used so far in this study and it is expected that higher angle of attack will result in higher heat transfer performance. In comparison to the baseline case (without winglets) it is found that the heat transfer performance increases significantly by using winglets with rise in pressure drop. It is also found that by increasing the number of winglets and changing the position of winglets has a significant effect on heat transfer performance.

Keywords: Fin-and-tube heat exchanger, Winglets, Heat transfer enhancement

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1 Introduction: Fin-and-tube heat exchangers has various applications such as refrigeration and air-conditioning equipments, petrochemical industries, heating and electronic cooling. Improving the heat exchanger performance is very important in meeting the higher efficiency of the equipments. Due to the low heat transfer coefficient of air, these heat exchangers have higher air side convective thermal resistance compared to thermal resistances offered by the wall side and liquid flowing inside the tube. Therefore in order to improve the performance of these heat exchangers attention should be focused to improve the air side heat transfer by providing winglets to avoid flow separation. Laminar flow usually results in low heat transfer coefficients. The fluid velocity and temperature vary across the entire flow channel so that the thermal resistance is not just near the wall as in turbulent flow. Hence small scale surface roughness is not effective in laminar flow. Wings and winglets can drive the flow towards the tube surface causing the no heat transfer zone behind the tube to get compress resulting in heat transfer enhancement.

Considerable amount of research work (both numerically and experimentally) has been done on the topic of heat transfer enhancement using vortex generators. Edward and Alker (1974) [1] experimentally studied the effect of longitudinal vortices on heat transfer performance. They found that the delta winglets provide higher heat transfer performance compared to cubes placed on a flat plate. Patankar and Prakash (1981) numerically investigated the effect of plate thickness on flow field and heat transfer. They found the flow field to be complex containing recirculation zones behind the trailing edges of the plates, and there occurs significant deflection of the through flow [2]. Russel et al. (1982) compared experimentally about inline and staggered arrangement of winglets and reported rectangular winglets to be more effective compared to delta winglets in two staggered rows arrangement [3]. Fiebig et al. (1986) experimentally compared different types of winglet geometries such as delta wing, rectangular wing, delta winglet pair and rectangular winglet pair for varying laminar flow and found delta winglet to be the most effective from heat transfer point of consideration [4]. Tiggelbeck (1991) experimentally determined that the second delta winglet pair acts as a booster for the coming longitudinal vortices in a two row inline arrangement of delta winglet pairs **[5]**. Fiebig (1995) experimentally and numerically studied the embedded vortices in internal flow and found that longitudinal vortices provides better heat transfer enhancement as compared to transverse vortices **[6]**. Biswas et al. (1996) experimentally and numerically studied the effect of longitudinal vortices on the flow structure and heat transfer performance. He found that the flow structure to be complex and consisting of a main vortex, an induced vortex and a corner vortex. Distortion of the temperature field takes place due to the resultant effect of all these vortices which helps in heat transfer augmentation. Torii et al. (1991) **[7]** experimentally investigated heat transfer enhancement by longitudinal vortices and found that delta winglet generates longitudinal vortices in the flow field that increases heat transfer and skin friction. Torii et al. (2002) experimentally investigated the heat transfer and pressure loss for inline and staggered tube arrangements and found that for Reynolds number (350-2100) the increase in heat transfer is 30-40% whereas the pressure loss is reduced by 40-50% in staggered tube arrangement **[8]**. Leu et al. (2004) experimentally and numerically investigated the flow structure and heat transfer performance in a plate-fin-and-tube heat exchanger with pair of inclined blocks as vortex generators and found greater heat transfer in the wake region of the tubes. Pesteei (2005) **[9]** experimentally investigated the effect of winglet location on heat transfer performance and pressure loss and found that the winglets are more effective when they are placed in the downstream side. Hiravennavar et al. (2007) numerically studied the winglet effects on the heat transfer performance and flow structure and found that in comparison to no winglet case heat transfer increases by 33% on using a single winglet and by 67% on using a winglet pair **[10]**. They also observed that winglet of finite thickness is slightly superior to the zero thickness winglet. Chu et al. (2009) numerically investigated the three row delta winglets type fin-and-oval-tube heat exchanger for Re = 500-2500**[11]**. They reported that, compared with the baseline case without longitudinal vortex generators, the average Nu with longitudinal vortex generators was increased by 14-33% and the corresponding pressure loss was increased by 29-40%, respectively. He et al. (2013) numerically investigated heat transfer and pressure drop in fin-and-tube heat exchangers with rectangular winglets **[12]**. They found that pressure loss of the heat exchanger can be reduced by changing the placement of the winglets from inline array to staggered array for the same amount of heat transfer.

From the literature review it appears that most of the previous studies were done taking single heat transfer element or tube. Also very few studies have been made on winglet position and their variation. Winglet geometry also affects the heat transfer performance and pressure loss and this also needs to be addressed. This paper focuses on all these areas. Numerical work is carried out to investigate the best winglet location and the effect of addition of winglets on heat transfer performance has been presented in this study.

2. Mathematical Formulation: In the present study a fin-and-tube heat exchanger without and with rectangular winglets is investigated. The schematic diagram of the fin-and-tube heat exchanger is shown in Fig. 1.

The rectangular winglets are placed symmetrically adjacent to the tubes. The height of the winglet is taken as 60% of the channel height. The winglets are positioned in “common flow up” orientation. The computational geometry with the dimensions are shown in Figure 1. The fluid flows is in X direction. The channel height (H) is 3.63 mm, flow length is 76.2 mm and span length is 12.7 mm. The fin material is aluminum and its thickness is 0.18 mm. the first tube is placed at a distance of 12.7 mm from the inlet and distance between the two successive tubes is 25.4 mm. The dimensions are taken from literature by He et al. (2013). Due to the geometry character of symmetry the region indicated by dashed lines in Figure (1) is selected as the computational domain. Since the heat transfer coefficient of the fluid flowing inside the tube is very high and due to the high thermal conductivity of the tube wall, the tube temperature is kept constant. The winglets with height 2.18 mm, length 10.67 mm and thickness of 0.18 mm are placed near the tubes as shown in Figure (2).

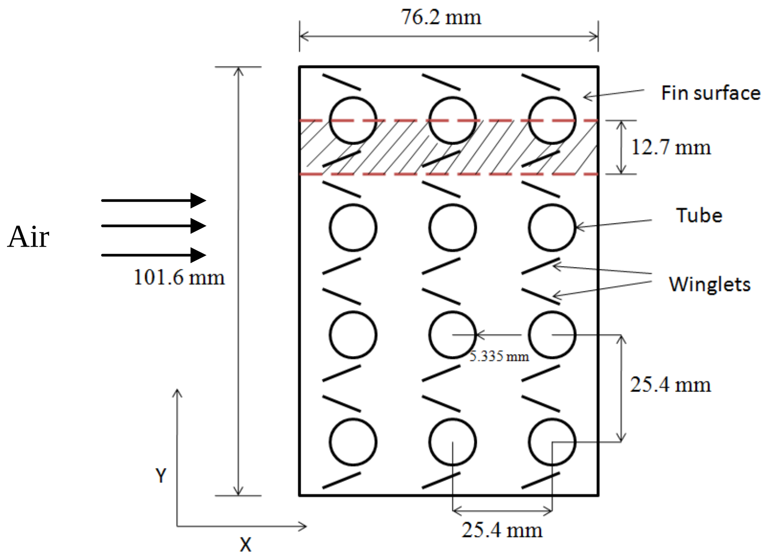


Figure (1): Top view of the computational domain

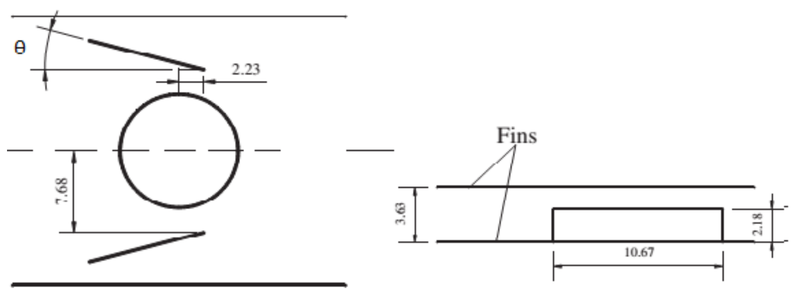


Figure (2): Placement of winglets near the tube wall

2.1 Governing Equations: The governing equations can be written as follows:

Continuity equation: $\frac{\partial}{\partial x_i}(\rho u_i) = 0$ (1)

Momentum equation: $\frac{\partial}{\partial x_i}(\rho u_i u_k) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_k}{\partial x_i}\right) - \frac{\partial p}{\partial x_k}$ (2)

Energy equation: $\frac{\partial}{\partial x_i}(\rho u_i \mathrm{T}) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial \mathrm{T}}{\partial x_i}\right)$ (3)

2.2 BOUNDARY CONDITIONS:

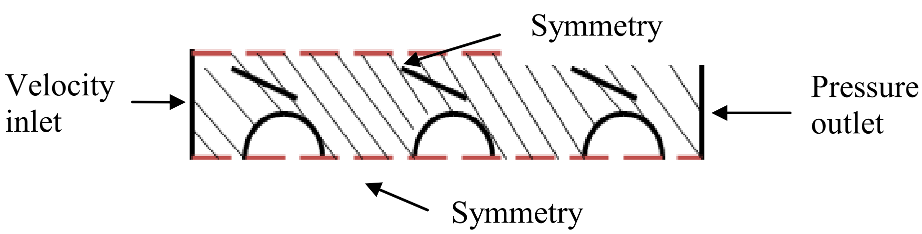


Figure (3): Boundary conditions from top view

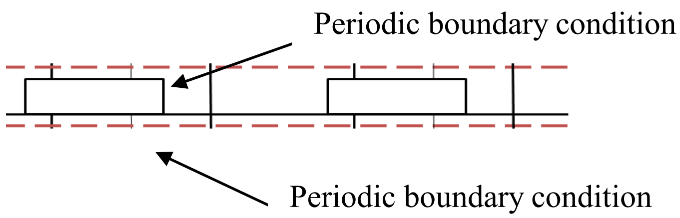


Figure (4): Boundary conditions from side view

Domain inlet

- At the inlet boundary

The air is assumed to have uniform velocity u_{in} and uniform temperature T_{in}.

The velocity components in the Y and Z directions are assumed to be zero, v = w = 0.

- At the upper and lower boundaries

Periodic boundary conditions, $u_{up} = u_{down}$ and $T_{up} = T_{down}$.

- At the side boundaries

Symmetric boundary conditions, $\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0$, $v = 0$, $\frac{\partial T}{\partial y} = 0$

Domain outlet

- At the upper and lower boundaries

Periodic boundary conditions, $u_{up} = u_{down}$ and $T_{up} = T_{down}$.

- At the side boundaries

Symmetric boundary conditions, $\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0$, $v = 0$, $\frac{\partial T}{\partial y} = 0$

- At the outlet boundary

Pressure outlet boundary was imposed at the domain outlet i.e. $p = p_a$.

Fin surface region

- At the upper and lower boundaries no slip and coupled boundary condition was given.

Tube surface

- The surface of the tube is imposed with no-slip and constant temperature boundary condition.

2.3 Numerical metods: GAMBIT 2.4.6 is used to create the model drawing and to generate the mesh. Governing equations and boundary conditions are solved by commercial CFD code (FLUENT 14). Because of the complex nature of the computational domain, it is difficult to use a single structured hexahedral mesh in the complete computational domain. The computational domain is shown in Figure (1). The fin surface, rectangular winglets, and the lower volume are meshed with hexahedral/wedge elements, the upper volume is meshed with T-grid elements. First order upwind-scheme is used to discretize the convective terms appearing in governing equations for momentum and energy. SIMPLE algorithm is used to couple Velocity and pressure. The criterion of convergence for the velocities is that the maximum mass residual of the cells divided by the maximum residual of the first 5 iterations is less than 1.0×10^{-5}, and the criterion of convergence for the energy is that the maximum temperature residual of the cells divided by the maximum residual of the first five iterations is less than 1.0×10^{-8}.

3. Results and discussion: The simulation parameters are summarized below:

Inlet temperature: 310.6 K

Reynolds number: 648

Tube wall temperature: 291.77 K

3.1 Grid Test: To make sure the accuracy and validity of the numerical results a grid test is performed for 6 set of grid numbers. They are about 4,45,000, 5,29,000, 750,000, 16,40000, 18,40,000 and 20,14,000 cells respectively. The temperature at the outlet almost remains constant for grid numbers of 16, 40,000 and above. Thus to maintain a balance between computational economics and accuracy the adopted grid number for the present study is 16, 40,000. This grid test is performed for a fin-and-tube heat exchanger with two winglets.

3.2 Effect of number of winglet pairs: A comparative investigation has been done for fin-and-tube heat exchanger with different winglet pairs to realize the effect of number of winglets on the heat transfer performance. The inlet velocity is maintained at 1.87 m/s and the angle of attack is kept 10°.

When a single winglet is used it drives the flow towards the adjacent tube as a result the temperature gradient becomes larger at the rear of the tube compared to the no winglet case. On comparing the single winglet and two winglet case it can be seen that the tube wake size is about the same behind the first tube. However the tube wake size reduces behind the second tube for the two rectangular winglet pair's case because of the impingement of the accelerated fluid on the adjacent tube surface caused by the rectangular winglet pairs causing more heat transfer. In Figure (6b) variation of Nusselt number with number of winglets is shown. Winglets drive the flow towards the tube. With a single rectangular winglet pair the heat transfer increases by 24-25 % compared to the no winglet case. When two winglets are used the increase in heat transfer becomes 62-63%. With three winglets in operation the enhancement in heat transfer becomes 77-78% compared with the no winglet case. Thus with increase in number of winglets heat transfer increases significantly.The results presented in Figure (6c) indicates the friction losses associated with the heat exchangers with different number of rectangular winglet pairs. In comparison with the no winglet case, the single rectangular winglet pair case increases the friction loss by 19-20 %. In two rectangular winglets pair case the friction loss inceased by 31-32 % and that for three winglet pairs incraesed by 38-39%. The additional frictional loss is due to the drag offered by the winglets.

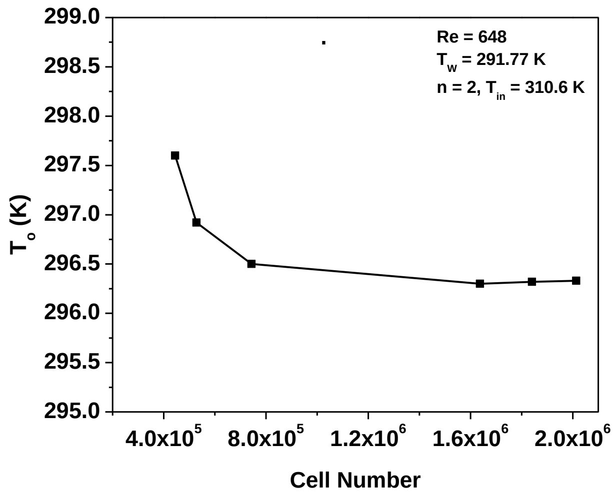


Figure (5): variation of outlet temperature for different cell numbers

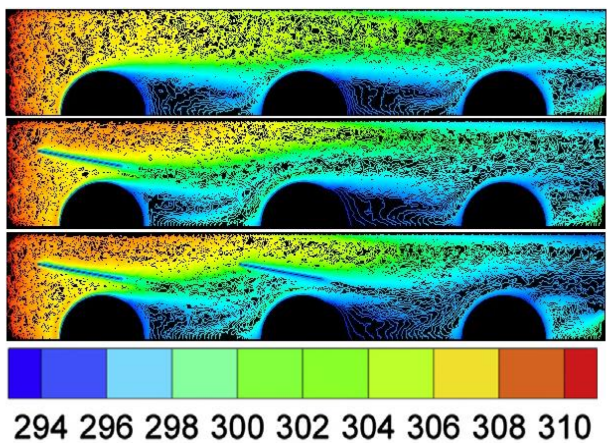


Figure (6a): variation of temperature at fin cross-section

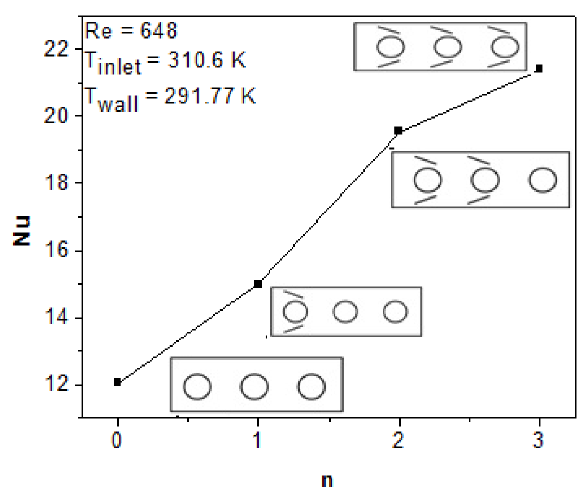


Figure (6b) : variation of Nusselt number with number of winglets

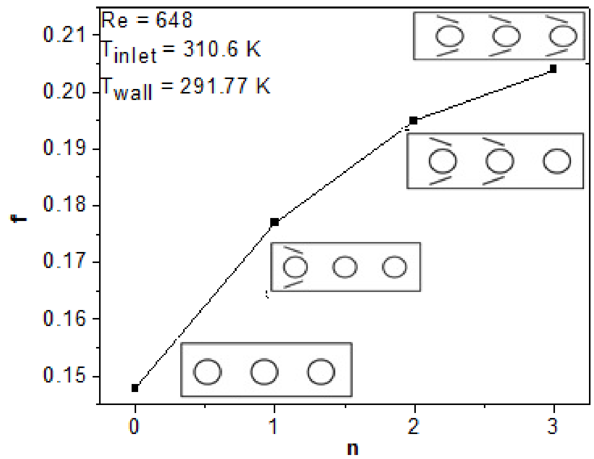
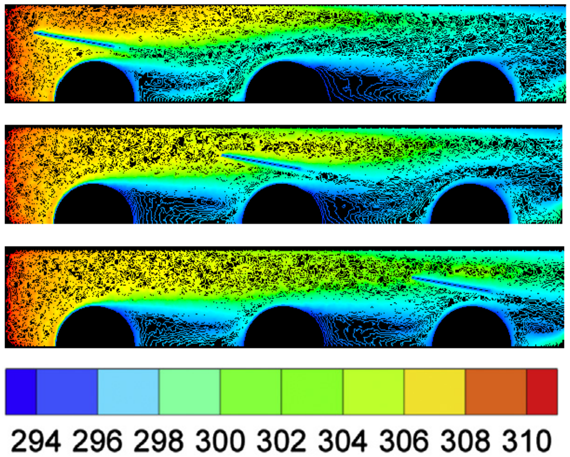


Figure (6c): friction losses with different number of rectangular winglet pairs

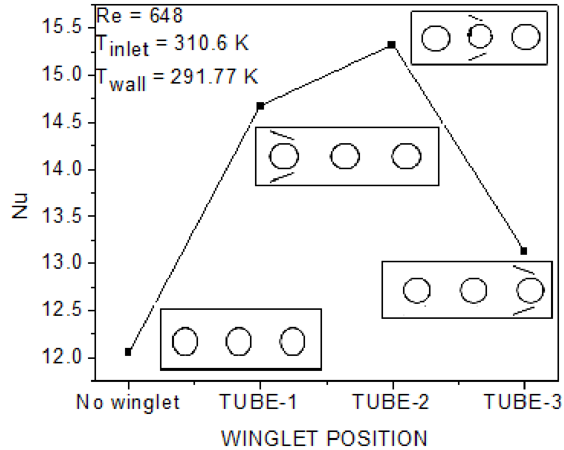
3.3 Effect of arrangement of single rectangular winglet pair: In order to examine the effect of rectangular winglet pairs position on the heat transfer performance of the fin-and-tube heat exchanger a comparative investigation is done for fin-and-tube heat exchanger with winglet pairs placed at first , second and third tube. The angle of attack is 10° and the Reynolds number is 648.

Figure (7a) indicates the local temperature distribution on the fin surface for different winglet positions. We can see from the figure that on placing the winglet near the tube narrowed the wake region behind the tube as result the local heat transfer increases. Figure (7b) gives the variation of Nusselt number with different winglet arrangements. The outlet temperature for winglet position one and two are about the same with winglet pairs in position two giving a little higher heat transfer. The incoming flow is separated into two parts by the rectangular winglet pairs. One part of the flow moves towards the adjacent tube whereas the other part flows towards the other side of the winglet. Thus the flow moving to the other side of the winglet has higher temperature compared to the one moving towards the tube. After a certain minimum distance these two separated parts gets mixed and heat transfer between the two streams of fluid takes place. In case of winglet at second tube position the incoming flow after passing through the first tube loses some of its heat before striking the winglet. When the flow gets separated by the winglet the flow moving to the other side of winglet has already lost some part of its heat and after travelling a certain distance it mixes with the flow coming from the tube side and its temperature further reduces. In the third case i.e. winglets at tube position three a very small part of the flow flows towards the tube with majority of the flow flowing with relatively higher temperature. Thus the resultant temperature obtained at the outlet is higher than the earlier two cases. Thus winglet placed near tube two gives the maximum heat transfer.

For the case when winglet is placed near the second tube the flow loses its velocity along the flow direction while passing through the tubes and consists of both u and v components of velocity. Thus pressure loss obtained at outlet is more in case of winglets placed near tube one compared to the other two arrangements. The increase in friction factor for winglet placed near first tube position is 20 % and that for the winglet placed near the third tube is 5 % compared to the no winglet case. For the winglet placed near the second tube the increase in friction factor is found to be 13 %.



(a)



(b)

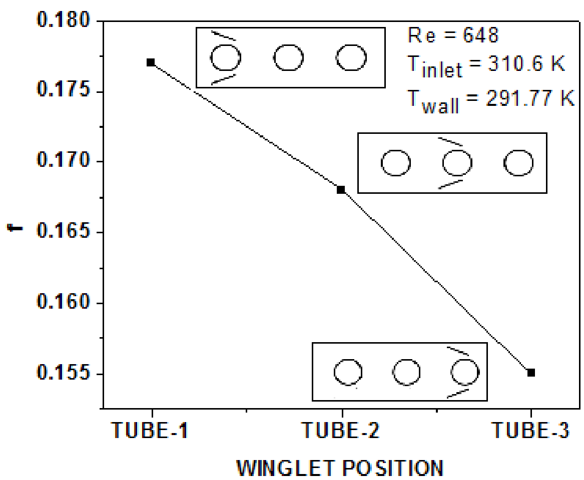


Figure (7): (a) Temperature distribution on the fin cross section , (b) variation of Nusselt number with different winglet positions, and (c) variation of friction factor with different winglet position

3.4 Effect of arrangement of two rectangular winglet pairs

In order to investigate the effect of two rectangular winglet pairs on the heat transfer performance of fin-and-tube heat exchanger two rectangular winglet pairs are placed at different positions. The angle of attack is 10° and the Reynolds number is 648.

Figure (8a) gives the variation of Nusselt number with different winglet arrangements. When the winglets are placed near first and second tube the flow gets separated by the winglet placed near first tube. One part of the flow flows through the narrow passage between the winglet and the tube outer surface. This accelerated flow again encounters winglet near the second tube position and its temperatures drops significantly after Passing through the tube surface. For Winglets placed near tubes one and three the temperature at outlet is little higher because a very small part of the flow passes through the nozzle like passage near tube three. For the case of winglets at tube position two and three again a very small part of the flow passes through the narrow passage at tube two and three and the resultant temperature obtained at the outlet is little higher than the earlier cases.

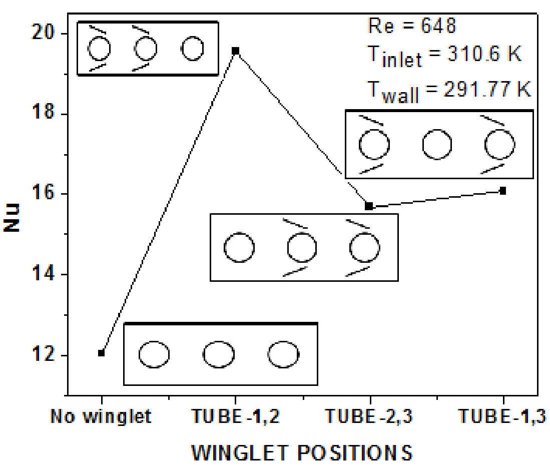


Figure (8a): variation of Nusselt number with different winglet positions

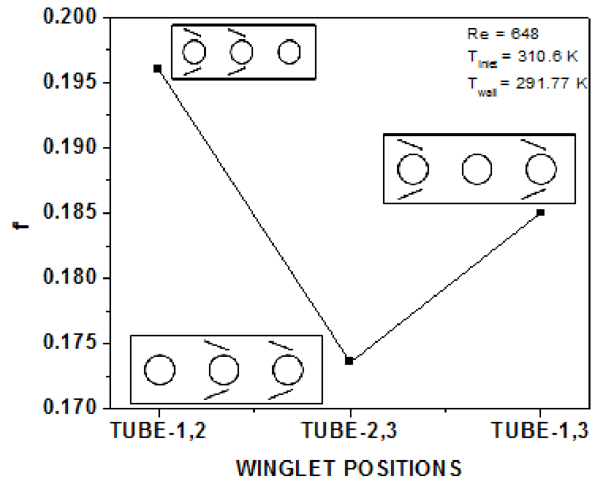


Figure (8b): variation of friction factor with different winglet positions

Figure (8b) presents the variation in friction factor with different winglet arrangements. The flow at inlet has maximum velocity and has velocity components along x-direction only.When the flow approaches the winglet located near the first tube its velocity reduces significantly. For winglet placed near second or third tube the incoming flow sufferes a loss in velocity after passsing the first tube due to wall friction and the majority of the flow elements are not parallel to flow direction. So the pressure loss is less in case the winglets are placed near tube two and three. It is seen that there is an increase of 33 % in friction factor when the winglets are placed near tube one and tube two. When winglets are placed near the tube two and tube three the friction factor increases by 18 %. Whereas the increase in friction factor is 25 % for the case when winglets are placed near tube one and tube three.

4 Conclusions: The present study is carried out using three dimensional numerical simulations to find the heat transfer performance and pressure loss in a fin-and-tube heat exchanger with rectangular winglet pairs. The simulations are performed at a fixed Reynolds number 648. The performance of the heat exchanger has been evaluated based on the comparison of Nusselt number and friction factor.Due to the “common flow up position” of the winglets a nozzle like passage is formed between the winglets and the outer surface of the tube. The accelerated flow through this passage delays the boundary layer separation of the fluid as well as removes the zone of poor heat transfer from the tube wake. This accelerated flow also impinges on the downstream tube which causes enhancement of local heat transfer.For a single winglet pair the rise in heat transfer is 25 % and the corresponding pressure loss is 19 %. When two winglets are employed the increase in heat transfer is found to be 63 % and the corresponding pressure loss is found to be 33 %. When three winglets are used the heat transfer enhances by 78 % and rise in pressure drop obtained is 39 %. Therefore increasing the number of winglets can increase the heat transfer performance of the heat exchanger significantly. However pressure loss of the heat exchanger also increases.With a single rectangular winglet pair placed near the first tube the heat transfer increases by 24-25 % with afriction loss 20 % compared to the no winglet case. When it is placed near the second tube the increase in heat transfer becomes 62-63% with 13 % rise in friction factor . when the rectangular winglet pairs are placed near the tube three the enhancement in heat transfer becomes 77-78% with 5 % increase in friction factor compared with the no winglet case. So placing the winglet pairs near the second tubes gives the best heat transfer performance.When the winglet pairs are placed near the tubes one and three the increase in Nusselt number is found to be 30 % and when they are placed near tubes two and three the Nusselt number increases by 34 %. When winglet pairs are placed near the tubes one and two the Nusselt number increased by about 63 % compared to the no winglet case. Thus on placing the winglets near the tube one and tube two gives the best heat transfer performance. However the winglets placed near the first two tubes causes more loss in pressure compared to the other two arrangements.

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